Study Of Heat Transfer Characteristics Of Impinging Air Jet Using Pressure And Temperature Sensitive Luminescent Paint

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STUDY OF HEAT TRANSFER CHARACTERISTICS OF IMPINGING AIR JET USING PRESSURE AND TEMPERATURE SENSITIVE LUMINESCENT PAINT

by

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ABSTRACT

Luminescent coating measurement system is a relatively new technology for quantitative pressure and temperature measurement. Usually referred to as Pressure Sensitive Paint (PSP) and Temperature Sensitive Paint (TSP), luminescent coatings contain sensor molecules, which undergoes a luminescent transition when excited with light of proper wavelength. The reaction is pressure and/or temperature sensitive. The image of TSP or PSP coated model surface can be captured with a scientific grade camera and then processed to obtain full field temperature and pressure distribution with very high fidelity. The preparation time of the technique is short. The measurement system offers an economic alternative to conventional testing methods using large number of pressure taps and thermocouples. The purpose of the experiment in this thesis is to take the benefits of the TSP and PSP technique, develop a well-controlled process and then apply the technique for a fundamental study on jet impingement heat transfer.

First, Uni-Coat TSP and Binary-FIB PSP purchased from ISSI Inc. are calibrated to high accuracy. The calibration uncertainty of TSP and PSP are found to be ±0.93 °C and ±0.12 psi over temperature and pressure ranges of 22 to 90 °C and 5 to 14.7 psia, respectively. The photodegradation of TSP is then investigated with the same calibration system. The photodegradation refers to the phenomenon of decreasing emission intensity as the luminescent paint is exposed to the illumination light during testing. It was found that photodegradation rate is a strong function of temperature and the optical power of illumination lighting. The correlation developed in this work is expected to compensate the degradation of TSP to achieve high measurement accuracy.
Both TSP and PSP were then applied in the flow and heat transfer measurement of single round impinging air jet. Various separation distance (Z/D) and jet Reynolds number are tested. Pressure measurement on the jet impinged target surface using PSP clearly shows the boundary of jet impingement zone, which broadens with separation distance. In heat transfer experiment using TSP, the “second peak” in local heat transfer occurring at radial distance r/D around 2 is clearly observed when the separation distance Z/D is shorter than the length of jet potential core. The slight variation in radial location and the amplitude of the “second peak” are captured as Z/D and jet Reynolds number change. The optimum Z/D of stagnation point heat transfer is found to be around 5. The effect of jet nozzle configuration is investigated. It is found that the heat transfer rate associated with “tube jet” is generally higher than that of “plate jet”. The difference in heat transfer between the two jet configurations is related to the weaker entrainment effect associated with “plate jet”, where the entrainment of surrounding air is confined by the injection plate, especially under small Z/D circumstances. When compared with the benchmark data in the literature, the averaged heat transfer data of “tube jet” matches the empirical data better than those of “plate jet”. The maximum difference is 3.3% for tube jet versus 15.4% for plate jet at Reynolds number of 60000 and Z/D of 5. The effect of surface roughness on jet impingement heat transfer is also studied. Heat transfer can be significantly increased by the enhanced roughness of the target surface. The largest roughness effect is achieved near stagnation point at high jet Reynolds number. Compared to the heat transfer to a smooth plate, as high as 30.9% increase in area-averaged Nusselt number is observed over a rough surface at r/D=1.5 and jet Reynolds number of 60000.
The most significant advance of the present work is that both temperature and pressure measurement be obtained with the same measurement system and with accuracy comparable to traditional testing methods. The procedures that were employed in this work should be easy to apply in any university or industrial testing facility. It provides a rapid testing tool that can help solve complex problems in aerodynamics and heat transfer.
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CHAPTER 1 INTRODUCTION

1.1 Impingement Cooling

Jet impingement heat transfer has become well established as a high performance technique for heating, cooling, or drying a surface. Engineering applications of jet impingement include annealing of metal, tempering of glass, drying of paper, cooling of electronic equipment, freezing of tissue in cryosurgery, anti-icing system for forward facing surfaces of civil aircraft, cooling of gas turbine components and the outer wall of combustors. Both empirical and theoretical studies in the topic are unabated and may have even accelerated in recent years, mainly stimulated by the need of higher temperatures in the gas turbine industry.

Direct impingement of turbulent jets onto a surface leads to high local heat transfer rate. It is often employed to achieve rapid heating or cooling. Jet impingement heat transfer has been extensively studied to determine both the peak and the spatial heat transfer distribution for various configurations of jets and surface. The impingement of unconfined axisymmetric (circular) and slot (two-dimensional) jets on a flat surface have received most of the research attention.

In gas turbine systems, jet impingement cooling is finding increased use as higher turbine inlet temperature are utilized due to the fact that thermal efficiency and power output of gas turbines increase with increasing turbine inlet temperatures. For turbine components including turbine guide vanes, rotor blade, endwalls, rotor disks and combustor walls, impingement cooling provides one of the useful ways to prevent overheating. However, the complex geometry
of turbine systems, the high turbulence and the rotation of the systems make the understanding of flow and heat transfer characteristics of impinging jets a challenging subject.

1.2 Luminescent Temperature and Pressure Sensitive Paint

Surface pressure and temperature distributions are of fundamental importance in almost all fluid mechanics and heat transfer experiments. The traditional method utilizes arrays of thermocouples and pressure taps embedded in a model for surface temperature and pressure measurement. This calls for time-consuming model preparation and tedious data collection. In addition, measurements are made only at discrete points. Spatial resolution is limited by the number of sensors instrumented. It is almost impossible to instrument thin edges and sharp corners on a model, and these are often the areas of most interest.

New techniques for temperature and pressure measurement, which are based on luminescence quenching, have been developed in recent years. These new sensors are named temperature sensitive paint (TSP) and pressure sensitive paint (PSP). A schematic of typical TSP/PSP measurement system as well as paint structure is shown in figure 1.1. the main components of this system for wind tunnel test are a model coated with the luminescent paint, an excitation light source, a digital camera used to acquire images, and a computer used for image processing. Both TSP and PSP incorporate luminescent molecules in a paint together with a transparent polymer binder. Light of the proper wavelength is directed at the painted model to excite the luminescent molecules. The sensor molecules become excited electronically to an elevated energy state. The molecules undergo transition back to the ground state by several mechanisms. The predominant mechanism is radiative decay (luminescence). Sensor molecules
emit luminescent light of a longer wavelength than that of the excitation light. Absorption and excitation wavelength of a typical PSP is shown in figure 1.2. Proper filters can separate excitation light and luminescent emission light and the intensity of the luminescent light can be determined using a photodetector. The excited energy state can also be deactivated by quenching processes. Through two important photo-physical processes known as thermal- and oxygen-quenching, the luminescent intensity of the paint emission is inversely proportional to local temperature and pressure.

For TSP, polymer binders are not oxygen permeable and hence TSP is not pressure sensitive. In contrast, a good polymer binder for PSP must have high oxygen permeability and thus PSP has both temperature and pressure sensitivity.

In principle, a full spatial distribution of the surface pressure and temperature can be obtained by using PSP/TSP technique. A PSP data-mapped transonic wing is shown in figure 1.3.

1.3 Objectives

Given the potential benefits of PSP and TSP measurement system, a research program was undertaken investigating the capabilities of this technology in the study of heat transfer characteristics of impinging air jets to a flat target surface. The objectives of this research are as follows:

1. TSP/PSP calibration
   a. Calibration techniques and error analysis
   b. Investigation of TSP degradation
2. Single round jet impingement test: apply luminescent coating to the jet impingement test rig and test at a range of jet Reynolds number and various nozzle-to-target plate separation distance (Z/D). In this effort, it is desired to:

a. Develop a well-understood and controlled process for preparing jet impingement test models; conducting experiments, and performing data reduction which would lead to optimum results

b. Perform the jet impingement test with two different type of jet nozzle: long tube nozzle and orifice-in-a-plate type of nozzle. Compare both local and averaged heat transfer distribution between these two configurations.

c. Compare the obtained heat transfer data to the literatures; determine the capability of PSP and TSP systems at a typical air jet impingement test,

d. Study the effect of target surface roughness on the impingement heat transfer.

![Diagram of TSP/PSP measurement system, paint layer structure and typical dimensions](image)

Figure 1.1 TSP/PSP measurement system, paint layer structure and typical dimensions
Figure 1.2 Absorption (left) and emission (right) wavelength of a typical PSP

Figure 1.3 Pressure-sensitive paint data mapped to surface grid for transonic wing [1].
CHAPTER 2 MEASUREMENT TECHNIQUE OF TEMPERATURE AND PRESSURE SENSITIVE PAINT

2.1 Brief history of the development of luminescent coating system

The development of luminescent paint technique is a typical example of multidisciplinary approach to problem solving. Chemists and aerodynamicists collaborated to develop and improve this new technique in the past 30 years.

Russian scientists started research on PSP since the late 1970s at the central Aero-Hydrodynamic Institute in the former Soviet Union. The first pressure sensitive coating was developed jointly with the Italian firm INTECO in 1981 and obtained the first pressure measurements in 1982. INTECO’s system was demonstrated in several wind tunnel tests in the early 1990s in the United States.

Over the last decade, PSP research has spread to aerospace institutions all over the world. A group of chemists at University of Washington is among the first researchers developing PSP independently in the United States. Considerable work on the PSP technique has also been done at McDonnell Douglas, NASA-Ames, Boeing and Purdue University. Useful reviews on PSP include McLachlan & Bell [2] and Liu et al [3]. Low speed PSP applications are covered in details in a thesis by Brown [4].

Bell laboratory started TSP system development in the early 1980s. This system was adapted at Purdue University for aerodynamic experiment. TSP formulations were also studied by the same group at the University of Washington. Other methods for surface temperature maps include liquid crystals and infrared thermography. These two-dimensional techniques provide
surface temperature distribution at much higher resolution than thermocouples and each has its own advantages and drawbacks.

A complete bibliography of papers on PSP/TSP is maintained by Purdue University and is available online at [http://roger.ecn.purdue.edu/~psptsp/references/ReferencesPSPTSP.html](http://roger.ecn.purdue.edu/~psptsp/references/ReferencesPSPTSP.html)

![Jablonski energy-level diagram](image)

Figure 2.1 Jablonski energy-level diagram showing luminescence processes for a typical luminophore. ES, fluorescence; ET, phosphorescence [1].

2.2 Photophysical foundations and measurement technique

Like conventional paints, the luminescent paints consist of three basic elements: a “pigment”, a binder, and a solvent. The “pigment” in a conventional paint is added to provide color. The equivalent ingredient in PSP or TSP is the pressure and temperature sensitive
molecule, referred to as the “luminophore”. The luminophore in a typical PSP/TSP is an organic molecule.

To a casual observer, a PSP/TSP coating “glows” when placed under blue ultraviolet lights. This “glow” is part of a complex photo-luminescent reaction involving the luminophore. The physical processes of luminescence have to be explained in quantum level and are best described with Jablonski energy-level diagram shown in figure 2.1, named in honor of the Polish physicist Professor Alexander Jablonski. It shows different excited states of a luminophore molecule with the energy of each presented by its height above the ground state \( S_0 \). The electronic energy states are labeled \( S_0, S_1, T_1 \), etc. Multiple vibrational energy levels exist within each electronic state. \( S_0, S_1, S_2 \), etc. are singlet states and \( T_1, T_2 \), etc. are triplet states. The ground state of the luminophore is singlet and hence there is no \( T_0 \). In each electronic state, the molecule has its electrons arranged in a different combination of the available orbits and spin orientations—the latter distinguishing the singlet and triplet states. Intersystem (singlet-to-triplet and triplet-to-singlet) transitions are possible but with low probability.

When the luminophore is exposed to electromagnetic radiation of proper frequency, absorption occurs. The luminophore molecules are promoted to an energy level above the ground energy level \( S_0 \). This absorption process places the luminophore in a state \( S_n \), where “n” is 1, 2, 3… The energy absorbed \( \Delta E_a \), is given as

\[
\Delta E_a = \frac{hc}{\lambda_a} = h\nu_a
\]  

(2.1)

where \( h \) is Plank’s constant, \( c \) is the speed of light, \( \lambda_a \) is the wavelength of absorbed light, and \( \nu_a \) is the frequency of absorbed light. This relation also holds for radiative emission, in which case \( \Delta E_a, \lambda_a \) and \( \nu_a \) are replaced by \( \Delta E_e, \lambda_e \) and \( \nu_e \).
The absorption process can be described symbolically. The change in energy state of the luminophore molecule, denoted “M”, with an energy level in the subscript, can be written as:

\[ M_{S_o} + h\nu_a \rightarrow M_{S_n} \text{ (rate } k_a) \]  

(2.2)

where \( k_a \) is the absorption rate. The process is occurring in about \( 10^{-15} \) seconds, which is almost instantaneous.

Once at an excited state, the luminophore will go through a process of “internal conversion” to achieve an identical energy level within \( S_1 \). Then the luminophore molecule will relax to the lowest-energy vibrational state of \( S_1 \) through vibrational relaxation. In this process excess energy associated with the luminophore is transferred to the thermal motion of the surrounding medium. A rise in temperature of the luminescent molecule will increase the probability that the molecule return to the lowest energy level by this non-radiative process. This is known as thermal quenching and is the basis of temperature sensitive paint. The entire process of intersystem conversion and vibrational relaxation to the lowest singlet state occurs very rapidly, typically within \( 10^{-12} \) seconds.

Once in the lowest singlet state, there are three possible processes that can take place, which will finally return the luminophore to the ground state, \( S_0 \). First, the internal conversion, through which the energy change is:

\[ M_{S_1} \rightarrow M_{S_0} + \text{Heat (rate } k_n) \]  

(2.3)

the timescale of this non-radiative process, \( 1/k_n \) is greater than \( 10^{-9} \) seconds.

The second possible process is the emission process of fluorescence:

\[ M_{S_1} \rightarrow M_{S_0} + h\nu_f \text{ (rate } k_f) \]  

(2.4)
the emission wavelength, $\lambda_f$, is slightly longer than the excitation wavelength $\lambda_a$ due to the energy dissipated through vibrational relaxation process. The characteristic reaction time, $1/k_f$, is between $10^{-11}$ and $10^{-7}$ seconds.

The third path from the excited singlet state to the ground state occurs via intersystem crossing to the triplet state and is described as:

$$M_{S1} \rightarrow M_{T1} + \text{Heat} \quad \text{(rate } k_{st} \text{)} \quad (2.5)$$

$1/k_{st}$ is less than $10^{-12}$ seconds.

Once transition to the triplet state has occurred, the molecule will return to $S_0$ by one of several paths. First, the molecule may transit back to $S_1$ state via intersystem crossing:

$$M_{T1} + \text{Heat} \rightarrow M_{S1} \quad \text{(rate } k_{ts} \text{)} \quad (2.6)$$

From here, the molecule rapidly returns to the ground state by fluorescence, as described by Eq. (2.4), or internal conversion, as described by Eq. (2.3). The fluorescence process, named “delayed fluorescence”, has a longer lifetime since the molecule spends a finite amount of time within the $T_1$ state.

A second path of triplet to ground state transition is by non-radiative intersystem crossing:

$$M_{T1} \rightarrow M_{S0} + \text{Heat} \quad \text{(rate } k_c \text{)} \quad (2.7)$$

the transition occurs with a time scale ($1/k_c$) of $10^{-9}$ to $10^{-4}$ seconds.

The third path is a radiative process called phosphorescence:

$$M_{T1} \rightarrow M_{S0} + h\nu_p \quad \text{(rate } k_p \text{)} \quad (2.8)$$
the frequency $\nu_p$ of phosphorescence is lower than that of fluorescence $\nu_f$. The lifetime $(1/k_p)$ is lengthy, ranging from $10^{-6}$ sec to tens of seconds. The term luminescence includes fluorescence and the phosphorescence.

The forth path is through oxygen quenching:

$$M_{T1} + ^3O_2 \rightarrow M_{S0} + ^1O_2 \text{ (rate } k_q[O_2]) \quad (2.9)$$

$^3O_2$ and $^1O_2$ denote ground state triplet and singlet molecule oxygen, respectively. $[O_2]$ is the concentration of oxygen. Quenching is a non-radiative process that competes with fluorescence for deactivation of the $S_1$ electronic state. Specifically, quenching is a process of energy transfer from the luminophore at excited state to another non-luminescent molecule, called “quencher”, through collision. The fluorescence intensity is reduced with the presence of a quencher due to the transfer of excess energy to the quenching molecules. The competition between quenching and fluorescence allows the concentration of quencher to be determined. The ground state of an oxygen molecule has special characteristics that make ground-state oxygen an extraordinarily effective quencher. Other gases that are known to quench luminescence are not found in air. Thus the emission intensity of the luminescent molecule changes inversely with the partial pressure of oxygen, hence the pressure of air. This is the basis of pressure sensitive paint.

Equation (2.2) through (2.9) completely describe the possible paths a luminophore can travel through to return to ground state after excitation. A balance can be established for the rate of change of the number of molecules in the $S_1$ and $T_1$ state. The conservation equations are:

$$\frac{dn_{S1}}{dt} = n_{S0}k_a - n_{S1}(k_f + k_{st} + k_a) + n_{T1}k_s \quad (2.10)$$

$$\frac{dn_{T1}}{dt} = n_{S1}k_{st} - n_{T1}(k_p + k_c + k_q[O_2] + k_n) \quad (2.11)$$
In steady state, the rate of change will be zero. Combining these two equations yields:

\[ n_{S0}k_a = n_{S1}(k_f + k_a) + n_{T1}(k_p + k_c + k_q[O_2]) \]  (2.12)

The quantum yield (or quantum efficiency) of luminescence \( \Phi \) is defined in order to quantify the photophysical processes.

\[ \Phi = \frac{I}{I_a} \]  (2.13)

where \( I \) is the luminescence intensity and \( I_a \) is the absorption intensity. The absorption intensity \( I_a \) is described by the Beer-Lambert law

\[ I_a = I_{ex}(1 - 10^{-\varepsilon I_p}) \]  (2.14)

where \( \varepsilon \) is the molar absorptivity, \( c \) is the concentration of the absorbing species, \( I_p \) is the path length traversed by the light beam, and \( I_{ex} \) is the monochromatic excitation light intensity.

For a PSP, the quantum efficiency is described by a simple model

\[ \Phi = \frac{I}{I_a} = \frac{n_{S1}k_f + n_{T1}k_p}{n_{S0}k_a} \]  (2.15)

Combining equation 2.12 and 2.15,

\[ \Phi = \frac{I}{I_a} = \frac{n_{S1}k_f + n_{T1}k_p}{n_{S1}(k_f + k_a) + n_{T1}(k_p + k_c + k_q[O_2])} \]  (2.16)

For most of the PSP’s, phosphorescence overwhelms fluorescence. Hence, it is assumed that triplet state population far outnumbers that of the singlet state in the steady state condition, i.e. \( n_{T1} \gg n_{S1} \). So equation 2.16 simplifies to:

\[ \Phi = \frac{k_p}{k_p + k_c + k_q[O_2]} \]  (2.17)
The expression tells us that the relative brightness of the luminophore is a function of the oxygen concentration. This concentration is called the solubility. Henry’s Law states that solubility of a gas is in direct proportion to the partial pressure of the gas above the medium. In PSP the medium is the binder of the paint. Since the partial pressure of oxygen exists at a constant 21%, the solubility of oxygen in the binder can be decomposed mathematically:

\[ [O_2] = \sigma \chi P \]  \tag{2.18}

where \( \sigma \) is Henry’s Law solubility constant, \( \chi = 0.21 \) (the fraction of oxygen in a normal atmosphere), and \( p \) is the static air pressure at the painted surface. It is necessary to define a solubility rate constant “\( k_s \)”, where

\[ k_s = k_d \sigma \chi \]  \tag{2.19}

Substitute into equation 2.17, we have

\[ \Phi = \frac{I}{I_a} = \frac{k_p}{k_p + k_c + k_s p} \]  \tag{2.20}

A more useful expression can be obtained by referencing this equation to the vacuum condition (\( p = 0 \)).

\[ \Phi_z = \frac{k_p}{k_p + k_c} \]  \tag{2.21}

Hence, luminescent emission intensity at vacuum condition \( I_z \) is:

\[ I_z = I_a \frac{k_p}{k_p + k_c} \]  \tag{2.22}

Dividing equation 2.22 by 2.20, we get the following relationship:
\[
\frac{I_z(T_z)}{I(T)} = \frac{k_p + k_c(T) + k_s(T)p}{k_p + k_c(T_z)}
\]  \hspace{1cm} (2.23)

\(I_z\) is the intensity at state \((p=0, T_z)\) and \(I\) is the intensity at state \((p,T)\). The absorption intensity \(I_a\) is eliminated in above equation. If \(T_z=T\), then

\[
\frac{I_z}{I} = 1 + k_{sv}(T)p
\]  \hspace{1cm} (2.24)

where

\[
k_{sv}(T) = \frac{K_s(T)}{k_p + k_c(T)}
\]  \hspace{1cm} (2.25)

Equation (2.25) is named Stern-Volmer equation. \(K_{sv}\) is the Stern-Volmer constant. It represents quenching efficiency since it is the ratio between quenching rate and the sum of the radiative and non-radiative processes. This constant is derived by measuring the intensity, \(I\), at a known pressure. But in Stern-Volmer equation, the knowledge of intensity at vacuum pressure, \(I_z\), is neither desirable nor possible. Therefore, for practical application purpose, the ratioing technique is developed to simplify the pressure measurement process. Applying equation (2.23) at a reference condition \((p_o,T_o)\), we have

\[
\frac{I_z(T_o)}{I(T_o)} = \frac{k_p + k_c(T_o) + k_s(T_o)p_o}{k_p + k_c(T_o)}
\]  \hspace{1cm} (2.26)

Equation (2.23) can also be evaluated at state \((p, T)\), where \(p\) and \(T\) are to be measured. Similarly, we get the result:

\[
\frac{I_z(T_o)}{I(T)} = \frac{k_p + k_c(T) + k_s(T)p}{k_p + k_c(T_o)}
\]  \hspace{1cm} (2.27)

Dividing equation (2.26) by (2.27), a useful correlation is obtained:
\[
\frac{I_o(p_o, T_o)}{I(p, T)} = A(T, T_o) + B(T, T_o) \frac{P}{p_o}
\]  

(2.28)

where A and B are referred as “Stern-Volmer coefficients”. Since the absorption intensity \( I_a \) is eliminated in above equation, the effects of spatial non-uniformities of illumination, paint thickness, and luminophore concentration can be eliminated by taking the ratio \( I_o/I \). This greatly simplifies the application of the PSP and TSP technique. A more general expression is obtained by taking into account nonlinear effects

\[
\frac{I_o}{I} = \sum_{n=0}^{N} A_n(T) \left( \frac{P}{P_o} \right)^n
\]  

(2.29)

where coefficients ‘\( A_n \)’ have to be determined through calibration tests. The result is that an increasing pressure causes the intensity of the paint to decrease.

For a TSP, it is assumed that the paint layer is not oxygen-permeable so that \([O_2]=0\). Hence, the quantum yield is

\[
\Phi = \frac{I}{I_a} = \frac{k_p}{k_p + k_c}
\]  

(2.30)

\( I_a \) can be eliminated when dividing above equation by a reference intensity \( I_o \) and hence

\[
\frac{I}{I_o} = f_n(T; T_o)
\]  

(2.31)

The function \( f_n(T; T_o) \) can be determined by fitting calibration data with a polynomial, exponential or other function. The net effect is that fluorescent intensity decreases as temperature increases.
2.3 Measurement system of PSP and TSP technique

The essential elements of the measurement systems for both TSP and PSP include illumination sources, optical filters, photo-detectors and data acquisition/processing units. There are several measurement systems that have been developed and used so far. Each system has advantages over the others and suitable one need to be chosen to meet the specific requirements of the tests.

2.3.1 CCD Camera System

The CCD (Charge-Coupled Device) camera system is the most commonly used system for luminescent paint application. The paint is excited by an UV lamp or a laser. The luminescence intensity is filtered optically to eliminate the illuminating light and then captured by a CCD camera. The intensity image is then transferred to a computer for image processing. Both a wind-on image and a wind-off image are obtained. The ratioing technique described earlier yields a relative luminescent intensity image. The surface temperature or pressure distribution can be computed form the relative intensity image by using the calibration relations.

A necessary step in data processing is taking the ratio between the reference image and the test image. This process eliminates the effects of spatial non-uniformity in illumination light and coating thickness. However, difficulties also arise due to this necessity because the wind-on and wind-off images are acquired at two different times. Aerodynamic forces may cause model motion and deformation in high speed wind tunnel tests. The wind-on image may not align with the wind-off image. Thus, the ratio between the non-.aligned images can lead to considerable errors in calculating temperature or pressure using the calibration relations. An image
registration method is needed to solve this problem. The principle of image registration is based on a mathematical transform. Reference marks are placed on the model to accomplish the transform. The displacement of these marks signifies how the model is shifted and deformed. The error induced by the variation in illumination level between wind-on and wind-off image taking is another issue relating to the CCD camera method and is difficult to estimate.

2.3.2 Two-color luminophore system

1. A two-color TSP is made by combining a temperature sensitive luminophore with a temperature insensitive reference luminophore. Similarly, a two-color PSP consists of a pressure sensitive luminophore with a pressure- and temperature-insensitive reference luminophore. The probe luminophore and reference luminophore can be excited by the same illumination and ideally have no overlap between the emission spectra such that two color luminescent images can be completely separated by optical filters. The ratio between these two images can eliminate effects of spatial non-uniformities in illumination, paint thickness. Hence the need for a wind-off reference image is eliminated, which is the biggest advantage of this method over the above-mentioned CCD camera method.

2. A temperature sensitive luminophore can be combined with an oxygen sensitive luminophore. This dual luminophore temperature/pressure paint can be used for temperature correction in PSP measurements. This is exactly the type of PSP that has been employed in this work. The wind-off image is still needed since the reference probe is also temperature sensitive. But ideally, a multi-color PSP can be developed to simultaneously correct the effects of both temperature variation and non-uniformities in lighting and paint thickness.
Two-color PSPs have been developed by several research groups and some of them are already commercially available, but wind tunnel application has not been reported in public literatures so far.

2.3.3 Laser scanning system

A low power laser (<1 milliwatt) instead of a lamp is focused to a small point of the model and scanned over the surface of the model using computer-controlled mirrors. The luminescence is detected by a low noise photodetector (e.g., a Photo-Multiplier Tube). The signal is digitized with a high resolution A/D converter and processed to obtain temperature or pressure. The mirror is synchronized to the data acquisition system so that the position of the measurement point (the laser spot) on the model is accurately known.

Compared with CCD camera system, the laser scanning system has some advantages and disadvantages.

a. Before the analog output from the PMT is digitized, amplification and band limited filtering can be used to improve signal to noise ratio (SNR). The signal is then digitized with a high resolution (12 to 24 bit) A/D converter.

b. The system provides uniform illumination by scanning a single light spot.

c. It can be applied in a test rig where optical access is very limited

d. Computer-control of the laser spot allows the resolution and scan area to be user-specified, permitting non-regular shape, variable density grids. The region of measurement in CCD camera system is limited to rectangular arrays.
e. The time to obtain full-surface measurement is longer. The spatial resolution is lower.

2.3.4 Lifetime based detection system

\[ \frac{I_o}{I} = A + B \frac{p}{p_o} = \frac{\tau_o}{\tau} \]  

(2.32)

In above equation, A and B are properties of a particular paint system. \( \tau \) is the fluorescent lifetime of the paint, the time it takes for molecular emission to die out once the excitation source is removed, which is also pressure and temperature dependent. Lifetime is independent of the concentration of luminophore and illumination intensity. This leads to a constant lifetime across the model in flow-off conditions regardless of any non-uniformity of either paint application or lighting, eliminating the need for wind-off maps and ratioing techniques.

The continuous excitation light is replaced by a pulsed excitation light, such as a pulsed laser or a flash lamp. After an exciting pulse ceased, the exponential decay of the luminescence is measured by a photodetector and recorded with a PC or an oscilloscope. Lifetime \( \tau \) can be determined at different pressure and temperature, fitting the data with a simple exponential function. At present, lifetime is a routine test in a photochemistry lab. The configuration of this system could be similar to laser scanning system except a pulsed exciting light would be used. Both laser scanning and lifetime-based method involve very different experimental setup as well as data processing techniques than the CCD camera method.
2.4 Time response of TSP and PSP

In short-duration wind tunnel tests and unsteady flow measurement, fast time response of the luminescent paint is desired.

There are two characteristic time scales that related to the time response of the paint system. One is luminescent lifetime which represents an intrinsic physical limit for the achievable temporal resolution of the paint. Luminescent paints usually have a lifetime ranging from 10-10 seconds to milliseconds. Another is the time scale of the diffusion processes: thermal diffusion for a TSP layer and oxygen diffusion for a PSP layer. In general the diffusion time is much larger than the lifetimes of most luminescent paints. Therefore, the time response of a luminescent is mainly limited by the diffusion processes for both TSP and PSP measurements.

Liu [5] shows that, for the $19 \mu m$ Ru(bpy)-Shellac paint, the measured time constant is 16 ms for air jet impingement cooling and 1.4 ms for Freon jet impingement cooling, respectively. While liquid crystal sample shows a response time of about 245 $\mu$s. So the time constant of the paint is a function of local convective heat transfer. The author claimed that most conventional TSP or PSP with polymer binders do not have adequate time response for unsteady measurement.

Carroll [6] tested the time response of several PSP and found that, for PtOEP in GP-197 (a M&D PSP), the response times are 1.4 s, 1.6 s and 2.6 s for paint thickness of $22 \mu m$, $26 \mu m$ and $32 \mu m$. So the response time of PSP depends on polymer diffusivity, coating thickness and structure of the paints.
2.5 Accuracy of TSP and PSP measurement

The accuracy of the luminescent paint technique depends on the measurement system and the paint itself. The error sources can be grouped into three classes [3]:

1. Errors related to the chemical and physical properties of the paint, such as photodegradation, temperature and pressure hysteresis and uncompensated temperature effect of PSP.

2. Errors of the measurement system, such as variation of illumination intensity, photodetector noise and wavelength overlap between illumination source and optical filters.

3. Errors from displacement and deformation of the model due to aerodynamic loads, contamination of the painted surface during wind tunnel tests and luminescence scattering interference between neighboring surfaces of complicated model geometry.

Photodegradation occurs when the paint is exposed to the illumination light. The process depends on the illumination intensity and the exposure time. Although a normalized calibration relation will reduce the effect of paint photodegradation, a severely photodegraded paint will have low signal to noise ratio (SNR). So in experiment, the exposure time should be as short as possible.

The temperature and pressure hysteresis are related to the polymer structural transformation from a relatively brittle state to a soft and rubbery state when its temperature exceeds the glass transition temperature [5]. So once the paint is heated to a certain temperature its characteristics change. To address this issue, the paint should be preheated to a temperature above the glass temperature before it is used for quantitative measurement.
Scientific-grade CCD cameras are nearly ideal photodetectors. They have excellent quantum efficiency (photoelectrons collected per incident photon) of up to 85%, high SNR ratio, very linear response and up to 16 bits of resolution. The spatial resolution of CCD cameras (typically from $512 \times 512$ to $2048 \times 2048$ pixels) is usually more than enough for TSP/PSP applications.

The illumination source should provide uniform and stable illumination at the excitation wavelength of the paint. The illumination must be bright enough to produce a luminescence emission that can saturate the photodetector and thus take advantage of the detector’s SNR potential in a reasonable exposure time (several seconds). The blue LED based illumination source employed in this work has a claimed stability better than 0.1% after a 10 minute warm-up.

Registration of wind-on and wind-off images is a common method to address image spatial nonalignment induced by model motion/deformation. It is achieved through a mathematical transform so that a location on the model is the same in both the wind-on and –off images. The level of transform required is determined by the degree of model deformation. The registration procedure needs reference marks to be placed on the model and their locations are used to determine the transform coefficients.

The perspective effects due to camera angle and model surface curvature produce spatial distortions of the model image. Each point in the image domain should be related to a corresponding point on the model surface. Data mapping is required to solve the issue and basically it is a process of coordinate transform. There are lots of sophisticated mathematical methods available to deal with it. Bell and McLanchlan [7] have given detailed introduction of the process.
General statements about accuracy of TSP and PSP measurement are impossible because errors are dependent on the measurement system and the test configuration. Liu et al.[8] has given uncertainty estimate of ±0.3°C for a TSP application on a swept-wing model in supersonic flow. In another study, Liu et al [3] report calibration error of ±0.8°C and ±2°C for two different TSP formulas. The standard deviation of 0.2 psid was given for a PSP measurement over a range from 3 to 14.5 psia [3]. Morris et al [9] conducted a series of PSP calibration on a PSP sample. The minimum pressure resolution near atmospheric pressure (13 to 16 psia) is found to be ±0.05 psid for their system. But the uncertainty estimates do not consider the major error sources such as temperature effects and model displacement.

2.6 Other optical techniques for surface temperature measurement

Besides TSP technique, other optical measurement techniques that are applied most often for capturing the surface temperature map are introduced briefly in this section. They are thermographic phosphors (TPs), infrared thermography (IR), and liquid crystal (LC). Each of these techniques has been developed rapidly in the past one or two decades and has been applied in various areas of thermal fluid and heat transfer studies. All of them are non-intrusive method and has the capability of two-dimensional surface temperature mapping.

2.6.1 Thermographic Phosphors

The measurement technique of thermographic phosphors (TPs) is based on the fact that certain rare-earth doped phosphors exhibit line emissions, the amplitude and decay time of which are functions of temperature under UV stimulation. Such thermographic phosphors consist of an
activator or impurity ion, such as europium, doped into a host matrix, such as yttrium oxysulfide, at concentration up to a few percent. The essential physics of TPs can be explained as thermal quenching of the emitting dopant ion. Virtually all the TPs studied so far are ceramics that are rare-earth doped Group III metal oxides and oxysulfides [10]. By proper selection of dopant emission line and host matrix, the range of strong temperature dependence can be shifted to suit a particular application. Detailed physical principles of TPs can be found in the paper of Allison et al. [11].

There are two techniques to measure temperature with TPs. One of them uses the relation of temperature to the ratio of the intensities of two distinct emission lines. The other technique uses correlation of the characteristic decay time, or lifetime, of those emission lines to the temperature. The decay time technique has been successfully applied to provide point temperature measurements, but it is not amenable to two-dimensional surface imaging because the decay time of most phosphors is too rapid for the detector to obtain the images and determine the decay time. The technique of ratioing the fluorescence intensity of two distinct emission lines is recognized to provide 2-D imaging of surface temperature.

The main components of a typical TP thermal imaging system include pulsed laser and image-intensified charge coupled device (ICCD) camera. Pulsed excitation source is required to measure the integrated fluorescent emission over a narrow range during fluorescent decay. The image detector must be capable of low-light level image capturing at gating speeds as fast as a few microseconds.

For low temperature application phosphor coating can be applied using a binder such as epoxy or silicon resin. In high temperature environment such as an operating turbine engine, a
molecular bonding technique, such as sputtering, electron vapor deposition or plasma spray, is usually selected to apply the phosphor coating. Coating thickness on the order of 100 μm is necessary to obtain acceptable fluorescence uniformity. The thermal resistance of the coating in most applications is negligible. Measurement accuracy of $\pm 10^{-50}$ K was reported \[11\] over the range of 300 to 1500 K using the technique of ratioing the fluorescence intensity of two distinct emission lines of a rare-earth ion-doped phosphor. Chyr and Bizzak \[12\] applied europium-doped lanthanum oxysulfide (La2O2S: Eu+3) to their Laser-induced fluorescence (LIF) thermal imaging system, which exploits the temperature sensitivity of both the fluorescence intensity and lifetime of the phosphor emission lines. They reported accuracy of $\pm 0.5 \, ^\circ\text{C}$ over the range of 15–60 °C and spatial resolution of 1.0 mm.

2.6.2 Infrared thermography

The measurement technique of infrared thermography is based on the principle that the radiation received from an object is a function of its temperature and spectral emissivity. The governing equation of the technique is the modified version of the Stefan-Boltzmann Law for real objects:

$$ W = \varepsilon \sigma T^4 $$

The radiant energy $W$ is related to the $4^{th}$ power of absolute temperature $T$ through $\sigma$, the Stefan-Boltzmann constant, and $\varepsilon$, the emissivity of the object. Emissivity of $\varepsilon=1$ defines the theoretical black body, and a value of $\varepsilon=0$ defines the perfect reflector. For wind tunnel models,
ε is a function of model material, surface finish, angle at which the model is viewed, the
temperature of the model and the wavelength of radiation being detected.

Most of the IR imaging systems consists of a single IR radiation detector, an optical
system to focus the radiation from a distant spot onto the detector, and a scanning mirror to scan
the measurement spot over a surface. The detector is usually cooled by liquid nitrogen or a
Peltier cooler to minimize detector noise. The output of an IR camera is essentially the same as a
video camera, and can be recorded with a VCR and digitized with standard frame-grabber
computer boards. Commercial IR detectors are sensitive to IR radiation either in the 3 to 6
micrometer (short wave) or in the 8 to 12 micrometer (long wave) bands. Shorter wavelength
detectors are better suited for high temperature applications and vice versa. The field of view of
the camera is limited by the objective lens and usually ranges from 2.5° to 40°.

Infrared thermography provides a non-intrusive measurement of two-dimensional
temperature distribution with high resolution. The area of a scene covered by a thermal imager
will depend on its field of view (FOV) and its distance from the object. The FOV can be altered
by changing the focal length of the lens. The typical response time of IR detectors is on the order
of 160 nanoseconds [13]. With a suitable optical system, an infrared imaging camera can
measure temperature distributions on a microscopic scale. By altering the aperture diameter of
the lens and/or by introducing or removing filters in the optical path, the response of a thermal
imaging camera to thermal radiation, and hence the temperature, can be varied over a large
range, typically from –20 to 1600 °C.

One obstacle to apply the IR method is that radiation from an object is a function of not
only the temperature of the object but also of its emissivity. Different material and surface
finishes have different emissivities, so that the relative brightness of different objects in a thermal image are not necessary a real indication of their relative temperatures. A further complication results from the reflectance. If the emissivity of a surface is less than unity, the object not only emits radiation but also reflects radiation from surrounding objects, which yields a main source of error of temperature measurement. In aeronautical research, models are usually made of aluminum or stainless steel of high quality surface finish. This type of model has high thermal diffusivity, low emissivity and high reflectance. High thermal diffusivity levels out temperature difference along and in-depth through the model. The high reflectance and low emissivity of the surface decrease the signal-to-noise ratio (SNR) on the thermograms. To make quantitative temperature and heat transfer measurements, the target must have an emissivity of approximately 0.80 or higher. An insulating film or compatible anti-reflective coating is usually applied, the thickness of which is on the order of a couple of microns.

For most of the wind tunnel applications, the IR system is usually arranged outside the test section. The optical-access window is another main problem to be solved for IR measurement. Neither standard glass nor quartz can be used due to the specific wavelength for which the IR detector is sensitive. Germanium’s transmission range is from 2 to 15 μm and is selected most often for IR application. Anti-reflection coating should be applied on both side of Ge window since it has high surface reflectivity.

IR method has been widely applied in aerodynamic researches as well as heat transfer studies. Lafferty and Collier [13] did a surface temperature measurement on a thin triangular wing model at Mach 14 using a commercial IR camera. The IR data was compared with the vapor-deposited thermocouple data and all the IR data was within ±5% of the thermocouple
measurements. Kang et al. [14] studied endwall heat transfer of a gas turbine stator vane using IR method. The endwall was made of thin copper layer atop a serpentine-patterned Inconel heating element. This makes the endwall a constant heat flux plate. The IR camera was calibrated in-situ using a thermocouple placed on the heater surface. The uncertainty of temperature measurement was 5 °C and the corresponding uncertainty in heat transfer coefficient was ±3.5%. Carlomagno et al [15] performed a heat transfer investigation of a single jet. A cold air jet impinged on a heated metallic foil while an IR camera captured image of the thin foil from the backside of the impingement. Figure 2.2 shows the color thermogram of the temperature map. The Nusselt number distribution for the single jet was calculated from the corresponding temperature map.

2.6.3 Liquid Crystal Thermography

Liquid crystals have been applied extensively as surface temperature sensors in fluid mechanics and heat transfer studies in the past two decades. Certain organic compounds exhibit behavior between that of an isotropic liquid and a nonisotropic crystalline solid. These compounds are called liquid crystals or mesophases. In this mesophase the molecules are moveable but still ordered. There are three groups of liquid crystals: the smectic; the sematic; and the cholesteric. A cholesteric liquid crystal compound reacts to changes in temperature by continuously changing color over an active range. The helix-shaped liquid crystal molecules scatter incident light if the light has a wavelength equal to the pitch length of the helix. Light of other wavelengths is transmitted through the liquid crystal layer. The molecules stretch and contract in response to a number of stimuli, including temperature, shear stress, and electromagnetic fields. When the molecules stretch or contract, the pitch length of the helices
changes, shifting the wavelength of light scattered. So a surface coated with liquid crystal will change color as its temperature changes or when the surface shear stress varies. Encapsulating the liquid crystal molecules in a polymer will eliminate shear stress sensitivity while retaining the material’s temperature dependence. As the temperature of a thin layer of a liquid crystal increases beyond the melting temperature of the solid, the layer changes color of the reflected light from red to orange-yellow-green until blue. At higher temperature, the layer is fluid and becomes transparent as in the solid state. Figure 2.3 shows a photograph of the response of liquid crystal color spectrum along a heating surface with linear temperature distribution between the two selected end temperatures. The active temperature range (or event temperature range) of the liquid crystals can be tailored by proper mixing of various types of liquid crystal compounds and this temperature range may be from 0.5 °C to 30 °C. The advantages of liquid crystal include easy handling, low cost, nearly instant measurement and good accuracy and resolution.

The background beneath the liquid crystal layer must be completely black; otherwise the color change will not be visible. Water-based black paint is usually selected since it is less likely to interfere with liquid crystals. Liquid crystal layer can be applied to the measurement surface with a paint sprayer or a brush. Dry layer thickness of 30-50 μm is required to produce good colors [15].

Liquid crystal imaging is particularly useful for producing a qualitative picture of two-dimensional thermal fields with both spatial and temporal temperature variation. For quantitative measurement it requires a measurable scalar to quantify ‘color’, which associates with temperature, of liquid crystal coated surface at every pixel in a thermal image. Hue angle, h, is chosen most often to represent color. It is defined in a polar chromaticity space determined by
the intensities of the red, green, and blue primaries. The video system separates the light striking each pixel into its red, green, and blue components. Hay et al. [16] proposed a hue definition using UVW primary system:

$$h \equiv \arctan\left(\frac{\sqrt{3}(G - B)}{2R - G - B}\right)$$

(2.34)

The color matching is achieved through a linear transformation between the RGB system and the UVW system.

The effect of viewing/illumination angle has been a major obstacle for LC application. Calibrations become invalid by changes in lighting angle and viewing angle. A typical LC calibration result in figure 2.4 shows how viewing angle changes the calibration curve. Hay et al. [16] suggested using a dimensionless calibration correlation. The data for different lighting conditions collapse to the same calibration curve after applying the proposed correlation. The maximum difference in lighting angle was 37°. The averaged uncertainty was 7-9% of the active range. But it was also observed that when lighting angle becomes too big, the error produced by this calibration method would be too large to be useful, which suggests that if severely off-axis lighting is anticipated in experiment, the lighting angle should be used in the initial calibration.

Liquid crystal permits the visualization of lines of constant temperature (isotherms) and hence has been used as a visualization tool in heat transfer studies. The heat transfer coefficient is defined as the surface heat flux divided by the difference between the wall temperature and the reference temperature. If liquid crystals are applied to a surface with uniform heat flux, then an isotherm can be translated into lines of constant heat transfer coefficient. Varying the magnitude of the heat flux on the surface and observing only a single isotherm results in a series of curves,
which represents contours of constant heat transfer coefficient. Goldstein et al [17] (see figure 3.10) applied liquid crystal in a jet impingement study using the above visualization technique. Figure 3.10 shows lines of constant Nusselt number for impinging heat transfer from arrays of 7 jets. Multiple event narrowband liquid crystal paints can be developed to cover a wider temperature range of measurement.

Figure 2.2 Color thermogram of the temperature map of a heated plate cooled by an air jet. Image captured by IR camera [15]

Figure 2.3 Response of TLC color spectrum along a heating surface
2.6.4 Summary of optical techniques for surface temperature measurement

Table 2.1 compares the four measurement techniques in details. In general liquid crystals (LC) thermography and temperature sensitive paint (TSP) techniques take the advantages of easy handling, low instrumentation cost, and good accuracy. Lighting and viewing angle issue need to be addressed for LC application when quantitative measurement is desired. The most attractive advantage of TSP technique is that essentially the same instruments can be employed for surface pressure measurement using pressure sensitive paint (PSP). Although the temperature sensitive range is lower than 100 C, it is sufficient for most of the wind tunnel application.
Table 2.1. Comparison of surface temperature measurement techniques

<table>
<thead>
<tr>
<th>Method</th>
<th>Temperature sensitive range (°C)</th>
<th>Time response</th>
<th>Temperature resolution (°C)</th>
<th>Spatial resolution</th>
<th>Cost e ($1000)</th>
<th>Accuracy</th>
<th>Complexity c</th>
</tr>
</thead>
<tbody>
<tr>
<td>TPs</td>
<td>200~1200 a</td>
<td>10~3000 μs a</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>0.3~0.4</td>
<td>1 mm</td>
<td>&gt;50</td>
<td>&gt;±0.5 °C</td>
<td>H</td>
</tr>
<tr>
<td>IR</td>
<td>20~1600</td>
<td>&lt;10 ms b</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>&gt; 0.02</td>
<td>&gt;65K pixels</td>
<td>&gt;30</td>
<td>&gt;±2%</td>
<td>M</td>
</tr>
<tr>
<td>LC</td>
<td>20~100 d</td>
<td>20~50 ms</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>&gt; 0.25</td>
<td>Depends on optical system</td>
<td>&lt;5</td>
<td>&gt; ±0.1°C</td>
<td>H</td>
</tr>
<tr>
<td>TSP</td>
<td>170~105</td>
<td>Varies with experiment setup</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>0.1</td>
<td>Depends on optical system</td>
<td>20</td>
<td>&gt; ±0.3°C</td>
<td>L</td>
</tr>
</tbody>
</table>

a depends on formula of TPs  
b depends on IR imager  
c complexity based on setup, operation, and data reduction: H=high, M=medium, L=low.  
d applicable to encapsulated liquid crystals (Moffat et al. 1989)  
e estimated cost of major equipments
CHAPTER 3 LITERATURE SURVEY OF JET IMPINGEMENT

3.1 Overview

Jet impingement has the most significant potential to increase the local heat transfer among all heat transfer enhancement techniques. With easy implementation, enhanced heat transfer rates are obtained when a jet flow is directed from a nozzle of a given configuration to a target surface. Since relatively high local heat transfer coefficients are obtainable compared to no impinging flows, the use of the jet impingement technique provides the designer with a means for more effective control over the temperature of the surface under consideration. Analysis is not yet able to provide reliable predictions for impingement heat transfer coefficients in the practical range of jet Reynolds number $Re$, so that experimentally based information is continuously needed.

The flow structure of jet impingement can be summarized into three characteristic regions, as shown in figure 3(a): the free jet region formed as jet exits with a velocity distribution $U$, the stagnation flow region formed upon jet impact and deflection, and the wall jet region formed upon re-acceleration of the flow along the target surface. Either circular or slot nozzle are usually employed, the slot jet provides a larger impingement zone, while the circular nozzle insures a more localized high transfer rate.

It has been accepted that, in advanced aircraft turbines, with high gas temperature and high pressure, simple convective cooling is not adequate to cool the turbine material to an acceptable level if not combined with other forms of cooling. Impinging cooling is often used to cool the turbine components like turbine airfoils and endwalls.
3.2 Hydrodynamics of impinging flow

![Diagram](image)

Figure 3.1. Comparison of (a) flow regions in an impinging jet with (b) flow regions of a free jet

The flow field of impinging jets from single round and slot nozzles can be divided into three regions as shown in figure 3.1 (a): the free jet region, the stagnation flow region, and the wall jet region. The velocity field of an impinging jet is also shown in the figure. The critical jet Reynolds number $Re_c$ (based on nozzle diameter and nozzle exit velocity), which distinguishes laminar jets from turbulent jets, is about 3000. In gas turbine component cooling, the jet nozzle diameters are quite small, but the Reynolds number can be quite large owing to the high velocity of the jet. In such applications, the jet developing from the nozzle is generally turbulent, and the turbulence intensity at the core can be as high as 25% [19]. The high velocity coupled with high
velocity fluctuation, increase turbulence mixing, and hence significantly increase the heat transfer capability.

In gas turbine application, arrays of jet are used rather than a single jet. For arrays of impinging jets, the wall jets of adjacent nozzles impinge upon each other and thus create secondary stagnation zones. Boundary separation and flow eddying are the characters of these secondary stagnation zones and lead to a considerable reaction on the other flow region.

3.2.1 Free jet region

In the free jet region, the jet broadens linearly with its length up to the top of the stagnation region. According to Schlichting [20], by a selection of the origin in the coordinate system, the width b (see figure 3.1 (a)) of the free jet is directly proportional to the distance along the jet centerline.

\[ b = \text{constant} \ z \]  \hspace{1cm} (3.1)

The jet retains the nozzle exit velocity in its potential core. The length of potential core is usually defined as the distance from the nozzle exit where the pressure head on the axis falls to 95% of its maximum value at nozzle exit. According to Abramovich [21], the pressure in this region is virtually constant and equal to the pressure in the surrounding fluid. For all cross sections normal to the jet, the momentum flux should be the same:

\[ J = \rho \int U^2 dA = \text{Constant} \]  \hspace{1cm} (3.2)

The core is eventually dissipated as a result of the mixing action. The velocity profiles at different axial location of the jet fall into one curve after the velocity is normalized to the centerline velocity.
The extent of the potential core is reported from 4 to 7.7 by various researchers [22, 23]. Unfortunately, the amount of divergence of core lengths reported by different investigators generally exceeds the experimental error. It might be explained by the effect of the scale and the intensity of turbulence within the jet. The velocity profile at the nozzle exit also seems to be a factor [24].

Beyond the potential core, the centerline velocity $U_m$ decays with $z^{-1}$ for the circular jet and with $z^{-1/2}$ for a slot jet.

As to turbulence distribution of axisymmetric jet, the turbulence generated by mixing is much more intense than that usually encountered in pipe flow. The axial and radial distribution of velocity and turbulence in an axisymmetric jet is shown in figure 3.2. The intensity of axial turbulence $u'/U_m$—the relative magnitude of the R.M.S. fluctuations of the axial velocity ($u'$), referred to the local value of the time average axial velocity ($U_m$)—increases continuously with jet length. According to Gardon and Akfirat [25] the absolute magnitude of the velocity fluctuations reaches a maximum in the neighborhood of $z/D=8$. Concerning the radial distribution of turbulence, initially the turbulence is greatest in the center of the mixing region, i.e. at $r=\pm 1/2D$. Then turbulence starts to penetrate into the potential core, the peak in the radial distribution of turbulence does not reach the centerline of the jet until some distance downstream.
3.2.2 Stagnation region

The height of stagnation flow region is about 1.2 nozzle diameter, according to Martin [23]. In this region the vertical velocity component is decelerated and transformed into an accelerated horizontal one, which increases linearly from 0 to a maximum value at the edge of
stagnation flow. This is due to the exchange of momentum with the surrounding fluid so that the accelerated stagnation flow finally must transform to a decelerated wall jet flow. At the stagnation point, the velocity is zero and the pressure is a maximum. Experimental data obtained for a circular jet shows that the pressure fell from a maximum value $P_o$ at the stagnation point to $P_\infty$ (ambient) at a radius of 1.6 to 3 nozzle diameters, which gives the boundary of the stagnation region [24]

3.2.3 Wall jet region

In wall jet region velocity reduces to zero exponentially. Unlike a viscous boundary layer, the velocity profile has a point of inflection in wall jet region. Glauert [26] divided the flow region into two parts: an inner layer where the effect of the wall is present, and an outer layer which is characterized by the features of a free turbulent flow. The velocity at the boundary between these two layers is the maximum velocity. This maximum velocity may be expressed by the relation

$$V_m = \frac{\text{const}}{r^n}$$

(3.3)

value of $n$ has been reported between 1.1 and 1.2 [24]. The spread of the wall jet in terms of the half value $Z_{1/2}$ ($V=V_m/2$ at $Z=Z_{1/2}$) can be predicted by

$$\frac{Z_{1/2}}{D} = \text{const} \left( \frac{r}{D} \right)^m$$

(3.4)

the value of the exponent $m$ was found to be close to 1 [24].

In the stagnation zone, the stabilizing effect of acceleration keeps the boundary layer laminar. Generally the transition to turbulence happens right after the decelerating starts.
3.3 Heat transfer of a single round jet impinging on a flat plate

3.3.1. Definition

The local heat transfer coefficient is generally defined as:

\[ h = \frac{q_w}{T_w - T_{ref}} \]  

(3.5)

where \( q_w \) is the wall heat flux, \( T_w \) is the wall temperature, and \( T_{ref} \) is a reference temperature-usually either the jet total temperature \( T_j \) or the adiabatic wall temperature \( T_{aw} \).

The Nusselt number is defined by

\[ Nu = \frac{hD}{k} \]  

(3.6)

where \( D \) is the nozzle diameter, and \( k \) is the thermal conductivity of the fluid. The local Nusselt number distribution can be averaged to obtained a mean Nusselt number

\[ \overline{Nu} = \frac{\int A NudA}{A} \]  

(3.7)

for an axisymmetric jet, the mean Nusselt number can be written as

\[ \overline{Nu} = \frac{2}{R^2} \int_0^R Nu(r)rdr \]  

(3.8)

One objective of research on an impinging jet is to find a reasonable empirical correlation for the local, stagnation point and averaged Nusselt number. There are many different factors that affect heat transfer between a solid plate and a turbulent jet. The factors include turbulence, exit jet velocity profile, nozzle (orifice) geometry, nozzle-to-surface distance, surface curvature, and etc. According to Eckert [27], in the absence of free stream turbulence, the non-dimensional
heat transfer coefficient $\text{Nu}$ at stagnation point on a circular cylinder or sphere immersed in uniform crossflow can be expressed as $\text{Nu}/(\text{Re}^{1/2}\text{Pr}^{1/3})=\text{constant}$. Near the stagnation region of the impinging jet, laminar flow predominates in the boundary layer even for turbulent jets. Hence, the power law relationship is most often used as an empirical correlation for the averaged Nusselt number near the stagnation point of impingement

$$\text{Nu} = C \text{Re}_j^a \text{Pr}^b \quad (3.9)$$

where $a=1/2$ and $b=1/3$, the value of $C$ varies from 0.88 to 1.09 [22], $\text{Re}$ is the Reynolds number based on jet velocity and nozzle diameter, and $\text{Pr}$ is the Prandtl number of the fluid.

3.3.2 Geometric effects

The local heat transfer coefficient depends on several factors, and the variation is complex. In general, $\text{Nu}$ (or $h$) in the stagnation point is higher compared to a location far from stagnation point. However, a second peak can occur depending on jet Reynolds number $\text{Re}$ and dimensionless nozzle-to-surface distance $Z/D$. If $Z/D$ is small enough, second peak dominates the heat transfer pattern.

Baughn and Shimizu [28] applied liquid crystal technique to measure heat transfer from a surface with constant heat flux and a turbulent impinging jet. A uniform heat flux is established by electrically heating a very thin gold coating on a plastic substrate. An isotherm on the surface represents a contour of constant heat transfer coefficient and is a line of a particular color of liquid crystal. This technique is described in detail by Moffat [15]. The liquid crystal employed had a narrow range of 1 degree C and temperature resolution of the green color was about 0.1 degree C. The position of the green line is shifted by changing the electrical heating of the gold
coating and thus the surface heat flux. A complete mapping of the heat transfer coefficient over the entire surface can be achieved by this technique. A pipe of 72 diameters long was used as the jet nozzle and the flow at the jet nozzle exit is fully developed. The turbulence level at the center of the jet at the exit was 4.1%. Figure 3.3 shows the Nusselt number along the surface for \( Re = 23750 \). When the separation distance \( Z/D \) is small (\( Z/D = 2 \) in this case), the heat transfer has a second peak other than the stagnation point maximum at approximately \( r/D = 1.8 \). The effect of separation distance \( Z/D \) on the stagnation point heat transfer is shown in figure 3.4. The maximum heat transfer occurs at a \( Z/D \) of approximately 6 to 7.

Gardon and Cobonpue’s investigation [29] on the heat transfer between a flat plate and impinging air jets has been cited most often in the literature. In their study the heat transfer was obtained between an isothermal hot surface and jets of cooling air. The average heat transfer rate was determined from the total electrical power input, while the local heat transfer rates were measured with a heat flux transducer. The single nozzle employed was a tube (unspecified length to diameter ratio) with well rounded inlets and short parallel throats. Nozzle diameters ranged from 0.089 to 0.354 in. and nozzle exit velocities from a few ft/sec to sonic velocity. Figure 3.5 shows a correlation in terms of the stagnation point Nusselt number \( \Nu_o \), the jet Reynolds number \( Re \), and the dimensionless nozzle-to-plate distance \( Z/D \):

\[
\Nu'_o = 13(Re_j)^{0.5} D / Z
\]

The correlation is good for turbulent jets in the region of fully developed jet flow (\( Zn/D > 20 \)). Below that range, a “residual effect” of the nozzle diameter was left unexplained. For values of \( Zn/D < 12 \), the larger nozzles produces progressively higher peak heat transfer rates at lower dimensionless nozzle-to-plate distance.
Martin [23] emphasized the studies with heat and mass transfer related to heating and drying processes. The local mass transfer and heat transfer measurement results of various studies were combined in one figure (Figure 3.6) based on the heat and mass transfer analogy. Empirical equation was developed for area averaged heat or mass transfer rate (as defined in equation 3.7).

\[
\left(\frac{Sh}{Sc^{0.42}}\right) = \left(\frac{Nu}{Pr^{0.42}}\right) = \frac{D}{r} \frac{1 - 1.1D/r}{1 + 0.1(Z/D - 6)D/r} F(Re) \quad (3.11)
\]

Where \( F(Re) = 2 Re^{0.5} \left(1 + \frac{Re^{0.55}}{200}\right)^{0.5} \)

Range of validity: \(2000 \leq Re \leq 400,000; 2.5 \leq r/D \leq 7.5; 2 \leq Z/D \leq 12\)

Figure 3.3 local heat transfer distribution along the target plate, Re=23,750 [28]
Figure 3.4 Effect of Z/D on the stagnation point heat transfer, Re =23,750 [28]

Figure 3.5 Effect of z/D on heat transfer coefficient at stagnation point [29]
Figure 3.6 Local Sherwood numbers for impinging flow from single round nozzles [23];
Sc=0.59

Figure 3.7 Radial distribution of Nu/Re^{0.76} for Z/D=2, 5, 8, 10. [30]
Goldstein et al [30] studied local heat transfer to an axisymmetric impinging air jet. A “quarter-ellipse” jet nozzle was connected to a long delivery tube. The stainless-steel heating foils (0.0254mm thick) was bonded on the textolite\(^1\) target surface, and 63 36-guage thermocouples are embedded in the test plate to obtain the surface temperature data. A correlation of average heat transfer coefficient (defined by equation 3.9) was developed based on the results with Z/D from 2 to 12 and Re from 61,000 to 124,000. Nu is basically a function of Re and Z/D. The integral of equation (3.8) was determined for R/D=0.5, 1, 2, 3, 6, 18 and 32 for Z/D≥6.

\[
\frac{Nu(R,Z)}{Re^{0.76}} = \frac{A-|Z/D-7.75|}{B+C(R/D)^{1.285}} \text{ for constant surface heat flux} \tag{3.12}
\]

The correlations are based on the results shown in figure 3.7. A, B and C are constants. Note that Z/D has a linear effect on Nu and there exists an optimum Z/D of 7.75 for maximum Nu. The distribution of the recovery factor was also investigated. The recovery factor is found to be independent of Re but dependent on Z/D. The optimum Z/D seems to coincide with the length of the potential core. Beyond the potential core, the heat transfer coefficient falls.

There has been considerable disagreement among investigators as to the physical explanation for the second peak in local heat transfer for small nozzle-to-plate separation distance Z/D. Popiel and Trass [31] studied regular vortex structure of free and impinging round jets by a smoke-wire visualization technique. They observed ring-shaped wall eddies induced consecutively by the large scale torodial vortices hitting the impingement surface. It is suggested that these wall eddies could be responsible for the additional enhancement of local heat transfer.

\(^1\) A non-flexible polymer composite
and possibly for the second peak in the local Nusselt number. In their study of 2-D jets, Gardon and Akfirat [25] attributed the second peak in heat transfer to the transition from laminar to turbulent boundary layers. The transition is triggered by the disappearance of negative pressure gradient which exist in the vicinity of the stagnation point and serve to stabilize the laminar boundary layer, in spite of locally already high turbulence in the free stream. The static pressure distribution on the impacted surface of a 2-D jet is shown in figure 3.8.

Figure 3.8 Distribution of static pressure over the impacted plate [25]

\[ \Delta P \]—static pressure on plate, \( \Delta P_0 \)—static pressure at the stagnation point, B—width of slot jet nozzle
3.3.3 Turbulence levels

Reference [25, 32] have attributed secondary heat transfer peaks observed at small Z/D for a single jet to transition from laminar to turbulent flow. Amano and Brandt [33] disputed and attributed the phenomenon to kinetic energy effects where mixing length is less than nozzle to plate spacing. The numerical study they performed shows that the secondary peak becomes larger and moves outwards as Re increases. Martin [23] claimed that turbulence level at the nozzle exit can greatly influence heat and mass transfer in the stagnation region. The absolute jet turbulence may be a function of nozzle design.

3.3.4 Surface curvature

Hrycak [34] studied heat transfer from a row of impinging jets to a concave cylindrical surface. The result shows that for a highly curved concave surface (like a turbine blade leading edge), the optimum Z/D is less than that for an equivalent flat plate case. When the nozzle to surface distance is short, heat transfer in the stagnation region is adversely affected by curvature, but total heat transfer is increased relative to a flat plate owing to the thinner boundary layer in the wall jet region. When the separation distance becomes larger, the curvature effect diminishes and the heat transfer characteristics approaches those of a flat surface.

3.3.5 Surface roughness

Bunker and Bailey [35] studied the impingement surface roughness effect with a single tube jet in a model of turbine blade platform cooling configuration. The elevated roughness was
achieved by applying nickel particles and silver epoxy over the entire impingement surface in a random mode. The rough surface of the model has an RA value of about 30 microns. With $\text{Re} = 130,000$ and $Z/D=2.5$, the heat transfer coefficient are increased as much as 50% compared to the case of smooth target surface. Roughness shows its largest effect in the immediate impingement zone. However, this roughness benefit is lost if the impingement cooling jet is not relatively close to the surface.
CHAPTER 4 TSP AND PSP CALIBRATIONS AND INVESTIGATION OF TSP DEGRADATION

Paint calibration involves independent pressure and temperature measurements to establish the relationship between image intensity ratio, pressure and temperature. This can be done either in situ, using data from pressure taps or thermocouples in the model, or a priori, whereby the intensity of a painted coupon in a calibration chamber is measured over an appropriate range of pressure and temperatures. A priori calibration method has been applied in this study.

4.1 Data acquisition equipment and TSP/PSP sensor

4.1.1 CCD camera and software

The choice and quality of the scientific-grade camera are dictated by the measurement accuracy desired. To ensure the lowest possible noise and highest sensitivity to paint emission wavelengths, a 12-bit (or higher A/D resolution) camera is needed. The camera must be thermoelectrically cooled and have a high quantum efficiency at the paint emission wavelengths.

A high resolution 14-bit CCD camera was utilized in this work. It is a PCO-1600 CCD camera provided by Cooke Corporation with spatial resolution of 1200 by 1600 pixel. The image data is transferred via IEEE 1394 ("firewire") cable and firewire PCI card to the PC. “CamWare” software provided by Cooke Corp. is used in the Windows operating system to control initialization, exposure time and image acquisition. The acquired image data are processed using MATLAB.
4.1.2 Light source

LED-based illumination source (peak wavelength at 464 nm) was selected as the excitation light for both TSP and PSP. The stability of the source provided by ISSI is about 1% after 10 minute warm up. The excitation spectrum of LED is shown in figure 4.1.

4.1.3 TSP and PSP sensor

Uni-Coat TSP formulated by ISSI is used in this work. The effective temperature range is 0-100 ºC, beyond which the temperature sensitivity of TSP becomes weaker. It is contained in aerosol can and can be applied conveniently. After heat treated above 100 ºC for 30 minutes the temperature sensitivity of the paint is about 0.9%/ºC. The TSP painted surface is smooth and non-sticky. The emission spectrum of TSP is shown in figure 4.2.

Temperature-corrected PSP is the trend of PSP application. A dual-probe (Bi-Luminophore) paint obtained from ISSI Company is selected due to its low temperature sensitivity and the temperature compensation capability. It is also called Binary-FIB PSP in the product list of ISSI since the basecoat is a separate product named Uni-FIB PSP. This paint consists of a pressure probe and a temperature/reference probe. The temperature probe has a temperature sensitivity very close to that of the pressure probe at a certain temperature range. This allows taking ratio of the emission intensity from each probe to determine the pressure and the temperature sensitivity is eliminated. The spectrums of PSP emission is shown in figure 4.3. To separate the illumination lights of the two probe luminescence, either two cameras or a proper filter changing system is needed. In this study, a filter changing system based on an automated filter wheel has been incorporated into the CCD camera system. The Bi-Luminophore paint
requires four measurements as opposed to two for the standard PSP. The wind-off (reference) images of both the pressure and the temperature probe, $I_{pr}$ and $I_{tr}$, are taken first, followed by the wind-on images for each probe $I_p$ and $I_t$. To determine pressure, the ratio of the wind-off images ($R_o$) is divided by the ratio of the wind-on images ($R$). This process results in a pressure field that is independent of temperature. The correlation between the luminescence intensity and pressure for this dual-probe paint can be written as:

$$
\frac{R_o}{R} = \sum_{n=0}^{N} A_n \left( \frac{P}{P_{ref}} \right)^n
$$

(4.1)

where $R_o=I_{pr}/I_{tr}$; $R=I_p/I_t$. In this case the coefficients $A_n$ of polynomial are no longer functions of temperature.

The effective pressure range of the Bi-Luminophore PSP is 0.5 to 30 psia. The range of temperature compensation is 5-50 °C.

Figure 4.1 Spectra of LED illumination [40](Qiu 2001)
4.1.4 Filters and the filter changing system

Proper optical filters are selected to separate the excitation light and the emission light from the paint.

- TSP: 590nm long pass filter
PSP: 645FG07-50 (Andover) for pressure channel; 530FG05-50 (Andover) and 03 FIV 044 (Melles Griot) for reference channel

In order to capture the luminescent intensity of both pressure and reference channels of the Binary-FIB PSP, a filter changing system (Figure 4.4) was selected and installed in front of the CCD camera lens (Figure 4.5).

The wheel is 6 inches in diameter and is rotated on the center axis by a small stepper motor. The wheel can hold up to five 50mm filters, which are mounted in place using an RTV silicone sealant. The shaft of the stepper motor makes friction contact with an O-ring that is secured to the circumference of the wheel. Small magnets are inserted into each wheel opposite the filters, which indicate to the electronics the current wheel ID and filter position. Hall effect sensors detect the magnets and send a logic control signal to the hand control micro controller. The numbers 1, 2, 3, 4, and 5 mark the filter position magnets on the wheel and correspond to the filter on the opposite side. (See Figure 4.4 a). The wheel is held in place by a center shaft which is spring loaded and can be pulled from the wheel by use of a small pull knob on the back of the wheel housing. Rotating the pull knob counter clockwise will allow the spring shaft to retract from the wheel. At that point, the door can be opened and the wheel removed by hand.
(a) filter changing system with control box

(b) Cross-section view of the filter wheel and housing

Figure 4.4 Filter changing system (drawing from IFW Inc.)
4.1.5 Calibration set up

The PSP/TSP calibration system is shown in figure 4.6. A 1/32\textsuperscript{nd} inch thick aluminum coupon coated with TSP or PSP is placed on top of a thermal electrical module. The temperature of the coupon is controlled by adjusting the power supply output to the thermal electrical module. A fine gauge type-E thermocouple is embedded under the coupon. The temperature difference between the top and the bottom surface of the thin aluminum coupon is less than 0.04°C. Pressure inside the chamber is manually controlled and monitored with the Scanivalve pressure transducer.
4.2 Painting application

The steps of painting application are vital to maintaining coating quality, which affects the measurement accuracy directly.

Safety precautions were followed to limit inhalation and skin contact to the paint. All painting was conducted either outdoor or in a painting room with good ventilation. Rubber medical gloves were worn during both the paint application and paint removal process to prevent
skin contact with various solvents and paint ingredients. It also helps to avoid the inadvertent transfer of skin oils onto the model surface. A 3M half mask respirator with a 3M model 6001 cartridge for volatile vapors was always worn.

Any previously applied paint was first removed from the model surface as the first step of paint application. The FIB based PSP can be removed with Acetone and the Uni-Coat TSP can be removed with either Acetone or Alcohol.

Model regions that are not desired to be painted were covered with 3M masking tape. Ideally, any surfaces which are in view of painted model surface should first be painted black, e.g., the bottom of the calibration chamber which is seen as background in the calibration image.

Application of the Binary-FIB PSP (both base coat and top coat) was performed with a standard commercial spray air gun. The paint was shaken by hand for one minute after poured into the container of the air gun. The selection of spraying air pressure, spraying distance, lateral moving speed, and spraying frequency are all critical to provide a uniform coating. Air pressure was set between 20 to 25 psi. The “fluid control” of the air gun was adjusted to a low setting and the “pattern control” was adjusted to full open. It is recommended to apply a few coating on a trial surface before painting the test model. The air gun was kept one foot away from the surface and paint was applied in a lateral sweeping motion (about 2ft/sec). Usually 8 to 10 cross coats (one cross coat means one sweep motion from left to right and then from right back to left) are required to obtain the paint thickness which will give enough emission signal. Three to five minutes waiting is required between coats to allow solvent to evaporate. Painting in a slower motion or not enough waiting time between coats would result in a wet coat, which appears like a skin-rash. A wet coat will cause imbalanced interfacial surface tension within paint molecules.
In order to minimize the free energy within the paint layer, groupings of paint molecules will take a spherical form to minimize the surface area per liquid volume. Therefore “pink dots” in skin-rash pattern show on the wet-painted surface. The model with a wet-painted surface will have high measurement scatter and hence adversely affect the accuracy. By applying the paint very lightly, the solvent evaporates before the paint can form undesirably large spheres. Heat treatment was conducted in the basecoat curing process in order to address the temperature and pressure hysteresis. The painted model was kept in the oven at 65 C for 20 minutes. The top coat of PSP is for temperature compensation purpose. For best results, 5 to 6 cross coats was applied over the cured basecoat. No heat treatment is needed. The painted model can be tested two to three hours after the painting process is finished.

The Uni-Coat TSP in aerosol can is much more convenient to apply. Although there is no pattern control or pressure setup to be adjusted, the spray pattern and painting flow rate is determined by the nozzle tip of the aerosol can. Therefore the nozzle should be protected carefully and be cleaned with alcohol after every painting to prevent clogging of the nozzle. The aerosol can should be shaken vigorously for 10 minutes every time before painting starts. This shaking time should be extended for a new TSP can. The application process is very similar to that of PSP, although the spraying frequency was about half that utilized for the PSP. The oven temperature was set at 100 degree C for TSP heat treatment.

The painted surface can be buffed using fine grit sanding sponge to improve the surface finish. The same method can be used to clean a slightly contaminated paint surface. Severely contaminated surface or highly degraded paint needs to be cleaned for re-painting.
4.3 Calibration result and uncertainty

4.3.1 TSP calibration result

UniCoat TSP is used in the experiment and calibration. The typical temperature dependencies of the paint are shown in figure 4.7. Normalized temperature theta is defined as theta=(T-Tr)/100 and is plot against intensity ratio I/Ir. The number 100 in the denominator of theta refers to the effective temperature range of TSP. The calibrations have been performed between 22 and 90 degree C. The reference temperature of 22 °C is chosen for all the calibrations that have been carried out. The standard error of fit $S_{yx}$ is found to be 0.0045 from the equation $S_{yx} = \left( \frac{\sum (y_{data} - y_{curve})^2}{N - 3} \right)^{1/2}$ [41]. From Table 4.4 of the reference [41] $t_{21,95}$ is 2.08. So that the precision interval $\pm tS_{yx}$ is found to be ±0.93°C with 95% level of confidence.

![Figure 4.7 TSP calibration result, Tr=22 °C](image)

$y = -1.0952x^3 + 2.6887x^2 - 2.9923x + 1.3959$

$R^2 = 0.9995$
4.3.2 PSP calibration result

The PSP is calibrated at different temperatures in the calibration chamber, in which both the pressure and temperature are well controlled. The PSP is calibrated under 15 C, 25 C, 35 C, and 45 C since the effective temperature range for “self-correction” is from 5 to 45 degree C. The reference condition is ambient pressure (14.7 psia) and 25 degree C. The calibration curve is shown in figure 4.8. The intensity ratio of pressure channel and reference channel are plot respectively in figure 4.8 (a) and (b), followed by the plot of Ro/R versus pressure, which is the final calibration result. It can be seen that calibration curves of different temperature as seen in figure 4.8 (a) collapse into one curve (figure 4.8 (c)) after temperature correction. Figure 4.9 plots the full pressure range PSP calibration curve and it is used for experimental pressure calculation. The pressure range is from 5 to 25 psig. The calibration error is calculated the same way as TSP calibration and the precision interval is found to be ±0.12 psi.

The base coat of the Binary-FIB kit (also called Uni-FIB PSP in ISSI product list) is a good PSP itself with relatively low temperature sensitivity. It can be applied in a test where there is very small temperature variation across the measurement domain.
(a) Pressure channel data

(b) Reference channel data
(c) PSP calibration data with temperature correction

Figure 4.8 Bi-luminophore PSP calibration data

\[ y = 0.04477x + 0.31631 \]

\[ R^2 = 0.99813 \]

Figure 4.9 Full range PSP calibration curve
4.4 Investigation of TSP degradation

4.4.1 Background of photodegradation

Photodegradation of the luminescent paint occurs when the paint is exposed to the illumination light. The degradation process depends on illumination intensity, exposure time, and the paint formula. This phenomenon is also commonly referred to as photobleaching. It is still a very poorly understood phenomenon, which leads to a dramatic loss of fluorescence emission intensity. The TSP measurement will be impaired if there is significant photodegradation occurring between the reference and the test conditions. Usually in TSP calibration test or wind tunnel experiment, in order to minimize the paint degradation, the illumination light is kept on only when the measurement pictures are taken. But the degradation must be quantified to make sure it has negligible effect on measurement accuracy. In the extreme case where the continuous lighting is required in the experiment (e.g., transient heat transfer experiment), the paint degradation effect has to be compensated. This is the objective of this paint degradation study.

Photodegradation occurs when a luminophore permanently loses the ability to luminance due to photon-induced chemical damage. Upon transition from an excited singlet state to the excited triplet state, luminophores may interact with another molecule to produce irreversible modifications [42]. The triplet state is relatively long-lived with respect to the singlet state, thus allowing excited molecules a much longer timeframe to undergo chemical reactions with components in the environment. This light-induced chemical reaction is an irreversible process and mechanistically distinct from oxygen quenching of luminescence pertaining to PSP. Oxygen quenching is a reversible photophysical process that does not produce chemically modified
ground state products [42]. A detailed discussion of all the photochemical processes that lead to photobleaching is beyond the scope of this work.

4.4.2 Test procedure

The test matrix of investigation on TSP degradation has been designed as follows:

1. To test the temperature effect: maintain the temperature of the aluminum test coupon at a certain temperature for each test. Temperatures to be tested are: 45 °C, 60 °C, and 75 °C

2. To test the effect of lighting intensity: the LED excitation light (464nm peak wavelength, provided by ISSI) is set up at different distances from the test coupon. The distances are 1 foot, 2 feet and 3 feet, which are the typical setup distance in the actual experiments.

3. To test the possible paint thickness effect: besides the coupons painted with 9 coatings (which is the number of coating applied consistently before), a test coupons with 12 coatings is also prepared. This 12 coating coupon will be tested at 75 C only, which is expected to bring high degradation arte. The LED is set up at 1 foot and 2 feet distances to compare with the results of thinner coatings.

The power output of blue LED illumination light is 0.25W. The directivity of light emission has a 15-deg half angle. The diameter of the illuminated area can be estimated at a certain distance from the light. The lighting intensity at various distances is then calculated by dividing the power by the area of the illuminated area. The calculated lighting intensities at
distances of 1 foot, 2 feet and 3 feet are 1.2 mW/cm², 0.3 mW/cm² and 0.13 mW/cm², respectively.

The camera setup (exposure time, aperture of the lens) was kept the same for each degradation test. The LED light source remains on during each test, which lasts for 2 to 3 hours. Measurement picture is taken every 30 minutes.

4.4.3 Results and discussion

Figure 4.10 shows the degradation behavior of the UniCoat TSP under various lighting intensity and temperature conditions. Two to three repetitions were completed for each test condition. The observations are:

1. Luminescence intensity of TSP decreases with time at elevated temperature condition when the paint is continuously exposed to lighting

2. This intensity change has an almost linear correlation with time up to 180 min.

3. The degradation becomes severe with higher temperature

4. To verify that if TSP degrades with high temperature only, the lighting was removed from the test coupon for one hour in two of the tests, while the temperature was maintained at the same condition. Figure 4.13 shows that when the paint is covered from light during the degradation test, the degradation stops for both 60 °C and 75 °C condition. After the lighting condition was recovered, the degradation continues at the same rate as it was before the lighting was removed. This test proves that photo degradation does not occur without lighting.

5. One test coupon painted with 12 coatings of TSP was tested at 75 °C with 1.2 and 0.3 mW/cm² lighting power. The trend of intensity change is essentially the same as 9 coating
coupons, which implies that the paint thickness does not have appreciable effect on degradation rate under normal condition of TSP application. The data of this test coupon was mixed with the other data of repetitions to develop the degradation curve.

A TSP degradation correlation is developed to quantify the change of TSP fluorescence intensity during photo degradation process. Theoretically the photodegradation process is described by the equation below [42]:

\[
I(t) = A + B \exp(-K_d t)
\]  \( (4.2) \)

where \( I(t) \) is the fluorescence intensity at time \( t \); \( A \) is fluorescence intensity at \( t = \infty \); \( A+B \) is the fluorescence intensity at \( t=0 \); and \( K_d \) is the rate constant for the intensity change. However, based on the fluorescence intensity change in less than three hours of degradation interval, it is sufficient to modify the correlation into a simpler form below for engineering purpose:

\[
\ln\left(\frac{I}{I_o}\right) = -K_d t
\]  \( (4.3) \)

where \( I_o \) is the initial intensity at \( t=0 \); \( I \) is the intensity at time \( t \), \( t \) is the continues time of exposure to lighting, and \( K_d \) is the degradation rate constant, which is a function of temperature and lighting intensity as shown in figure 4.11 and 4.12.

\[
K_d = A I_e \exp\left(-\frac{B}{T}\right)
\]  \( (4.4) \)

\( I_e \) is the optical power of lighting in mW/cm\(^2\), \( T \) is the temperature in Kelvin at which the TSP is maintained, \( A \) and \( B \) are constants and equal to 11.8 mJ/cm\(^2\) and 3380 K, respectively. Figure 4.12 shows that the optical power of lighting has a linear effect on the degradation rate constant \( K_d \). The correlation (4.4) overestimates the degradation rate at high lighting power by as
much as 7.1% and underestimates degradation rate at low lighting power by greater magnitude (see Table 4.1). However, the correlation has the best accuracy at higher lighting power and higher temperature range, where the degradation rate is the highest and correction is most needed. Actually under 45 °C conditions, the degradation rate is so low that, even at high lighting power of 1.2 mW/cm², the intensity change causes only 2.5 °C in measurement error after 180 minutes of continuous degradation. While at 75 °C, the corresponding measurement error will be 8.2 °C. By applying equation 4.4 to compensate the intensity change, the temperature measurement can be corrected to within -2 °C accuracy at 75 °C.

(a) TSP degradation under 45 C
(b) TSP degradation under 60°C

(c) TSP degradation under 75°C

Figure 4.10 TSP degradation under various temperatures
Figure 4.11 Effect of temperature and lighting intensity on degradation rate constant

Figure 4.12 Effect of lighting intensity Ie on degradation rate constant Kd
Figure 4.13 (a) History of TSP intensity change with light removal in the process

Figure 4.13 (b) “Continuous” TSP intensity change without counting the time of light removal
Table 4.1 comparison of degradation constant $k$ obtained from experiment and correlation

<table>
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<th>temperature (K)</th>
<th>Intensity $I_e$ (mW/cm²)</th>
<th>$k$-experiment</th>
<th>$k$-correlation</th>
<th>difference (%)</th>
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<td>3.43E-04</td>
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<td>5.30E-04</td>
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<td>-22.10%</td>
</tr>
<tr>
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</tbody>
</table>
5.1 Jet nozzle consideration

There have been numerous experimental and numerical investigations on flow and heat transfer characteristics of impinging jets. Consider the jet nozzles that have been used, there are two extreme types: (i) tube type nozzle, which has large length to diameter ratio (L/D). The air jet coming out from the pipe has an initial velocity profile similar to that of pipe flow. (ii) Orifice in a plate type of nozzle, which has ratio of L/D close to unity. Most nozzles used in literature fall between these two extremes [43]. The second type of jet nozzle is more similar to those used in jet impingement design in gas turbine components and was considered in this study as a baseline configuration. It is called “plate jet” hereinafter. However, most of the studies in the literature applied tube type nozzle in the experiment, thus the applicability of the empirical heat transfer correlation for the impinging jet. In order to validate the experimental result and compare to the benchmark data in the literature, heat transfer of an impinging jet shooting out of a long tube nozzle was also studied under the same thermal boundary condition. This second type of jet is called “tube jet” hereinafter. A flat target surface was employed through out the experiments of the study. The target surface of impingement within turbine components, typically turbine endwalls, midchord regions of turbine vanes and blades, can be approximated as flat surfaces.
5.2 Instrumentation of heat transfer surface

Steady state heat transfer measurement was conducted in this work since the TSP does not have sufficient time response for a transient study. A typical heat transfer experimental set up using heating foil was instrumented for the steady state study.

The selection of the heating foil turned out to be a difficult issue. Initially a thin Inconel foil (δ=0.025 mm) was chosen due to the fact that the thickness of the foil relative to the surface area is such that negligible current conducts in the direction normal to the surface plane. Therefore, the electrical current and potential fields are two-dimensional. The electrical boundary conditions for the heater surface are a uniform potential at the bus bar/Inconel foil junctions and zero current flow normal to the two unbounded edges. Hence the assumption of constant heat flux generation becomes valid over the heated surface area, as long as the rectangular shaped foil is properly connected to two bus bars that are mounted in parallel to each other at two opposite edges of the foil. In a typical heat transfer experimental set up using heating foil, the foil is either glued or taped down to the model surface, usually being made of low thermal conductive material, to prevent the deformation of the foil when it is heated up during the experiment. However, in the current test, the Inconel surface that faces the acrylic plate (the low conductive material in this case) is painted with TSP. The painted surface has to remain intact all the time. Once it is glued or taped, the paint will be spoiled and the temperature measurement will not be accurate. Finally to resolve the foil warping/deformation issue, first a 0.5 mm and then a 0.2 mm thick Inconel 600 heat resisting foil (provided by GoodFellow Ltd.) were chosen. The Biot number in the direction of the foil thickness is 0.028 and 0.011 for 0.5 mm and 0.2 mm thick foil sheet, respectively. Hence the temperature difference across the
thickness of the foil is negligible. The determination of the heat flux distribution over the foil surface will be discussed in data reduction section below.

Figure 5.1 shows the instrumentation of the Plexiglas target plate for heat transfer experiment. A 10 cm by 16 cm Inconel foil heater is secured to the 12.7 mm thick Plexiglas target plate, which serves to be the thermal insulation material. One side of the Inconel foil is painted with TSP before it is attached to the Plexiglas plate surface. Two copper bus bars are mounted in parallel at the two edges of the foil. The unpainted side of the foil is facing the jet and the CCD camera captures the TSP image from the backside of the plate. The voltage across the bus bars is measured with a multimeter at each test run. Meanwhile the current is read from display window of the power supply directly.

5.3 Test Apparatus and Test Procedure

5.3.1 Experimental set up for plate jet

The test unit as shown in figure 5.2 (top) consists of an air supply line, a diffuser, a plenum and a setting chamber. Jet air is introduced into the plenum chamber from a compressor-tank system, which is designed for a blow-down type supersonic wind tunnel. A pressure regulator is installed in the air supply line to control the flow rate. In addition, a shutoff valve is provided as a safety measure. A McMillan thermal mass gas flow meter is installed at the inlet of the diffuser to read the actual flow rate through the jet nozzle. This flow rate reading is used to determine the average jet velocity at the nozzle exit and this average jet velocity is the characteristic velocity that is used to define the jet Reynolds number. The jet flow goes through a set of honeycomb screens before it reaches the setting chamber. The temperature of air in the
chamber is within 1 degree C of the ambient temperature for all tests performed. Figure 5.2 (bottom) shows the schematic of the test section. A 9.5 mm thick acrylic plate is attached to the top side of the chamber with a straight hole drilled at the center as the jet nozzle. The diameter of the jet nozzle is also 9.5 mm. In the heat transfer experiment the target plate prepared as in figure 5.1 is installed onto the threaded rods in the test section of the rig as shown in figure 5.3. Variation of Z/D is achieved by sliding the target plate along the threaded rods and securing it with nuts. To measure the surface pressure distribution, the jet-impinged surface of the smooth Plexiglas plate is painted with PSP coating. The CCD camera captures the PSP image from the backside of the target plate. The image data of TSP and PSP are processed with a MATLAB routine.

The CCD camera and the LED light were mounted on tripods behind the target plate. A 50mm fixed focal length Nikon lens was used in the experiment. The CCD camera was focused manually. The distance between the target plate and the camera was adjusted such that the rectangular TSP or PSP painted area is at the center of the field of view with about 20 mm margin along the four edges. The LED light was adjusted to obtain the most uniform luminescent intensity profile on the painted area as possible.

The CCD camera was allowed to cool to its regulated temperature in approximately 10 minutes. The LED light takes about 10 to 15 minutes to warm up. Camera exposure time was set to maximize CCD quantum well depth without risk of saturation. Eight images were taken for both initial wind-off and later wind-on image averaging. Since there is no motion or deformation occurred to the test surface in the jet impingement experiment, registration of wind-on and –off images is not necessary for both PSP and TSP image processing. It takes about 30 minutes for
the heated foil surface to reach thermal equilibrium in each test run after the power input is changed.

Free jet characteristics were also studied by measuring the centerline velocity and turbulence distribution with Pitot probe and single wire hot wire anemometer. The fixture of the probe holder is attached to the threaded rods and traverses are made in a plane parallel to the rods (along the axis of the jet) to measure the velocity and the turbulence intensity along the axis of the free jet.
Figure 5.1 Target plate setup

(a) Jet impinging side of the target plate

(b) Back side of the target plate
Figure 5.2 Schematic of jet impingement rig for plate jet (top) and its test section (bottom).
Figure 5.3 Pictures of experimental setup for plate jet
(a) Schematic of the tube jet test setup

(b) Picture of the tube jet test setup

Figure 5.4 Experimental setup for the tube jet test
5.3.2 Experimental set up for tube jet

The experimental set up on the target plate side of the tube jet test is the same as plate jet test. The plenum and the injection plate are replaced by a long copper tube, which is secured with a wooden frame and arranged perpendicular to the target surface. Figure 5.4 shows the schematic of the tube jet test setup. The tube diameter is 0.5 inch and length to diameter ratio is 15. The tube is fed by the same compressed air source as in the plate jet experiment. The nozzle outlet edge was smoothed with fine sand paper to make sure there are no burrs to disturb the jet flow at the nozzle exit.

The separation distance, Z/D, that has been tested are1.5, 3, 5, and 8, combined with jet Reynolds numbers of 20000, 40000 and 60000.

5.4 Data reduction

Due to the thickness of the Inconel foil, the constant heat flux assumption is no longer valid. Instead it is treated as constant heat generation per unit volume:

\[ q_{\text{gen}} = \frac{\text{Volt} \times \text{Amp}}{V} \]  

(5.1)

where \( q_{\text{gen}} \) (W/m\(^3\)) is the volumetric heat generation inside the foil. Volt and Amp are the Voltage and Amperage of the power input across the bus bars respectively. \( V \) is the volume of the heating foil, which is the heated foil surface area multiplied by the foil thickness \( \delta \).

After obtaining the surface temperature with TSP, the temperature is averaged in circumferential direction due to the axisymmetric pattern of temperature distribution. Then this averaged temperature is plotted as a function of radial distance \( r \) (see figure 6.4 to 6.19). By
applying energy balance to the differential control volume in the foil sheet as shown in figure 5.5, the following form of heat equation is obtained in cylindrical coordinates..

\[ q''_r - q''_{r+dr} - q''_{\text{cond}} + q''_g - q''_e = 0 \]  \hspace{1cm} (5.2)

where

\[ q''_r = -k_r \frac{\partial T}{\partial r} \quad q''_r - q''_{r+dr} = \frac{1}{r} \frac{\partial}{\partial r} (k_r r \frac{\partial T}{\partial r}) \]  \hspace{1cm} (5.3)

In equation (5.2), \( q''_e \) is the effective local surface heat flux that is removed by convection of impinging jet, \( q''_g \) is the heat generation term which equals to \( (q''_{\text{gen}} \times \delta) \), and \( q''_{\text{cond}} \) is the conduction heat loss through the Plexiglas plate. The lateral heat conduction term \( q''_r \) is induced by the temperature gradient in \( r \) direction along the jet impinged foil surface. If the foil were thin enough, the lateral conduction would be negligibly small compared to the heat
generation term. $k_f$ is the thermal conductivity of the Inconel foil material. Hence $q''_e$ can be expressed as:

$$q''_e (r) = \frac{1}{r} \frac{\partial}{\partial r} (k_f r \frac{\partial T}{\partial r}) - q''_{\text{cond}} + q''_g$$  \hspace{1cm} (5.4)$$

The local heat transfer coefficient $h$, which is a function of $r$, can be expressed as:

$$h(r) = \frac{q''_e (r)}{T_w (r) - T_j} = \frac{1}{r} \frac{\partial}{\partial r} (k_f r \frac{\partial T}{\partial r}) + q''_g - q''_{\text{cond}}$$  \hspace{1cm} (5.5)$$

The radiation loss is found to be less than 0.3% of $q''_e$ and is neglected in the heat equation. The surface temperature $T_w (r)$ is measured by TSP, and $T_j$ is the total temperature of jet air.

Experimental results of heat transfer are presented in terms of Nusselt number:

$$Nu(r) = \frac{h(r)D}{k_{\text{air}}}$$  \hspace{1cm} (5.6)$$

As the jet is axisymmetric, the radial distribution of $Nu$ can be averaged to give the mean Nusselt number:

$$\overline{Nu} = \frac{2}{R^2} \int_0^R Nu(r) r dr$$  \hspace{1cm} (5.7)$$

Appendix A shows a detail example of calculation for both local and average Nusselt number based on the power input and circumferentially averaged temperature distribution.
CHAPTER 6 RESULTS AND DISCUSSIONS

6.1 Experimental results of plate jet

6.1.1 Flow measurement of the plate jet

Free jet centerline velocity ($U_m$) distribution is measured using Pitot probe and plotted in figure 6.1. It is normalized by the jet velocity at nozzle exit $U_e$. The length of the potential core is found to be about 4 times the diameter of jet nozzle for Re = 20000, 40000 and 60000. Free jet centerline turbulence intensity distribution is measured with hot wire anemometer. Figure 6.2 shows the distribution of absolute magnitude of the velocity fluctuations, as shown by the curve of $u'/U_m$, in which the R.M.S fluctuation of the centerline velocity ($u'$), normalized by the local time average centerline velocity ($U_m$), increases continuously with jet length up to $Z/D=9$.

Figure 6.3 to 6.5 are the PSP images of pressure distribution across the jet impacted surface for $Z/D=1.5$, 5 and 8 at jet Reynolds number of 60000. This static pressure data is averaged circumferentially and plotted as a function of $r/D$ in figure 6.6. The region of impingement is marked by the end of negative pressure gradients on the impacted surface. As expected, the boundary of the impingement region expands with the increasing $Z/D$. Beyond the impingement region, the pressure is atmospheric.

The PSP measurement for impinging jet of lower Reynolds number was not successful due to small pressure change between reference and test conditions. For example, at jet Reynolds number of 40000, the plenum pressure is only about 0.5 psig and the maximum static pressure at the target surface is less than 0.3 psig. The minimum pressure difference (the difference between the pressure to be measured and the reference pressure—usually atmospheric pressure) that PSP
can discern is about 0.3 psi. A pressure change of 0.3 psi corresponds to an intensity change of 1.4%, which is of the same order of intensity change caused by high frequency spatial noise that is induced by microscopically granular surface finish of the painted surface. Details about this high frequency spatial noise are discussed in the next section. The adverse effect of the high frequency spatial noise is already seen in figure 6.5 where the maximum surface pressure is about 0.4 psig for Z/D=8 and jet Reynolds number of 60000. In figure 6.6, the effect of spatial noise is minimized by circumferentially averaging the PSP data. Since PSP is an absolute pressure measurement technique, high accuracy is hard to achieve when the pressure to be measured is within 0.3 psi difference to the reference pressure.

![Graph showing free jet centerline velocity measured with Pitot probe](image)

Figure 6.1 Free jet centerline velocity measured with Pitot probe
Figure 6.2 Distribution of free jet centerline turbulence intensity

Figure 6.3 Pressure distribution across the target surface, $Z/D=1.5$, $Re=60000$
Figure 6.4 Pressure distribution across the target surface, Z/D=5, Re =60000

Figure 6.5 Pressure distribution across the target surface, Z/D=8, Re=60000
Figure 6.6 Distribution of static pressure over the impacted surface, Re = 60,000.

6.1.2 Heat transfer result of plate jet

Figure 6.7 to 6.22 show the temperature distribution on the impingement surface measured by TSP and circumferentially averaged temperature in radial direction for each test run of plate jet experiment. To the naked eye, a processed image of both TSP and PSP has a somewhat granular quality when the image is zoomed in to pixel level, especially when the image is taken by a CCD camera of high spatial resolution. This is caused by the microscopically rough surface finish after paint dries, characterized by a fine granularity. The surface roughness can cause high frequency spatial noise in a processed TSP or PSP image. The high frequency noise was usually smoothed out first by “medfilt2” operator in MATLAB code. The spatial resolution (8 to 10 pixels per millimeter) is usually more than enough for data presenting. Hence the final TSP images were obtained by averaging temperature over each 8 by 8 pixels area.
(depending on the actual resolution of the image) and thus converting the x-y coordinates from raw pixel to millimeter. In this step the spatial resolution is sacrificed to further smooth out high frequency noise in the fully processed image.

The jet nozzle to target distances Z/D tested with the plate jet are 1.5, 5, 8, and 12. The power input and the calculated volume heat generation in the foil heater for each test are listed in table 6.1. The power input values are decided after several trial tests. In order to reduce the measurement error in heat transfer, especially in the stagnation region where maximum heat transfer occurs, higher temperature difference between the jet air and target surface \((T_w-T_j)\) is desired. However, the temperature gradient from the stagnation point to downstream of wall jet region is steep. TSP starts to melt when the temperature is above 120 °C. The power input that are finally applied is a compromise such that \((T_w-T_j)\) at stagnation region is at least 15 °C while the highest temperature on the foil surface is no more than 100 °C.

In order to verify the concentric pattern of the temperature distribution, Figure 6.23 plots the temperature distribution in the circumferential direction at various r location for Z/D=1.5, and 5 with Re of 10000 and 60000. The average temperature fluctuation at each r location is only about 2 to 3 °C, which proves the axisymmetric jet flow condition and, meanwhile, justifies the data reduction procedure.

Radial distributions of Nusselt number for all Z/D cases are plotted in figure 6.24 to 6.27. Instead of having the highest heat transfer rate at the stagnation point, which occurs at all the other Z/D cases, the peak Nusselt number is found at about r/D=1.8 when Z/D is 1.5. The radial location of this “second peak” seems to coincide with the data in the literature (Fig 3.6). It is located right at where the negative pressure gradient ends, according to the PSP result shown in
figure 6.6. Hence this local high heat transfer rate can be explained by the transition from laminar to turbulent boundary layer that occurs at the edge of the stagnation region. The second peak of Nusselt number decreases as Reynolds number decreases. When Re =10000, there is merely a hump remaining at the location of the second peak, the value of which is already lower than the stagnation point Nusselt number. For all the other Z/D cases, the Nusselt number decreases monotonically from the stagnation point.

The effect of Z/D on the stagnation point Nu₀ is plotted in figure 6.28. Nu₀ reaches maximum at about Z/D equal to 5. From the Pitot probe measurement it is found that the length of the potential core is 4 times the jet nozzle diameter. Beyond the potential core, with further increase in axial distance, the interaction between the attenuation of approaching jet velocity and the continuous increase in centerline turbulence intensity brings about a maximum heat transfer coefficient at Z/D around 5.

In figure 6.29 to 6.32, for various Z/D, the experimental data of area averaged Nusselt number Nu_avg are plotted against r/D and compared to those calculated using Martin’s correlation [23] for single jet (Equation 3.13):

Note that the valid range of Martin’s correlation is: 2000 ≤ Re ≤ 400,000; 2.5 ≤ r/D ≤ 7.5; 2 ≤ Z/D ≤ 12. As r/D increases, the data from the empirical correlation decreases monotonically from the beginning of the valid range (r/D=2.5). Detailed comparison between experimental and empirical data can be found in Table 6.2. The difference in percentage is defined as: 100 * (experiment – correlation) / correlation. Although Z/D=1.5 is not in the valid range of the correlation, Nusselt number is also calculated for this case and compared with the experimental data for verification purpose.
For all Z/D cases, the experimental data matches the correlation data closely at lower Reynolds number of 10000 and 20000. Correlation overestimates Nu at all Z/D for jet Reynolds number of 40000 and 60000. The difference increases with jet Reynolds number as well as r/D, the latter indicates that the experimental Nusselt number decreases with r/D faster than the empirical one.

In figure 6.33 the area averaged Nusselt number $\text{Nu}_{\text{avg}}$ is plotted against Z/D and also compared with Martin’s empirical data. For Z/D=1.5 and 5, the experimental values of $\text{Nu}_{\text{avg}}$ are essentially the same for lower jet Reynolds number. At the highest Reynolds number of 60000, the maximum $\text{Nu}_{\text{avg}}$ appears at Z/D=5. As Z/D further increases, the $\text{Nu}_{\text{avg}}$ starts to decrease. Nusselt number calculated from Martin’s correlation are higher than the experimental data and the difference between them increases with increasing jet Reynolds number, as already seen in the previous plots. Additionally, since empirical data decreases with Z/D monotonically, it could not reflect any possible optimum Z/D where Nusselt number reaches maximum. Comparatively, experimental average Nusselt number still has an optimum value at Z/D about 5, although this optimum is not as appreciable as stagnation point Nusselt number.

The difference between the experimental and empirical data might result from the difference in the jet nozzle configuration. Note that Martin’s correlation was based on the experimental data that obtained with either a long tube-type nozzle or a shaped nozzle with a smooth contraction. At the next step, heat transfer measurement is performed with a tube jet and the same setup of the target plate that is employed in the plate jet test.
Figure 6.7 (a) Plate jet temperature distribution; Z/D=1.5, Re=10000

Figure 6.7 (b) Circumferentially-averaged temperature as a function of r, Z/D=1.5, Re=10000; plate jet

$y = -0.1753x^5 + 1.2384x^4 - 2.3781x^3 + 3.8216x^2 + 0.3704x + 39.099$

$R^2 = 1$
Figure 6.8 (a) Plate jet temperature distribution, $Z/D=1.5$, $Re=20000$

Figure 6.8 (b) Circumferentially averaged temperature as a function of $r$, $Z/D=1.5$, $Re=20000$; plate jet
Figure 6.9 (a) Plate jet temperature distribution, $Z/D=1.5$, $Re=40000$

Figure 6.9 (b) Circumferentially averaged temperature as a function of $r$, $Z/D=1.5$, $Re=40000$; plate jet

$$y = 0.032x^5 - 0.6581x^4 + 4.1077x^3 - 6.2985x^2 + 3.61x + 40.342$$
$$R^2 = 0.9998$$
Figure 6.10 (a) Plate jet temperature distribution, $Z/D=1.5$, $Re=60000$

$$y = 0.0116x^5 - 0.445x^4 + 3.5093x^3 - 6.0807x^2 + 3.1396x + 38.668$$

$R^2 = 0.9998$

Figure 6.10 (b) Circumferentially averaged temperature as a function of $r$, $Z/D=1.5$, $Re=60000$; plate jet
Figure 6.11 (a) Plate jet temperature distribution, Z/D=5, Re=10000

\[ y = -0.1148x^5 + 1.1891x^4 - 4.4598x^3 \\
+ 7.631x^2 + 0.8843x + 40.223 \]

\[ R^2 = 0.9998 \]

Figure 6.11 (b) Circumferentially averaged temperature as a function of r, Z/D=5, Re=10000; plate jet
Figure 6.12 (a) Plate jet temperature distribution; Z/D=5, Re=20000

Figure 6.12 (b) Circumferentially averaged temperature as a function of r, Z/D=5, Re=20000; plate jet
Figure 6.13 (a) Plate jet temperature distribution, Z/D=5, Re=40000

Figure 6.13 (b) Circumferentially averaged temperature as a function of r, Z/D=5, Re=40000; plate jet

\[ y = -0.1426x^5 + 1.4483x^4 - 5.2124x^3 + 8.2041x^2 + 1.3462x + 41.053 \]

\[ R^2 = 0.9998 \]
Figure 6.14 (a) Plate jet temperature distribution, Z/D=5, Re =60000

Figure 6.14 (b) Circumferentially averaged temperature as a function of r, Z/D=5, Re=60000; plate jet

\[ y = -0.0958x^5 + 1.0124x^4 - 3.8173x^3 \\
+ 6.3766x^2 + 0.8245x + 35.117 \\
R^2 = 0.9999 \]
Figure 6.15 (a) Plate jet temperature distribution, Z/D=12, Re =10000

![Plate jet temperature distribution](image)

Figure 6.15 (b) Circumferentially averaged temperature as a function of r, Z/D=8, Re=10000; plate jet

![Circumferentially averaged temperature](image)

\[ y = 0.0109x^3 + 0.3624x^4 - 3.4389x^3 + 8.9275x^2 - 1.1723x + 38.52 \]

\[ R^2 = 0.9996 \]
Figure 6.16 (a) Plate jet temperature distribution, Z/D=8, Re =20000

Figure 6.16 (b) Circumferentially averaged temperature as a function of r, Z/D=8, Re=20000; plate jet
Figure 6.17 (a) Plate jet temperature distribution, Z/D=8, Re =40000

Figure 6.17 (b) Circumferentially averaged temperature as a function of r, Z/D=8, Re=40000; plate jet
Figure 6.18 (a) Plate jet temperature distribution, Z/D=8, Re =60000

\[
y = -0.0672x^5 + 0.816x^4 - 3.5989x^3 \\
+ 6.7158x^2 + 0.112x + 38.873
\]
\[ R^2 = 0.9999 \]

Figure 6.18 (b) Circumferentially averaged temperature as a function of r, Z/D=8, Re=60000; plate jet
Figure 6.19 (a) Plate jet temperature distribution, Z/D=12, Re=10000

Figure 6.19 (b) Circumferentially averaged temperature as a function of r, Z/D=12, Re=10000; plate jet

\[ y = -0.0399x^5 + 0.5011x^4 - 2.4669x^3 + 5.6088x^2 - 1.203x + 39.722 \]
\[ R^2 = 0.9999 \]
Figure 6.20 (a) Plate jet temperature distribution, Z/D=12, Re =20000

Figure 6.20 (b) Circumferentially averaged temperature as a function of r, Z/D=12, Re=20000; plate jet
Figure 6.21 (a) Plate jet temperature distribution, $Z/D=12$, $Re=40000$

Figure 6.21 (b) Circumferentially averaged temperature as a function of $r$, $Z/D=12$, $Re=40000$; plate jet
Figure 6.22 (a) Plate jet temperature distribution, $Z/D=12$, $Re=60000$

$$y = -0.0582x^5 + 0.6628x^4 - 2.9561x^3 + 6.1704x^2 - 0.597x + 43.755$$

$R^2 = 0.9999$

Figure 6.22 (b) Circumferentially averaged temperature as a function of $r$, $Z/D=12$, $Re=60000$; plate jet
Table 6.1 Power input and heat generation for the plate jet experiment

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<th>Re</th>
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<th>40000</th>
<th>20000</th>
<th>10000</th>
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<td>206</td>
<td>182</td>
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<td>V (Volt)</td>
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<td>10000</td>
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(a) \( Z/D = 1.5, \text{ Re} = 60000 \)

(b) \( Z/D = 1.5, \text{ Re} = 10000 \)
Figure 6.23 Typical temperature variations in circumferential direction (theta)
Figure 6.24 Local Nu distribution of plate jet, Z/D=1.5

Figure 6.25 Local Nu distribution of plate jet, Z/D=8
Figure 6.26 Local Nu distribution of plate jet, Z/D=8

Figure 6.27 Local Nu distribution of plate jet, Z/D=12
Figure 6.28 Effect of Z/D on Stagnation point Nuo of plate jet
**Figure 6.29** Area-averaged Nusselt number distribution of plate jet, $Z/D=1.5$

**Figure 6.30** Area-averaged Nusselt number distribution of plate jet, $Z/D=5$
Figure 6.31 Area-averaged Nusselt number distribution of plate jet, $Z/D=8$

Figure 6.32 Area-averaged Nusselt number distribution of plate jet, $Z/D=12$
Figure 6.33 Effect of Z/D on area-averaged Nu (averaged over r/D=3.5)
Table 6.2 Comparison of averaged Nusselt number between experimental data of plate jet and empirical data

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6.2 Experimental result of tube jet

Figure 6.34 to 6.45 show the temperature distribution across the target surface and circumferentially averaged temperature for each test run of tube jet experiment. The power input and the calculated volume heat generation in the foil heater are listed in table 6.3.

From the plate jet test we observe the “second peak” in local Nusselt number at small spacing of Z/D=1.5. Since this ‘second peak” phenomenon is of most interest, in tube jet experiment Z/D=3 is tested in addition to Z/D=1.5. From the raw TSP images of Z/D=1.5 and 3 (Figure 6.37 to 6.42), a blue color cold ring zone outside the low temperature stagnation region can be clearly seen, which corresponds to the local high heat transfer rate other than the stagnation region (second peak in Nu). This local low temperature ring zone becomes more and more dominant as Z/D decreases and jet Reynolds number increases. At Z/D=1.5 and Re=60000, the width of the blue ring zone reaches the maximum and the local temperature drops down to a value that is closest to the lowest temperature at stagnation point.

Figure 6.46 to 6.49 show the radial distribution of Nusselt number. For both Z/D=1.5 and 3, as r/D increases, Nu starts to drop from the first peak at the stagnation point, reaches a local minimum at about r/D=1.1 to 1.25, and then goes back up to form the second peak. The wavy pattern of the heat transfer distribution is much more pronounced than the plate jet case with Z/D=1.5. When Z/D is fixed, the radial location of the second peak does not seem to change with jet Reynolds number. However, it is interesting to notice that, by comparing the Nusselt number distribution for Z/D=1.5 and 3 (Figure 6.50 to 6.52), the second peak in Nu shifts radially inward as Z/D is decreased. It is speculated that the enhanced flow acceleration as spacing Z/D decreases may reduce the area of impingement zone, which is marked by the end of the pressure
gradient along the impingement surface. Hence the transition from laminar to turbulent wall jet flow starts at a smaller radial location as Z/D decreases, which corresponds to where the second peak in heat transfer occurs.

It is also interesting to compare the data between Z/D=1.5 and 3 to find that, the height of the second peak (the difference between the local minima and the second peak) for Z/D =1.5 is larger than that of Z/D=3. Generally local Nu value of Z/D=3 is higher than that of Z/D=1.5. The difference in local Nu between these two cases decreases as jet Reynolds number goes down. For Re equals to 20000, the second peak in Nu of Z/D=1.5 exceeds that of Z/D=3 due to its larger amplitude of the second peak as well as its inward shift of the second peak location.

The reason of disappearance of the second peak in local heat transfer when Z/D is beyond the potential core could be twofold. First the turbulence intensity on the jet axis increases continuously as Z/D increases from 2 to 9, which results in progressively higher heat transfer rate at stagnation region. On the other hand, the approaching velocity continuously decreases as a result of increasing Z/D, which leads to weaker flow acceleration at stagnation region compared to lower Z/D condition. The already higher local turbulence intensity plus the smaller pressure gradient along the impingement surface diminish the effect of transition from laminar to turbulent boundary layer. Therefore, the stagnation point heat transfer dominates in local heat transfer distribution when Z/D is larger.

The effect of Z/D on the stagnation point Nu_0 is plotted in figure 6.53. Similar to plate jet case, Nu_0 reaches maximum at about Z/D equal to 5. This optimum Z/D becomes more pronounced at higher jet Reynolds number.
In figure 6.54 the area averaged Nusselt number (averaged over r/D=3) is plotted against Z/D and compared with Martin’s empirical data. Unlike plate jet which barely has any difference for Z/D<5, tube jet seems to have a maximum Nu_{avg} at about Z/D=3. Above all, the experimental data matches Martin’s correlation very well. For Reynolds number of 20000, the slope of the decreasing Nusselt number is about the same between experimental and correlation, although the experimental data is approximately 10% higher than the empirical one. For Reynolds number of 40000 and 60000, the agreement between experimental and empirical data is also very good, especially for Z/D=3 and 5.

Figure 6.55 to 6.58 compare the experimental and empirical average Nusselt number as a function of r/D. Difference in percentage is listed in table 6.4. Although Z/D=1.5 is not in the valid range of the correlation, the data calculated from the correlation still matches the experimental data reasonably well at Reynolds number of 20000 with maximum difference of 6.2% at largest r/D. At higher Reynolds number, the experimental Nu is greater than the empirical one. For Z/D=3 and 5, the agreement between two sets of data is exceptional, especially for Z/D=5. The maximum difference is only 3.6% for Reynolds number of 40000 and 60000. For Reynolds number of 20000, experimental data is consistently larger than the empirical data with an average difference about 10%, which is of the same order of measurement uncertainty. Just like the plate jet case, at Z/D=8, the correlation data does not match the experimental data very well. It underestimates heat transfer rate at lower Reynolds number and overestimates heat transfer rate at higher Reynolds number.

Finally, local Nusselt number of plate jet and tube jet are compared in figure 6.59 to 6.61, for Z/D of 1.5, 5, and 8, respectively. The heat transfer rate of tube jet is higher than that of plate
jet for all combinations of Z/D and Reynolds number. The difference is most significant at smaller Z/D and higher Reynolds number. As Z/D increases, the difference goes down. When separation distance Z/D is relatively large, the difference in flow structure of impinging jet for these two cases is not appreciable. When Z/D decreases, the tube-type nozzle has larger entrainment effect than plate nozzle since there is no blockage to prevent entraining surrounding air into the jet flow. Meanwhile, the flow of plate jet is confined between the two parallel plates (the injection plate and the target plate), especially when Z/D is short. The greater entrainment is expected to generate higher turbulence intensity across the jet flow, which leads to higher overall heat transfer rate.
Figure 6.34 (a) Tube jet temperature distribution, Z/D=8, Re =60000

\[ y = -0.002x^5 + 0.1147x^4 - 1.069x^3 + 3.1602x^2 + 1.9199x + 43.461 \]
\[ R^2 = 0.9999 \]

Figure 6.34 (b) Circumferentially averaged temperature as a function of r, Z/D=8, Re=60000; tube jet
Figure 6.35 (a) Tube jet temperature distribution, Z/D=8, Re=40000

y = 0.0246x^5 - 0.0602x^4 - 0.9624x^3 + 4.1703x^2 + 1.7024x + 48.232

R² = 1

Figure 6.35 (b) Circumferentially averaged temperature as a function of r, Z/D=8, Re=40000; tube jet
Figure 6.36 (a) Tube jet temperature distribution, Z/D=8, Re=20000

Figure 6.36 (b) Circumferentially averaged temperature as a function of r, Z/D=8, Re=20000; tube jet

\[
y = 0.1069x^5 - 1.081x^4 + 3.417x^3 - 2.5363x^2 + 1.4221x + 38.732
\]

\[
R^2 = 0.9996
\]
Figure 6.37 (a) Tube jet temperature distribution, Z/D=5, Re =60000

Figure 6.37 (b) Circumferentially averaged temperature as a function of r, Z/D=5, Re=60000; tube jet
Figure 6.38 (a) Tube jet temperature distribution, Z/D=8, Re =40000

Figure 6.38 (b) Circumferentially averaged temperature as a function of r, Z/D=5, Re=40000; tube jet
Figure 6.39 (a) Tube jet temperature distribution, $Z/D=8$, $Re=20000$

\[ y = -0.0735x^5 + 0.9941x^4 - 4.7284x^3 + 9.2002x^2 - 0.6811x + 47.895 \]

$R^2 = 0.9999$

Figure 6.39 (b) Circumferentially averaged temperature as a function of $r$, $Z/D=5$, $Re=20000$; tube jet
Figure 6.40 (a) Tube jet temperature distribution, Z/D=3, Re =60000

Figure 6.40 (b) Circumferentially averaged temperature as a function of r, Z/D=3, Re=60000; tube jet
Figure 6.41 (a) Tube jet temperature distribution, Z/D=3, Re =40000

Figure 6.41 (b) Circumferentially averaged temperature as a function of r, Z/D=3, Re=40000; tube jet
Figure 6.42 (a) Tube jet temperature distribution, Z/D=3, Re=20000

Figure 6.42 (b) Circumferentially averaged temperature as a function of r, Z/D=3, Re=20000; tube jet

\[ y = -0.294x^5 + 3.287x^4 - 12.7x^3 + 19.658x^2 - 6.3497x + 48.621 \]

\[ R^2 = 0.9982 \]
Figure 6.43 (a) Tube jet temperature distribution, $Z/D=1.5$, $Re=60000$

Figure 6.43 (b) Circumferentially averaged temperature as a function of $r$, $Z/D=1.5$, $Re=60000$; tube jet
Figure 6.44 (a) Tube jet temperature distribution, Z/D=1.5, Re=40000

\[
y = -0.6666x^5 + 6.421x^4 - 21.27x^3 + 27.668x^2 - 9.1843x + 48.887
\]

\[R^2 = 0.982\]

Figure 6.44 (b) Circumferentially averaged temperature as a function of r, Z/D=1.5, Re=40000; tube jet
Figure 6.45 (a) Tube jet temperature distribution, $Z/D=1.5$, $Re=20000$

$$y = -0.5608x^5 + 5.5149x^4 - 18.758x^3 + 25.578x^2 - 8.5834x + 50.392$$

$R^2 = 0.9926$

Figure 6.45 (b) Circumferentially averaged temperature as a function of $r$, $Z/D=1.5$, $Re=20000$; tube jet
Table 6.3 Power input for the tube jet test

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Figure 6.46 Local Nu distribution of tube jet, $Z/D=1.5$
Figure 6.47 Local Nu distribution of tube jet, Z/D=3

Figure 6.48 Local Nu distribution of tube jet, Z/D=5
Figure 6.49 Local Nu distribution of tube jet, Z/D=8

Figure 6.50 Comparison of local Nu distribution between Z/D=1.5 and 3, Re=20000

138
Figure 6.51 Comparison of local Nu distribution between Z/D=1.5 and 3, Re=40000

Figure 6.52 Comparison of local Nu distribution between Z/D=1.5 and 3, Re=60000
Figure 6.53 Effect of Z/D on Stagnation point Nuo of tube jet

Figure 6.54 Effect of Z/D on area-averaged Nu (averaged over r/D=3) of tube jet
Figure 6.55 Area-averaged Nusselt number distribution of tube jet, Z/D=1.5

Figure 6.56 Area-averaged Nusselt number distribution of tube jet, Z/D=3
Figure 6.57 Area-averaged Nusselt number distribution of tube jet, $Z/D=5$

Figure 6.58 Area-averaged Nusselt number distribution of tube jet, $Z/D=8$
Table 6.4 Comparison of averaged Nusselt number between tube jet experimental and empirical data

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<tr>
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<td>-13.8%</td>
</tr>
<tr>
<td>3.11</td>
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<td>146</td>
<td>-13.7%</td>
</tr>
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</table>
Figure 6.59 Comparison of plate jet and tube jet-local Nu as a function of r/D, Z/D=1.5

Figure 6.60 Comparison of plate jet and tube jet-local Nu as a function of r/D, Z/D=5
6.3 Study of surface roughness effect on jet impingement heat transfer

6.3.1 Overview

From the literature it is found that the roughened impingement target surface has elevated heat transfer compared to the smooth surface. The effect is most significant near the stagnation zone. The flow structure of the rough wall boundary layer near the stagnation zone is not clear. Near the stagnation region the flow is highly accelerated and it would tend to damp boundary layer turbulence originated by either roughness or free stream turbulence. Roughness elements would disrupt a thin thermal boundary layer and lower the thermal resistance, thus increasing the heat transfer.
The motivation of the present study is due to the fact that many surfaces that require impingement cooling in gas turbine components are rough, while the existing correlations of impingement heat transfer apply to smooth surfaces only.

6.3.2 Design of experiment

The test rig set up is essentially the same as the plate jet experiment as shown in figure 5.2. Only Z/D=5 is tested in this experiment since it is proved in the previous sections that the maximum heat transfer rate is achieved at this separation distance. The surface roughness is artificially created by randomly spreading Nickel particles onto Inconel surface prepared with a thin epoxy layer on it. A high temperature epoxy (JB Welder) is selected. After properly mixing the welder and the hardener, the epoxy is spread uniformly over one side of the Inconel plate. Before the epoxy cures, Nickel particles were sifted through a coffee filter and randomly spread over the epoxy surface. As the epoxy dries, the particles were glued firmly and act as roughness element in the surface. Two different sizes of the metal particle mesh are used to test the effect of surface roughness on the impingement heat transfer. They are 40 micron and 110 micron. The plate roughened with 40 micron particles is called rough-I plate hereinafter, and the other plate roughened with 110 micron particles is called rough-II plate. A baseline test was performed first with only thin smooth epoxy layer on the impingement surface, which is prepared exactly the same way as the first step in making rough plate. The purpose of this test is to find out the thermal resistance of the thin layer of epoxy. At the same time, the TSP measured temperature needs to be adjusted for a temperature drop through the thin epoxy layer, to obtain a local surface temperature for calculating local heat transfer coefficient. The heat transfer result obtained from
the roughened surface is then compared directly to the data of this test. Figure 6.62 shows the picture of four Inconel plates used in the experiment. The upper left plate is the smooth plate used in the early test in section 6.1 and 6.2. The lower left one is the plate coated only with the epoxy layer. Upper right and lower right are the two rough plates prepared with Nickel particles.

Figure. 6.62 Inconel plates used in the experiment
Figure 6.63 (a) Temperature distribution over the epoxy surface, Re = 60000

Figure 6.63 (b) Circumferentially averaged temperature as a function of r (cm), epoxy plate, Re = 60000
Figure 6.64 (a) Temperature distribution over the epoxy surface, Re =40000

Figure 6.64 (b) Circumferentially-averaged temperature as a function of r (cm), epoxy plate, Re=40000

\[ y = -0.04421 \times^3 + 0.96423 \times^2 + 4.85214 \times + 48.00828 \]
\[ R^2 = 0.99934 \]
Figure 6.65 (a) Temperature distribution on the epoxy surface, Re = 60000

Figure 6.65 (b) Circumferentially-averaged temperature as a function of r (cm), epoxy plate, Re = 60000
(a) Re=60000

(b) Re=40000
Figure 6.66 Comparison of local Nu distribution between smooth plate and epoxy plate

6.3.3 Results and discussion

Figure 6.63 to 6.65 show the temperature distribution on the epoxy plate with different jet Reynolds number. The data reduction procedure is the same as the smooth plate test to find out the heat flux distribution. Recall equation (5.5):

\[ h(r) = \frac{q_e^e(r)}{T_w(r) - T_j} \]  

(6.1)

The temperature drop across the thin epoxy layer is:

\[ \Delta T(r) = T_{ep}(r) - T_w(r) = \frac{q^n_e(r)\delta_{ep}}{k_{ep}} \]  

(6.2)
Where $T_{tp}(r)$ is the Inconel surface temperature measured by TSP. The estimated thermal resistance is an average value over the epoxy layer and it is defined as the ratio between the thickness ($\delta_{ep}$) and thermal conductivity ($k_{ep}$) of the epoxy layer. The thermal resistance of the epoxy layer is estimated such that the resultant heat transfer coefficient after temperature correction based on this estimation matches the heat transfer data of the smooth plate:

$$h_{ep}(r) = \frac{q''_{c}(r)}{T_{tp}(r) - \Delta T(r) - T_{j}} \approx h_{sm}(r) \tag{6.3}$$

Where $h_{ep}(r)$ is the heat transfer coefficient over the epoxy coated target plate; $h_{sm}(r)$ is the heat transfer coefficient over the smooth target surface that is already known in section 6.1 of this chapter. The temperature drop $\Delta T(r)$ is subtracted from the TSP measured temperature $T_{tp}(r)$ to find actual jet impingement surface temperature $T_{w}(r)$, knowing the local heat flux $q''_{c}(r)$ and the estimated thermal resistance of the epoxy layer ($\delta_{ep} / k_{ep}$). This corrected surface temperature is then used to calculate the impingement heat transfer coefficient $h_{ep}(r)$. After comparing $h_{ep}(r)$ and $h_{sm}(r)$, the thermal resistance of epoxy layer, $\delta_{ep} / k_{ep}$, is found to be $4.75 \times 10^{-4}$ m$^2$/kW. The corrected Nu distribution along the epoxy test plate is compared to the smooth plate Nu distribution in figure 6.66. The average differences in Nu between these two cases are 1.8%, 3.5%, and 5.1% for Re of 60000, 40000, and 20000, respectively.

The TSP images of full field temperature distribution and circumferentially averaged temperature distribution for rough plate I and II are shown in figure 6.67 to 6.72. Table 6.5 lists the power input to the Inconel plate for each test run.
The local and area averaged Nusselt numbers are plotted in figure 6.73 and 6.74. All the Nusselt numbers are obtained with the same correction procedure described earlier for the epoxy plate. The figure clearly shows that both local and average heat transfer can be significantly increased by the presence of roughness elements, which can disrupt the thin thermal boundary layer. The heat transfer enhancement increases with increasing jet Reynolds number and the enhancement is most pronounced near the stagnation region (smaller r/D). Table 6.6 gives the difference in percentage between the average Nusselt number of rough plates and epoxy plate, the latter represents the Nu obtained with smooth surface. The largest roughness effect is achieved at highest Reynolds number near stagnation region. For example, at Reynolds number of 60000, there is a 30.9% increase in the Nusselt number at r/D=1.5 for rough surface II, while at the same Reynolds number and same r/D, the increase is 23.4% for rough surface I. At larger r/D of 3.5, the increase drops to 16.3% and 9.4% for rough surface II and I, respectively. In general, the heat transfer of high roughness plate (rough-II) is greater than that of low roughness plate (rough-I), and this difference increases with increasing jet Reynolds number. For jet Reynolds number of 20000, the enhancement in average Nusselt number for both rough-I and rough-II plate are less than 10%, most of which falls under the experimental uncertainty. This implies that the roughness is ineffective for increasing heat transfer under the Reynolds number of 20000, allowing the surface to behave as if they were smooth. The effect of decreasing Reynolds number is to thicken the thermal boundary layer and thus to reduce the influence of roughness. At stagnation region, the thermal boundary layer has the minimum thickness along the entire impingement surface; hence the roughness effect is most significant near the stagnation zone.
Figure 6.67 (a) Temperature distribution over the rough-I plate, Re=60000

Figure 6.67 (b) Circumferentially-averaged temperature as a function of r, Rough-I plate, Re=60000

\[ y = -0.4094x^3 + 3.1375x^2 + 2.8598x + 44.595 \]
\[ R^2 = 0.9997 \]
Figure. 6.68 (a) Temperature distribution over the rough-I plate, Re=40000

\[ y = -0.9758x^3 + 6.6744x^2 - 1.6255x + 49.083 \]
\[ R^2 = 0.9997 \]

Figure 6.68 (b) Circumferentially-averaged temperature as a function of r, Rough-I plate, Re=40000
Figure 6.69 (a) Temperature distribution over the rough-I plate, Re=20000

Figure 6.69 (b) Circumferentially-averaged temperature as a function of r, Rough-I plate, Re=20000

\[ y = -0.6832x^3 + 5.1683x^2 - 0.2321x + 52.38 \]
\[ R^2 = 0.9998 \]
Figure 6.70 (a) Temperature distribution over the rough-II plate, Re=60000

Figure 6.70 (b) Circumferentially-averaged temperature as a function of r, Rough-II plate, Re=60000
Figure 6.71 (a) Temperature distribution over the rough-II plate, Re=40000

\[ y = -0.5009x^3 + 3.6069x^2 + 0.9074x + 47.834 \]

\[ R^2 = 0.9999 \]

Figure 6.71 (b) Circumferentially-averaged temperature as a function of r, Rough-II plate, Re=40000
Figure 6.72 (a) Temperature distribution over the rough-II plate, Re=20000

Figure 6.72 (b) Circumferentially-averaged temperature as a function of $r$, Rough-II plate, Re=20000

The equation is:

$$y = -0.3957x^3 + 3.1497x^2 + 2.0681x + 52.14$$

$R^2 = 0.9993$
Table 6.5 Power input for the roughness test

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<td>6.47E+07</td>
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<td>0.97</td>
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<td></td>
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<td>6.40E+07</td>
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Figure 6.73 Local Nu of rough surfaces and epoxy surface as a function of r/D, Z/D=5
Figure 6.74 Average Nu of rough surfaces and epoxy surface as a function of r/D, Z/D=5

Table 6.6 Difference in average Nu at various r/D between rough and epoxy plate

<table>
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<th>Diff @ r/D=2.5</th>
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<tr>
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<td>40000</td>
<td>7.8%</td>
<td>10.1%</td>
<td>16.1%</td>
</tr>
<tr>
<td>rough-I</td>
<td>60000</td>
<td>9.4%</td>
<td>15.0%</td>
<td>23.4%</td>
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<td>rough-II</td>
<td>20000</td>
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<td>9.2%</td>
<td>7.0%</td>
</tr>
<tr>
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<td>40000</td>
<td>12.6%</td>
<td>15.9%</td>
<td>17.3%</td>
</tr>
<tr>
<td>rough-II</td>
<td>60000</td>
<td>16.3%</td>
<td>25.6%</td>
<td>30.9%</td>
</tr>
</tbody>
</table>
6.4 Uncertainty analysis

Recall that heat transfer coefficient $h$ is calculated from equation (5.5)

$$h(r) = \frac{q_e(r)}{T_w(r) - T_j} - \frac{1}{r} \frac{\partial}{\partial r} (k_i r \frac{\partial T}{\partial r}) + q''_g - q''_{\text{cond}}$$

(5.5)

where \(q''_e\) is the effective surface heat flux. The uncertainty in heat transfer coefficient $h$ is calculated from:

$$U_h = \left[ \left( \frac{\partial h}{\partial q_e} U_{q_e} \right)^2 + \left( \frac{\partial h}{\partial q_{\text{cond}}} U_{q_{\text{cond}}} \right)^2 + \left( \frac{\partial h}{\partial T_w} U_{T_w} \right)^2 + \left( \frac{\partial h}{\partial T_j} U_{T_j} \right)^2 \right]^{0.5}$$

(6.1)

The lateral conduction term, \(\frac{1}{r} \frac{\partial}{\partial r} (k_i r \frac{\partial T}{\partial r})\), is generally less than 10% of the heat generation term \(q''_g\). Thus its uncertainty is neglected in this calculation. The uncertainty of \(q''_g\) is 3%, which is dominant in calculating the uncertainty of effective local heat flux \(q''_e\). The error in surface temperature measurement, \(U_{T_w}\), is 0.93 °C. The lowest value of \((T_w - T_j)\) is 15 °C, which corresponds to the highest error in heat transfer calculation. The uncertainty of thermocouple that was used to measure jet temperature \(T_j\) is 0.8 degree C. Consequently, according to the root-sum-square method, the maximum uncertainty in calculating heat transfer coefficient $h$ is 8.6%.
CHAPTER 7 CONCLUDING REMARKS AND RECOMMENDATIONS FOR FUTURE WORK

7.1 Concluding remarks

Over the last decade, PSP technique has been widely accepted within the aeronautics community. Comparatively, TSP technique, which turns out to be a convenient tool for surface temperature measurement, has received little attention by experimentalists in thermo-fluid area. Neither PSP nor TSP is ready to completely replace pressure taps and thermocouples so far. The major issues with PSP are the temperature dependence and the limited ability in testing small pressure change. Compared to PSP, TSP has temperature dependence only and the data processing is more straightforward. Although the high temperature sensitivity of TSP is only effective over a limited temperature range, TSP technique still has a high potential of application in most of the heat transfer studies.

Many strides have been made in this dissertation towards applying TSP and PSP technique in the study of a typical heat transfer subject. First, the selected TSP and PSP have been calibrated to high accuracy. The phenomenon of TSP degradation has been investigated in detail in order to reduce measurement error. Then flow and heat transfer characteristics of impinging air jet were studied and compared to the literature. Specific observations made in the course of this work are:

1. It is believed that the investigation of photodegradation made in this work is among the first of similar studies concerning temperature sensitive paint. The study was conducted from an engineering point of view for practical application purpose rather than trying to find out the
physical mechanism(s) behind the phenomenon. It is found that the TSP degradation rate is a strong function of the environment temperature and the optical power of the excitation light. A correlation was developed based on the observed fluorescence intensity change due to TSP degradation. By applying the correlation, the temperature measurement can be corrected to high accuracy, especially for degradation occurring at high temperature and strong lighting condition.

2. Pressure measurement across the jet impinged target surface using PSP is successful at high jet Reynolds number. The surface pressure distribution identifies the impinging zone along the target surface and it resembles the pattern of the velocity distribution of approaching jet. Due to small change between test pressure and reference pressure, high measurement accuracy is difficult to achieve at low jet Reynolds number test as PSP is an absolute pressure measurement technique.

3. The “second peak” in local Nu is clearly observed in both plate jet and tube jet experiment at small Z/D, thanks to the high resolution picture of the temperature field obtained with TSP. The Nusselt number of tube jet is generally higher than that of the plate jet, especially at closer spacing (smaller Z/D) and higher jet Reynolds number. This could be resulting from the stronger entrainment effect involved with tube jet, which has no blockage to prevent entraining surrounding air into the jet flow.

4. Area averaged Nusselt number of tube jet matches the empirical data better than that of the plate jet. The reason is believed to be the similarity in the jet nozzle configuration between the tube jet nozzle employed in this work and the one that has been used in the reference work.

5. Higher surface roughness of the target plate enhances the jet impingement heat transfer. The effect is most significant at stagnation region and it increases with jet Reynolds
number. The finding in this study can lead to an economical approach to enhance the local impingement heat transfer by artificially increase the roughness of the jet impinged target surface.

7.2 Recommendations for future work

7.2.1 TSP degradation

TSP degradation investigation was conducted on aluminum coupons. The degradation rate might also be dependent on the material that TSP is coated on. It was observed in another TSP application that the TSP has larger degradation rate at elevated temperature when it is painted on the surface of plastic material with low glass transition temperature. This must involve more complicated chemical reactions between the substrate material and the fluorescence sensor molecule. The mechanism behind this complex photo bleaching phenomenon is remaining to be uncovered. Therefore, care must be taken if it is intended to apply the correlation of this study to find out the TSP degradation rate associated with nonmetallic test model. It is emphasized here that effort should be made at the first place to avoid photodegradation in any TSP application. Only if the degradation is unavoidable should the quantification of degradation be exercised.

7.2.2 PSP application

Temperature dependence is the major source of error for PSP application. It would be ideal to apply PSP at isothermal test condition to reduce measurement error induced by the
temperature variation across the measurement surface. The Binary FIB PSP applied in the
current research has a limited effective temperature range to compensate the temperature
effect (5 to 50 degree C). However, the base coat of this dual probe paint is a good PSP with
low temperature sensitivity (see figure 4.8 a) and can be used at a wider temperature range. If
the temperature variation over the test surface can be controlled at a few degree C, this paint
can be used with reasonable accuracy by applying the calibration correlation obtained at the
temperature of test condition.

7.2.3 Jet impingement test

In order to explain the observed difference in heat transfer between the tube jet and
the plate jet, it is recommended to perform more measurement on the distribution of velocity
and turbulence intensity within the jet flow, especially near the impingement surface. The
reason for this recommendation is that the turbulence intensity plays a major role on local, as
well as averaged heat transfer. If a test scheme can be designed to measure the turbulence
intensity right above the jet impinged surface without disturbing the flow, more useful
information could be acquired to clarify the effect of jet nozzle used in jet impingement
experiment.

7.2.4 Surface roughness effect on impingement heat transfer

In the test of roughness effect it is found that there is minor difference in measured
heat transfer rate between the two rough plates. It is recommended to have more test plates
prepared with larger variation in surface roughness in order to develop a correlation to
quantify the roughness effect on impingement heat transfer. A correlation between normalized surface roughness and jet Reynolds number could be found first to define a “rough” surface. For example, if we are interested in an average Nusselt number that is averaged over r/D=3, then we can define a critical Reynolds number for different surface roughness condition such that the average Nu is 10% higher than that of the smooth surface. In other words, for certain surface roughness condition, only when the jet Reynolds number is larger than the critical Reynolds number, the surface can be considered rough and has enhanced heat transfer.

At the same time, it is desired to improve the procedure of creating the rough surface such that the surface roughness is more repeatable if it is prepared by different testers.
APPENDIX
LOCAL AND AVERAGE NUSSELT NUMBER CALCULATION (MATHCAD FILE)
Local and average Nusselt number calculation based on the temperature distribution and the power input to the foil heater

Volume heat generateion $q'''$ (W/m$^3$) $q := 15708923.08$

Inconel foil thickness $t$ (m) $t := 0.0005$

Jet total temperature $T_j$ (C) $T_j := 20.7$

Room temperature $Trm$ (C) $Trm := 21$

Thermal conductivity of Inconel $kf$ (W/m.k) $kf := 12$

Thermal conductivity of air $kair$ (W/m.k) $kair := 0.025$

Thickness of plexiglas target plate $tp$ (m): $tp := 0.0127$

Thermal conductivity of Plexiglas $kp$ (W/mk) $kp := 0.2$

Natural convection $htc$ $hnat$ (W/m2k): $hnat := 5$

Diameter of jet nozzle $D$ (m) $D := 0.0095$

Coefficients of 5th order polynomical of radial temperature distribution $A := \begin{pmatrix} 35.117 \\ 0.8245 \\ 6.3766 \\ -3.8173 \\ 1.0124 \\ -0.0958 \end{pmatrix}$

Radial distance $R$ (cm) $R := 0.2, 0.4.. 4.0$

Surface temperature $Tw$: $Tw(R) := A_0 + A_1 \cdot R + A_2 \cdot R^2 + A_3 \cdot R^3 + A_4 \cdot R^4 + A_5 \cdot R^5$

Conduction heat loss $qc$ (W/m2k): $qc(R) := \frac{(Tw(R) - Trm)}{\frac{tp}{kp} + \frac{1}{hnat}}$

Effective heat flux $qe$ (W/m2): $qe(R) := q \cdot t + t \cdot kf \cdot \left( \frac{A_1}{R} + 4 \cdot A_2 \cdot R + 9 \cdot A_3 \cdot R + 16 \cdot A_4 \cdot R^2 + 25 \cdot A_5 \cdot R^3 \right) \cdot 10000 - qc(R)$
Local Nusselt number $\text{Nu}(R)$ as a function of $R$

$$
\text{Nu}(R) := \frac{\text{qe}(R) \cdot D}{(\text{Tw}(R) - T_j) \cdot \text{kair}}
$$

Area averaged Nusselt number $\text{Nua}(R)$ as a function of $R$

$$
\text{Nua}(R) := \frac{\int_{0}^{R} \left[ q \cdot t + t \cdot k_f \left( \frac{A_1}{r} + 4A_2 + 9A_3 \cdot r + 16A_4 \cdot r^2 + 25A_5 \cdot r^3 \right) \cdot 10000 \right] \cdot D \cdot r \, dr}{R^2}
$$

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