Doctoral Thesis

In-situ measurement of blade heat transfer coefficients and gas recovery temperature

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IN-SITU MEASUREMENT OF BLADE HEAT TRANSFER COEFFICIENTS AND GAS RECOVERY TEMPERATURE

A dissertation submitted to the
Swiss Federal Institute of Technology
Zürich

for the degree of
Doctor of Technical Sciences

presented by
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Abstract

This thesis presents the results of a study for the measurement of in-engine heat transfer coefficients on cooled rotating blades in an operating gas turbine engine. These are the first successful in-situ measurements of the heat transfer coefficients in the harsh environment (high temperature, combusted gases) of an engine in operation. The heat transfer coefficient distributions along the midspan of the rotor blades of the low-pressure turbine in the heavy duty (288 MW) gas turbine engine were measured at different engine operating conditions.

In the measurement technique, blade surface temperatures and an upstream reference gas temperature are measured while the engine is subjected to a thermal transient. The temperatures on the surfaces of the rotating blades are measured by means of an infrared pyrometer, whose optical access is from the tip endwall of the upstream vanes. The variation in the reference gas temperature is measured using a rake of thermocouples positioned at the exit of the second combustor. The distribution of heat transfer coefficients is then determined from both the variation in blade surface temperatures and the variation in the upstream gas temperature. Heat transfer data along the midspan of the blades are presented in terms of Nusselt numbers for a set of TBC coated and uncoated blades, respectively. For the leading edge area of the blades, the measured heat transfer rates are compared with empirical correlations. These correlations show excellent agreement with the experimental data. The influence of Reynolds number and incidence angle on the heat transfer rates are demonstrated and discussed. The associated measurement errors in heat transfer coefficient are assessed using a Monte Carlo analysis and the mean error is found to be 15.8%. This mean error is slightly higher than those resulting from experiments in the benign laboratory environment. The accuracy of the measurements and the low relative cost of the hardware substantiate that the present measurement approach is well suited for in-situ
heat transfer measurements of large gas turbine engines.

Three-dimensional Navier-Stokes simulations of the rotor blade flow are used to predict the heat transfer coefficients on the isothermal blades. These CFD predictions are compared with the experimental data over the range of Reynolds numbers observed in the engine experiment.

A novel measurement approach is presented for the experimental determination of the local adiabatic gas recovery temperature distribution around the blades. The method is based on the distribution of both the measured surface temperatures and the heat transfer coefficients. A least-squares fitting procedure is used to equate the heat transfer rates due to convection and conduction at the blade surface. From the least-squares fit, the local adiabatic gas recovery temperature is derived. The measured recovery temperature distribution at different Reynolds numbers is compared with the predicted distribution using the result of the a through-flow solver. The mean measurement uncertainty in the recovery temperature is estimated to be 5.9%.

In the context of in-engine heat transfer measurements, results of thermal blade calibration work are presented. The thermal diffusivity of different blade materials is measured in a bench experiment. The method used to measure the thermal diffusivity is based on the thermal excitation of the sample surface using a pulsed laser beam and the subsequent measurement of the temperature response on the surface. The diffusivity data is then compared with data from the literature.

Experiments on TBC coated and uncoated blade samples are conducted to determine their surface emissivities. In the measurement method, a defined area on the sample surface is covered with a paint of known emissivity. A pyrometer is used to measure the radiosities of the painted and unpainted areas. From these comparative measurements, the emissivity of the sample surface is derived. A data reduction procedure is also developed to account for the amount of reflected radiation emitted from the measurement area.
Zusammenfassung


Mittels dreidimensionalen CFD Simulationen wird die Verteilung der Wär-
meübergangszahlen berechnet. Die Resultate der CFD Berechnungen werden mit den experimentellen Daten verglichen. Die Übereinstimmung zwischen Experiment und Simulation ist zufriedenstellend, obwohl die Wärmeübergangszahlen tendenziell etwas zu tief vorausgesagt werden.


Im Bezug auf die Wärmeübergangsmessungen in der Gasturbine werden die Resultate der thermischen Kalibrierung von Turbinenschaufeln präsentiert. In einem Laborexperiment wird die thermische Diffusivität von Metallproben gemessen. Mittels eines pulsierenden Laserstrahls wird die Oberfläche der Proben periodisch aufgeheizt. Die resultierende Temperaturverteilung an der Oberfläche wird mittels eines Pyrometers gemessen und daraus die thermische Diffusivität bestimmt.

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August 2007

Matthias Brunner
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Chapter 1

Introduction

1.1 Motivation

World energy resources and consumption have become a matter of ever more public interest over the past decades. The political, social and ecological issues regarding the use and supply of energy will most likely remain a vital concern for many years to come.

One of the main objectives in saving energy resources is to reduce energy consumption by increasing the efficiency of energy conversion and utilization. As the gas turbine is the most common device for power production and energy conversion, improvements of its efficiency would be a major contribution to energy resource savings.

One of the basic rules of gas turbines is that both the thermal efficiency and output increase with increasing firing temperature. Thus, in recent years, the demand for more efficient gas turbine engines has led to a steady increase in turbine inlet temperatures. In 2007, a turbine inlet temperature of 2250 K (3600°F) was reported [60] in a high-performance jet engine, a world first.

Sophisticated cooling techniques are employed in modern gas turbines, allowing the components to operate at temperatures well beyond the melting point of the components’ materials, see Figure 1.1. The cooling techniques include internal convection cooling, film cooling and the application of thermal barrier coating systems (TBC). In order to benefit from the increased turbine firing temperatures, efficient use of cooling air is required. Thus, a
Introduction

Figure 1.1: Variation of turbine inlet temperature and development of cooling techniques over recent years [59].

good understanding of the heat transfer distribution around the blade profile is vital.

The present thesis responds to the attention given by the gas turbine community to the measurement and understanding of the heat transfer distribution on turbine blades. Until now, the knowledge of blade heat transfer has derived from measurements in linear cascades and dedicated rotating test facilities. This thesis describes a unique method developed for the measurement of both the convective heat transfer coefficients and the local adiabatic gas recovery temperature on the rotor blades of an operating gas turbine. These are the first successful measurements that are conducted in the harsh environment of a gas turbine engine in operation.

The severe environment inside an operating gas turbine engine sets high standards on the measurement equipment. The difficult conditions consist of high temperatures and pressures as well as particles in the exhaust gas. For these reasons, heat transfer researchers usually turn to the more benign measurement conditions that prevail in cascades and rotating rigs. However, many real-engine effects are difficult to reproduce in the laboratory environment.
These in-engine effects are temperature, pressure and turbulence profiles, flow unsteadiness, leakage and coolant flows. All these influences affect blade heat transfer rates. Also, when measuring in-engine, no approximations on the effects of geometry or scale need to be made. Thus, a successful in-engine measurement of heat transfer coefficients designates a significant step forward in experimental heat transfer research.

The proposed measurement method has proved to be an effective tool in the development of more powerful and more efficient gas turbine engines. Thus, the proposed measurement techniques could make a significant contribution to energy resource savings in the near future.

1.2 Previous Work

The current section gives an overview of the published work relevant to the present study. As the present work describes the first in-engine measurement of heat transfer coefficients, the literature review concentrates on the experimental efforts of heat transfer measurements conducted in research facilities. Given the vast number of publications in this field of research, only a selection of studies are reviewed in Section 1.2.1.

Section 1.2.2 identifies and describes several heat transfer mechanisms pertinent to turbine blade heat transfer.

At various stages within this project, surface temperatures are measured using infrared pyrometry. Section 1.2.3 reviews the literature pertaining to pyrometric surface temperature measurements. It also gives an introduction to the measurement method and presents applications to gas turbine engines.

1.2.1 Blade Heat Transfer Experiments

To date, many experimental studies on blade heat transfer from various test rigs have been published. Since the early years, heat transfer experiments were conducted in instrumented linear or annular cascade wind tunnels. Rotating rigs are increasingly used for aerodynamic and heat transfer measurements. Heat transfer measurements are generally conducted in short-duration (300 to
1000 ms) experiments. In long-duration tests, aerodynamical measurements are performed.

**Cascade experiments.** The first heat transfer experiments conducted in linear cascades date back more than 50 years. Some of the first researchers to present blade heat transfer data were Wilson and Pope [108]. They measured heat transfer from cascade experiments for different Reynolds numbers and incident angles. They calculated the heat transfer distribution from the knowledge of the power input to electrical heating elements on the blade surface and from the surface and air temperature measurements. Turner [101] measured the variation of the local heat transfer coefficient around gas turbine blades in a linear cascade. Turner’s measurement method was based on the determination of the metal surface temperature of the blade as well as the coolant temperature and the heat transfer coefficient in the internal cooling passages. Using a solution of Laplace’s equation, he then determined the temperature gradient at the blade surface, which relates to the heat transfer coefficient. More recently, Kirsten et al. [58] determined the heat transfer rates to a nozzle guide vane in a linear cascade and compared the data with predicted distributions from a boundary layer code. In the transient technique that was employed, thin film gages were used to measure the surface temperature history on the cooled vane. The one-dimensional transient heat conduction equation was then used to calculate the heat transfer rates. Bunker [22] used embedded thermocouples and thin-foil surface heaters to measure airfoil heat transfer distributions in a transonic linear vane cascade. In the measurement program, a range of Reynolds numbers and inlet turbulence levels was covered to study the effects of surface roughness on external heat transfer. Arts et al. [9] measured heat transfer coefficients on a high pressure gas turbine blade profile in a two-dimensional linear cascade arrangement. Thin film gages were used to measure the local time dependent surface temperature history. The heat flux was then obtained from the semi-infinite solution of the unsteady conduction equation. Boyle et al. [19] measured turbine vane heat transfer distributions in a three-vane linear cascade. An infrared camera was used to measure the surface temperature distribution on an electrically heated vane. The heat flux was then determined from the electrical power required to heat the vane. Giel et al. [47] reported measurements of heat transfer coefficients on power generation turbine blades in a transonic linear cascade. Heat flux data were obtained by a steady-state technique using a thin-foil heater wrapped around a low thermal conductivity blade. The blade surface temperatures
1.2 Previous Work

were measured using calibrated liquid crystals. Nix et al. [77] studied the effect of free stream turbulence on turbine blade heat transfer. They used heat flux micro-sensors and K-type thermocouples embedded in the blade surface to determine the heat transfer rates to the blade.

Rotating rig experiments. Dunn [33] measured the time-averaged blade heat flux distribution on a full-stage rotating turbine in a short-duration, shock-tunnel facility at Calspan [36]. The heat transfer distributions were measured by means of thin-film gages. The blade results were compared with predictions from flat plate correlations and from CFD. Abhari et al. [3] conducted heat transfer experiments in a short-duration (0.3 s test time) rig at the MIT blowdown turbine facility [38]. The heat flux distribution was obtained using thin film heat flux gages, which consisted of two nickel temperature transducers. Time-resolved heat transfer distributions at the blade midspan were presented and compared with numerical calculations. Blair [15] examined the surface heat flux in a large-scale turbine rotor rig (United Technologies Research Center) at ambient temperature. Local heat transfer coefficients were determined using thermocouples to measure the temperature difference between the heated blade surface and the freestream. Blair reported that three-dimensional effects (secondary flows, tip-leakage vortices) produced locally enhanced heat transfer rates that could only be identified in full-stage models with rotation. Moss et al. [69] studied the influence of rotation on the blade surface heat transfer in the Oxford Rotor Facility [5]. Thin-film gages were used to measure the heat flux. They concluded that the rotor data did not differ from the cascade measurements on the pressure surface. On the suction surface, however, the rotor heat transfer rates were between the laminar and turbulent levels from the cascade experiments. Didier et al. [32] determined the convective heat transfer coefficients around the rotor of a transonic turbine stage in the compression tube turbine test facility of the von Karman Institute [89]. Blade surface temperature histories during a blowdown test were measured by means of thin film gages. From the temperature histories, the heat flux was derived by solving the one-dimensional unsteady conduction equation. In Didier’s experiments, the presence of periodic fluctuations due to shocks and wakes resulted in high levels of time-averaged heat transfer at the leading edge. However, only unrealistically high turbulence intensities could cause such levels of heat transfer in a linear cascade with isotropic turbulence.

An active field of research is the heat transfer to the tip of turbine blades. In-
teresting work on this subject have been published, among others, by Metzger et al. [67], Ameri et. al. [8] and Bunker [23]. In the present work, heat transfer rates along the midspan of turbine blades are measured. Thus, references to studies on tip heat transfer are limited to those mentioned above.

A summary on the available cascade and rotating rig heat transfer experiments was presented by Simoneau and Simon [90]. In his 2001 IGTI scholar lecture [34], Dunn discussed both experimental and computational research activities with respect to convective heat transfer in axial flow turbines. They both reported that time averaged and time resolved three-dimensional heat transfer data are routinely measured in full-stage research facilities but to date, have not been measured in full-scale operational facilities.

Further references to past work are also indicated in Section 1.2.2 below, where influences affecting the heat transfer rates to gas turbine blades are introduced.

\subsection{1.2.2 Heat Transfer Mechanisms}

From the published heat transfer experiments, several different heat transfer mechanisms and factors affecting the convective blade heat transfer were identified. Among these mechanisms are: Laminar, transitional and turbulent flow heat transfer; stagnation point heat transfer; separation and reattachment; effects due to surface roughness, free stream turbulence, pressure gradient and surface curvature.

The different boundary layer regions on a typical turbine blade are schematically shown in Figure 1.2. Heat transfer rates are closely linked to the state of the boundary layer. In a laminar boundary layer, the heat transfer is lower than in a turbulent boundary layer. Main stream flow characteristics not only influence the heat transfer within the laminar and turbulent region but also affect the location of the transition point. In the stagnation region, high heat transfer rates are observed. Flow separation causes low values of heat transfer, whereas at the point of reattachment, the heat transfer is increased.

\textbf{Effect of Reynolds number.} The general effect of Reynolds number on blade heat transfer is well understood. Overall heat transfer rates increase with an increase in Reynolds number. Many empirical correlations show that for turbulent boundary layer flow, the Nusselt number scales with the
Reynolds number to the 0.8 power and to the 0.5 power for laminar flow (Haldeman et al. [48, 49]). Blair [15] also demonstrated this behavior on large-scale rotating blades. He reported that the transition region on the blade suction side becomes smaller and occurs earlier with increasing Reynolds numbers. This is in agreement with the observations of Schiele et al. [85] and Suslov et al. [96]. They described an upstream shift of the laminar-turbulent transition on the suction surface and a reduction of transition length when the Reynolds number was increased.

Effect of free stream turbulence. According to the measurements of combustor turbulence spectra of Moss [70], turbulence levels well over 10% are encountered in gas turbines. High free stream turbulence levels are generated in the combustion chamber and progress towards the downstream vane and blade rows. Huffman [51] made detailed measurements in turbulent boundary layers and found that the free stream turbulence affected the turbulent transport in the boundary layer. This caused an increase in wall friction and consequently, an increase in heat transfer. Simonich [91] reported a 5% increase in Stanton number for every 1% increase in turbulence intensity in flat plate turbulent boundary layers. Moss [71] observed Nusselt number enhancements of up to 40% due to turbulence intensities of 16% in flat plate experiments. However, Moss found that the increase in heat transfer rates were not only dependent on the turbulence intensity, but also on the turbulence length scale of the main flow. The general effect of high free stream turbulence levels on
turbine airfoils is to enhance the overall heat transfer rates along the leading edge and on the pressure surface (Radomsky and Thole [80]). The most visible effect, according to Didier et al. [32], was the displacement of the onset of transition on the suction surface towards the leading edge of the blade. This effect was already demonstrated by Blair [14] on zero pressure gradient, fully turbulent boundary layer flows.

**Effect of pressure gradient.** Along most of the surface of a turbine blade, a favorable pressure gradient exists. Thus, the main flow is accelerated, which affects the heat transfer rates. Accelerated flows are characterized by the acceleration parameter $K$, that is defined by Equation 1.1 [55].

$$K = \frac{\nu}{u_\infty^2} \frac{du_\infty}{dx}$$

For turbulent boundary layers, Kays et al. [56] noted that for $K > 1.0 \times 10^{-6}$, the favorable pressure gradient caused the viscous sublayer to thicken. This resulted in an increase in heat-transfer resistance in the sublayer and a decrease in Nusselt number. For $K > 3.5 \times 10^{-6}$ the turbulent boundary layer may relaminarize, thus also causing a reduction in heat transfer. Rued and Wittig [83] found that in flat plate measurements, flow acceleration delays transition, which leads to a reduction in Nusselt number in the downstream part of the plate. This result was confirmed by Volino and Simon [105] in measurements from heated boundary layers along a concave test wall. The experimental conditions were chosen to simulate the downstream part of the pressure surface of a turbine blade. They reported lower heat transfer levels in the accelerated than in the unaccelerated flow.

In the laminar boundary layer, the opposite effect is observed. Flow acceleration leads to an increase in heat transfer due to a reduction in boundary layer thickness, as reported by Rued and Wittig [83].

**Effect of surface roughness.** Surface roughness in fluid dynamics is commonly described by the sand grain roughness $k_s$ that was introduced by Schlichting [86]. Three different regimes are distinguished according to the roughness Reynolds number $Re_{k_s}$:
Smooth \( Re_{ks} < 5 \) : The surface behaves aerodynamically smooth.

Transitional \( 5 < Re_{ks} < 70 \) : There are noticeable effects due to roughness.

Fully rough \( 70 < Re_{ks} \) : The friction coefficient \( c_f \) is independent of Reynolds number.

The effect of roughness is to destabilize the viscous sublayer and to reduce it as \( Re_{ks} \) increases. When \( Re_{ks} \) rises to a value of about 70, the viscous sublayer disappears and the friction coefficient (and the heat transfer) become independent of the Reynolds number.

Many researchers measured the effect of surface roughness on heat transfer of turbine airfoils (e.g. Abuaf [4], Blair [15], Bons [16, 17], Boyle [19], Stripf [95]). Generally they showed increased heat transfer rates with increasing surface roughness. The heat transfer augmentation is more pronounced at high Reynolds numbers. Increased surface roughness also affects the transition from laminar to turbulent flow by causing the transition to occur at lower flow Reynolds numbers.

Bons et al. [18] performed contact stylus measurements of surface roughness on approximately 100 turbine components from four land-based turbine manufacturers. They concluded that the type of roughness varied considerably with the location inside the machine, the place of operation, coatings and cooling configuration.

**Stagnation point.** Very high heat transfer rates can be observed at the leading edge of the blade. There are thin viscous layers at the stagnation point and near the leading edge. Both the velocity and temperature gradients are high. At the stagnation point, the values of the heat transfer coefficients are commonly estimated from results of cylinders in crossflow. According to Kays [55], the stagnation point heat transfer of a circular cylinder in crossflow is given by Equation 1.2.

\[
Nu_R = 0.81 \cdot Re_R^{1/2} Pr^{0.4} 
\]  

(1.2)
Hence with increasing radius, the heat transfer coefficient decreases. The body shape of the leading edge and the free stream turbulence intensity also influence the heat transfer rates in the stagnation region. Lowery and Vachon [61] and Smith and Kuethe [92] devised empirical correlations for the effect of freestream turbulence on the local heat transfer rate at the stagnation point of circular cylinders. Among the recent publications on blade leading edge heat transfer are those by Giel [46] and Van Fossen [102, 103].

1.2.3 Infrared Pyrometry

At various stages within this project, surface temperatures are measured by means of infrared pyrometry. This section gives a short introduction to the principles of radiation thermometry, followed by a description of the previous work and challenges in this field of research.

**Fundamentals of radiation thermometry.** At temperatures above 0 K, every object emits thermal radiation. The rate of emission of radiation from a blackbody is described by the spectral radiance, which is given by Equation 1.3.

\[
L_{\lambda,b}(T) = \frac{C_1}{\lambda^5} \left[ \exp\left(\frac{C_2}{\lambda T}\right) - 1 \right]^{-1},
\]

where \( C_1 \) and \( C_2 \) are constants, \( \lambda \) is the wavelength and \( T \) the temperature. Equation 1.3 is called Planck’s law. Thermal radiation is emitted in the wavelength range from approximately 0.1 to 1000 \( \mu \text{m} \). This range contains the wavelength band that is visible to the human eye (0.38 - 0.76 \( \mu \text{m} \)). Thus, we can see thermal radiation emitted from a body above a temperature of approximately 775 K.

Depending on the wavelength and temperature, a maximum to Planck’s law is found according to Equation 1.4.

\[
\lambda_{\text{max}} T = 2898 \ [\mu \text{mK}]
\]

Figure 1.3 shows the variation of radiant flux with wavelength for a blackbody at different temperatures. The region that is visible to the human eye is
1.2 Previous Work

Figure 1.3: Variation of radiant flux with wavelength and temperature. Shaded area indicates visible region (0.38 - 0.76 μm).

indicated by the hatched area. The sun emits a spectrum similar to that of a blackbody at 5780 K.

If the term $e^{C_2/λT}$ is much larger than 1, Planck’s law reduces to

$$L_{λ,b}(T) = \frac{C_1}{\lambda^5} [e^{C_2/λT}]^{-1}$$  \hspace{1cm} (1.5)

which is Wien’s formula. In the wavelength region that is considered in this work ($λ \approx 1.5$ μm), Wien’s approximation to Planck’s law is accurate to within 1% up to a temperature of 2000 K.

The total radiance of a blackbody is found by integrating Equation 1.3 with respect to $λ$, to obtain the Stefan-Boltzman law:

$$L_b = \frac{σT^4}{π}$$  \hspace{1cm} (1.6)

where $σ$ is the Stefan-Boltzman constant. From the 4th power law in Equation 1.6 it is evident that the total radiance increases rapidly with temperature. This is one of the reasons for the high measurement accuracy of radiation thermometry at elevated temperatures.
Real surfaces have a spectral radiance which is less than that of a blackbody at the same temperature. The surface emissivity $\varepsilon$ is used to characterize the radiant behavior of real surfaces and it is defined by Equation 1.7.

$$\varepsilon = \frac{L_\lambda}{L_{\lambda,b}}$$  \hspace{1cm} (1.7)

The emissivity depends on the physical condition of the surface, the wavelength and the temperature. Generally, the uncertainty in surface emissivity is the single most important factor affecting the measurement accuracy of pyrometers.

**Pyrometer working principle.** Pyrometry is a non-contact means of temperature measurement that is based on the thermal radiation emitted by the measurement object. The measurement instrument operates by collecting thermal radiation from a defined surface area and optically transferring it to a detector. The detector produces an electrical signal that is proportional to the radiant power of the surface. From the radiant power, the surface temperature is determined using Planck’s law and the known emissivity of the surface.

A number of advantages of pyrometry over traditional measurement techniques were identified [57]:

- No physical contact with the target surface resulting in minimum sensor interference. This is especially advantageous for turbine blades, since their surface temperature strongly depends on the flow field;

- Excellent temperature resolution due to the rapid increase of the radiance with temperature (Stefan-Boltzman law);

- Fast response capability due to the absence of any thermal inertia;

- Capability of non-intrusiveness into the gas stream.

**Turbine applications.** Different researchers have undertaken surface temperature measurements on rotating turbine blades. DeLucia et al. [28] measured blade surface temperatures along the span at mid-chord in a heavy duty gas turbine by means of a single point pyrometer (Land Infrared, 100 kHz). Different measurement errors were identified and the experimental data was fitted to an analytical model of the measured temperature pattern. In an
1.2 Previous Work

earlier work, DeLucia and Masotti [30] used a pyrometer for the surface temperature measurement on a turbine blade model and developed a computer program for the reduction of reflection errors. Eggert et al. [37] developed a fast response (500 kHz), short wavelength (0.9 µm) pyrometer system for turbine applications. It was tested on the second stage turbine blades of a heavy duty gas turbine of 180 MW power. Frank et al. [42, 41] used the same pyrometer system to measure surface temperatures on the first and second stage blades. In their experimental setup, the pyrometer extended into the gas stream and used a mirror to get optical access to the blades. It was mounted on a traversing system, which allowed a large portion of the blade span on the pressure surface to be measured. In the same test engine, Markham et al. [65] used short (0.9 µm) and long (10.5 and 11.8 µm) wavelength versions of the pyrometer to study the influence of the thermal barrier coating (TBC) on the measurement result. They concluded that the short wavelength pyrometer showed good agreement with a calculated temperature distribution. The long wavelength version measured slightly higher surface temperatures.

In the present work, short wavelength (1.5 µm) infrared pyrometers are used to measure surface temperatures on the pressure and the suction surface of the low-pressure turbine blade. Further technical details on the pyrometry system used are given in Section 2.1.3.

Other applications. Today radiation thermometry or thermal imaging is not only used for the measurement of temperatures in gas turbine engines, but for a variety of different applications. The areas of application include, but are not limited to, the steel and aluminum industry; building energy management; non-destructive testing; medical applications.

- **Steel and aluminum industry.** In the steel and aluminum industry, radiation thermometers are commonly used for improved quality control and increased productivity and energy efficiency. Due to the application of radiation thermometers, coupled with computer control, many steel mills increased rolling steel rates and improved the efficiencies of steel furnaces [31].

- **Building energy management.** Specialized companies provide infrared scans to find weaknesses or leakages in the insulation of buildings. For this purpose, infrared cameras are used, which usually feature an array of detectors to produce two-dimensional pictures. Another application
is the quantification of the extent of roof moisture (water) that is in the roof system [94].

- **Non-destructive testing.** Non-destructive testing with infrared thermography is performed in two basic ways: pulsed thermography or modulated lock-in thermography [64, 66]. Both techniques are used to reveal material inhomogeneities and defects of different materials such as metals, plastics or composites. This technology is also called Thermal Wave Imaging.

- **Medical diagnostics.** Infrared thermal imaging of the skin is used to monitor the temperature distribution of human skin. Abnormalities such as inflammations and infections cause localized temperature increases that show as hot spots [53]. The same approach is also used to screen travellers for likelihood of fever. Optical imaging technologies are applied for non-invasive imaging of intact tissues using fluorescence [78]. In these applications, the near-infrared wavelength band is chosen due to the minimal photon absorption of water and hemoglobin within this wavelength band.

### 1.3 Research Objectives

In the past, practical validation of heat transfer distributions were performed in cascade wind tunnels or rotating test facilities. The experimental costs for achieving engine representative conditions in these test facilities are considerable. The majority of the experimental data were also acquired near room temperature. Therefore, discrepancies between measured and predicted heat transfer coefficients could possibly arise due to either inadequate experimental conditions or insufficient prediction capabilities.

The work described in this thesis aims at developing an experimental technique that would allow the heat transfer coefficients to be measured in-situ on an operating gas turbine blade. To the author’s knowledge, this would be the first time that real engine heat transfer data could be measured. The proposed in-engine measurement technique for blade heat transfer coefficients could prove to be an invaluable tool in the development process of modern gas turbines. The availability of measured heat transfer data will allow the adjustment of cooling models and the amendment of cooling specifications.
In addition, an experimental method was devised for the in-situ measurement of the local adiabatic gas recovery temperature. The availability of real engine data also enables a more exact and extensive comparisons with existing heat transfer models and CFD predictions.

Thus, the key contribution of the present work is identified to be the demonstration of an accurate in-engine measurement of blade heat transfer coefficients by means of infrared pyrometry. It is shown that for these measurements, an accurate knowledge of the blade emissivity is not required. The effects of Reynolds number and incidence on heat transfer that are known from lab experiments, are identified on operating blades.
1.4 Thesis Outline

Chapter 1 of this thesis presented an introduction to turbine blade heat transfer. Previous research work was summarized with respect to blade heat transfer experiments in cascade wind tunnels and rotating rigs. The most relevant heat transfer mechanisms in the blade situation were described, as well as the basics of infrared pyrometry.

The measurement of in-engine heat transfer coefficients are presented in Chapter 2. The measurements were conducted in an ALSTOM GT26 heavy duty gas turbine engine. In this chapter, the experimental setup and the instrumentation used are described. The data reduction procedure is explained, followed by the description of the mean line analysis. The experimental results are given in terms of Nusselt numbers. Finally, the uncertainty analyses in both the surface temperature measurement and the measurement in heat transfer coefficients are presented.

Chapter 3 describes the three-dimensional Navier-Stokes simulations that are conducted to predict the heat transfer rates on the blade. The predicted heat transfer distributions are compared with the experimental data.

A novel method for the measurement of the local adiabatic gas recovery temperature is introduced in Chapter 3.7. The theoretical background on the concept of recovery temperature is presented and the data reduction process for the novel method is explained. The measured data is compared with predicted values from a through-flow solver and with results from the CFD analysis.

Chapters 5 and 6 are dedicated to the calibration work that is necessary for accurate in-engine heat transfer measurements. Chapter 5 presents the measurements of thermal diffusivity on samples of different blade materials. Chapter 6 describes the measurements of surface emissivity on TBC coated and uncoated blade samples. Contrary to the measurements of heat transfer coefficient and gas recovery temperature, these experimental calibration programs were conducted in laboratory bench experiments.

The conclusions of the present work and some suggestions for further research are given in Chapter 7.
Chapter 2

Measurement of Heat Transfer Coefficients

The present chapter describes the experimental setup and the data reduction method applied in the measurement of in-engine heat transfer coefficients. The experiments are conducted in the ALSTOM GT26 gas turbine engine, which is described in detail (Section 2.1.1). Section 2.1.2 describes the experimental program and the engine operation for the generation of temperature transients. The blade surface temperatures are measured by infrared pyrometry. These measurements are presented in Section 2.1.3. For the measurement of the reference gas temperature, thermocouples are used. The determination of their time constant is described in Section 2.1.4.

Section 2.2.1 demonstrates the generation of surface temperature-time transients from the pyrometry data for specific locations on the blade. For that part of the data reduction, a key phaser signal is used. The signal conditioning of the key phaser signal is presented in Section 2.2.2. The data reduction method of the reference gas temperature and the data synchronization of gas and surface temperatures are described in Sections 2.2.3 and 2.2.4, respectively. From the surface temperature time-transients and the reference gas temperature histories, the heat transfer coefficients on the blades are determined using a one-dimensional heat conduction model. This data reduction procedure is presented in Section 2.2.5.

A mean line analysis is conducted in order to determine the Reynolds numbers and incidence angles at different flow conditions. This mean line analysis is
presented in Section 2.3.

Finally, Section 2.4 shows the experimental results in terms of Nusselt numbers and compares the data with empirical correlations. The associated measurement errors for the surface temperature and the heat transfer measurements are assessed in Section 2.5.

2.1 Experimental Setup and Instrumentation

2.1.1 Test Engine

The in-situ measurements described in this work are made on an ALSTOM GT26 test gas turbine. The GT26 has a design output power of 288 MW. Its primary components include: A 22-stage axial compressor, a one-stage high-pressure turbine (HPT) and a four-stage low-pressure turbine (LPT), Figure 2.1. The engine has a single shaft and is equipped with sequential combustors for high efficiency and reduced environmental impact. The sequential combustion system has a dual-fuel capability; for the present study, gas is used as a fuel. The measurements of the heat transfer coefficients described in this present work are taken from second stage LPT rotor blades. In the test engine, this blade row has both TBC coated and uncoated blades. In the commercial version of the GT26, all the blades on the second LPT rotor are TBC coated. The measurements are conducted on two contiguous blocks of 10 coated and uncoated blades, respectively.

The second stage low-pressure turbine blades are internally cooled and have a half shroud. The midspan characteristics of this blade at approximate design load are given in Table 2.1. The Reynolds number is based on the blade chord at midspan and flow conditions at the trailing edge station. The Biot numbers of the coated and uncoated blades, based on the midspan blade wall thickness, are 0.85 and 0.4, respectively. These Biot numbers suggest that the resistance to conduction within the blade wall and the resistance to convection across the fluid boundary layer are of similar magnitude. Thus, both convection and conduction need to be taken into account in the data reduction method.

The method of determining heat transfer coefficients described in this work is based on the measurement of changes in both the blade surface temperature
2.1 Experimental Setup and Instrumentation

Figure 2.1: ALSTOM GT26 sequential combustion system with one-stage high and four-stage low pressure turbine [79].

and the reference gas temperature. The reference gas temperature is measured upstream of the second blade row, at the exit of the second combustor (SEV). A key phaser signal recorded from the single shaft is used for the data analysis. A schematic of the experimental setup for the measurement of heat transfer coefficients (HTC) is shown in Figure 2.2.
Table 2.1: Midspan characteristics of the second low-pressure turbine blade.

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Blade chord, C</td>
<td>0.14 m</td>
</tr>
<tr>
<td>Reynolds number, Re_C</td>
<td>$2.4 \times 10^6$</td>
</tr>
<tr>
<td>Mach number, M</td>
<td>0.85</td>
</tr>
<tr>
<td>Relative flow angle</td>
<td>-62°</td>
</tr>
<tr>
<td>Turning</td>
<td>90°</td>
</tr>
</tbody>
</table>

### 2.1.2 Engine Operation

The surface and gas temperature data used in this study are acquired as a part of a controls test program for the GT26 engine. This test program consists of different unit load steps at various initial load settings. Figure 2.3 shows the engine load of the entire test block versus time. The white circles indicate the different load cases considered in this study. Table 2.2 shows the entire test matrix. According to the test program, various unit load steps and load gradients are applied. (They differ by the step size of the load change as well as the applied load gradient.) Figure 2.4 shows a typical standard load profile. The corresponding variations in gas and surface temperature are shown in Figure 2.5. The reference gas temperature history in this figure is not corrected for the effect of thermocouple dynamics. Power changes of high load gradients are used to determine the heat transfer coefficients. The high load gradients cause a fast gas temperature change, resulting in a well-defined temperature response on the blade surface. Rather than a step in gas temperature, a temperature oscillation is observed.

The heat transfer coefficients are measured by subjecting the blades to a thermal transient. The thermal transient is generated by imposing an abrupt load change to the engine. Since the engine is connected to the grid, the rotational speed remains constant as the load changes. Increased power is achieved by increasing the massflow; in order to do this, the compressor IGVs are restaggered. The restaggering changes the pressure in the combustion chambers and the blade Reynolds number. In the test matrix, the initial load setting to which a unit load step is applied, is specified (Table 2.2). Also shown in Table 2.2 are the normalized Reynolds numbers; the Reynolds number at an engine load of 275 MW is used for normalization. The flow
incidence angle, relative to the 275 MW load case, is also indicated in the test matrix. These incidence angles are determined from a mean line analysis. Due to the limited information on the test engine, several assumptions are necessary to complete the mean line analysis. For the upstream stage, the exit flow angle and the stage pressure ratio (total to total) are assumed to be
constant. Also, fixed capacities are assumed for both the second rotor row and the downstream stator row. The relative total inlet temperature of the second blade row is calculated from the measured SEV exit temperature using a correlation from an ALSTOM through flow code. The mean line analysis is described in Section 2.3.
Table 2.2: Some characteristics of the engine test cases.

<table>
<thead>
<tr>
<th>Case</th>
<th>P (MW)</th>
<th>Re</th>
<th>α (deg)</th>
<th>Case</th>
<th>P (MW)</th>
<th>Re</th>
<th>α (deg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>275</td>
<td>1.00</td>
<td>0.0</td>
<td>10</td>
<td>140</td>
<td>0.68</td>
<td>-2.9</td>
</tr>
<tr>
<td>2</td>
<td>245</td>
<td>0.88</td>
<td>-0.2</td>
<td>11</td>
<td>130</td>
<td>0.67</td>
<td>-2.9</td>
</tr>
<tr>
<td>3</td>
<td>220</td>
<td>0.81</td>
<td>-0.5</td>
<td>12</td>
<td>120</td>
<td>0.65</td>
<td>-3.9</td>
</tr>
<tr>
<td>4</td>
<td>215</td>
<td>0.80</td>
<td>-0.5</td>
<td>13</td>
<td>100</td>
<td>0.62</td>
<td>-4.3</td>
</tr>
<tr>
<td>5</td>
<td>200</td>
<td>0.77</td>
<td>-0.8</td>
<td>14</td>
<td>70</td>
<td>0.55</td>
<td>-5.6</td>
</tr>
<tr>
<td>6</td>
<td>200</td>
<td>0.77</td>
<td>-0.7</td>
<td>15</td>
<td>65</td>
<td>0.54</td>
<td>-5.7</td>
</tr>
<tr>
<td>7</td>
<td>170</td>
<td>0.72</td>
<td>-1.6</td>
<td>16</td>
<td>50</td>
<td>0.57</td>
<td>-8.0</td>
</tr>
<tr>
<td>8</td>
<td>170</td>
<td>0.72</td>
<td>-1.7</td>
<td>17</td>
<td>35</td>
<td>0.60</td>
<td>-10.9</td>
</tr>
<tr>
<td>9</td>
<td>140</td>
<td>0.68</td>
<td>-2.9</td>
<td>18</td>
<td>35</td>
<td>0.61</td>
<td>-11.0</td>
</tr>
</tbody>
</table>

2.1.3 Surface Temperature Measurement

The blade surface temperature is measured by means of a single-point infrared pyrometry system. The pyrometer system transduces the radiant power from a defined surface area into an electrical signal. From this signal, the surface temperature is deduced using Planck’s law according to Equation 2.1.

\[ I = E_s \int_{\lambda_1}^{\lambda_2} \frac{C_1}{\lambda^5 \left[ \exp \left( \frac{-C_2}{\lambda} \right) - 1 \right]} d\lambda. \quad (2.1) \]

Here, \( E_s \) is an instrument constant, \( C_1 \) and \( C_2 \) are Planck’s constants and \( \lambda_1 \) and \( \lambda_2 \) describe the wavelength working bandwidth of the pyrometer. A short wavelength pyrometry system developed by ALSTOM is used in this work. Some salient specifications of the system are summarized in Table 2.3.

Optical fibers are introduced through the turbine casing to provide access to the blade surface. A sapphire lens of 1.4 mm diameter at the tip of the optical fiber collects the infrared radiation. The lens has a numerical aperture of 0.05, which corresponds to a cone half-angle of 2.9° according to Equation 2.2.
Table 2.3: Specification of the pyrometry system.

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Acquisition frequency</td>
<td>100 kHz</td>
</tr>
<tr>
<td>Temperature resolution</td>
<td>0.1 K</td>
</tr>
<tr>
<td>Temperature accuracy</td>
<td>±2 K at 1000 °C</td>
</tr>
<tr>
<td>Temperature range</td>
<td>350 – 1200 °C</td>
</tr>
<tr>
<td>Numerical aperture</td>
<td>0.05</td>
</tr>
<tr>
<td>Spatial resolution (in-engine)</td>
<td>3 – 4 mm</td>
</tr>
<tr>
<td>Typical spot size (in-engine)</td>
<td>3 – 8 mm</td>
</tr>
<tr>
<td>Wavelength range</td>
<td>1.55 ± 0.1 μm</td>
</tr>
<tr>
<td>Detector type</td>
<td>InGaAs</td>
</tr>
</tbody>
</table>

\[ NA = n \sin \theta, \tag{2.2} \]

where \( NA \) is the numerical aperture, \( \theta \) the cone half-angle and \( n \) the index of refraction (1.0 for air). The use of a pair of pyrometers allows simultaneous measurements on both the pressure and suction sides of the blade. Along the midspan measurements are made from 0 to 50% wetted distance on the pressure surface and 0 to 25% wetted distance on the suction side, Figure 2.27. In order to protect the probe tips from particulate damage, the tips are inserted in a groove in the tip endwall of the upstream vane. Thus, an inclined view of the blade midspan region results. The size and shape of the defined area on the blade surface are a function of the distance between the surface and the tip of the probe, and the respective viewing angle. As the distance from the probe tip increases, the defined area is enlarged. Therefore, the pyrometer signal is averaged over this larger area. The shapes of the defined area are either circular or elliptical. Typical shapes and dimensions of the defined area range from a circular area of 3 mm diameter at the suction surface leading edge to an elliptical area with major and minor axes of 16 and 8 mm, respectively, at the pressure surface trailing edge.

The sampling frequency of the pyrometer system is 100 kHz. Thus temperatures along the blade surface are recorded at intervals of 3-4 mm.
2.1 Experimental Setup and Instrumentation

The blade pressure side, this yields approximately 20 measurement points per blade passing (pyrometer No. 1 in Figure 2.27.) One additional measurement point on the pressure surface trailing edge is recorded by pyrometer No. 2. On the blade suction side, 10 measurement points are obtained, since at some blade positions, pyrometer 2 points towards the downstream vane.

The signal-to-noise ratio in the pyrometry measurements is high, since the measurements are obtained in an environment of elevated temperatures. Furthermore, the spatial averaging over millimeter size areas reduces the noise. The measured temperature data are corrected for the surface emissivity (Section 2.2.1); an emissivity of 0.8 is used, which is typical for well-oxidized, nickel alloy turbine blades [57].

Figure 2.7 shows the continuous surface temperature data of pyrometer No. 1 during one engine revolution. The LPT 2 rotor has 86 blades; 22 of them are uncoated nickel alloy blades mounted in two contiguous blocks of 10 and 12 blades. Four of the TBC coated blades are painted with a high-temperature, black paint. These blades have a higher emissivity than the unpainted blades; thus, they appear hotter in the surface temperature measurement. The four painted blades are marked by an arrow in Figure 2.7. This figure also clearly illustrates that the nickel alloy blades have a different temperature pattern that that of the TBC coated blades.

Figure 2.8 shows the uncoated nickel alloy blades in a close-up of the contin-

---

**Figure 2.6: Top view of the second LPT stage showing pyrometer locations.**
uous temperature signal of Figure 2.7. The blade passing frequency of the LPT2 rotor is 4300 Hz. At the pyrometer bandwidth of 100 kHz, a number of 23.3 measurement points are recorded per blade and blade passing. As described in Section 2.2.2 below, the recorded raw temperature data are interpolated to 25 measurement points per blade.

![Figure 2.7: Temperature data from pyrometer 1 obtained at 245 MW power for the 86 blade rotor. Arrows indicate TBC coated blades that are painted black.](image)

The pyrometer locations, viewing angles of the pyrometer optics and the midspan blade geometry are used to develop a two-dimensional model of the measurement geometry (Figure 2.27). From this model, the measurement points that are entirely located on the blade surface are determined. (For some rotor positions, measurement points are on the surface of the downstream vane or simultaneously overlap two adjacent blades.) Pyrometer No. 1 scans the blade pressure side, starting at the leading edge and progressing downstream along the blade midspan to approximately 50% wetted distance (positive values indicate the pressure surface). During the changeover to the adjacent blade, the pyrometer records an average temperature of the midchord area of the present blade and the leading edge area of the next blade (see Figure 2.27). The changeover temperatures, which are indicated by the white circles in Figure 2.8, are not used for the determination of heat transfer coefficients. Thus, from the 25 measurement points per blade, a total number of 20 points is used from pyrometer 1.

The continuous temperature signal from pyrometer No. 2 is shown in Figure 2.9. As seen from Figure 2.27, pyrometer 2 mainly collects temperature data on the upstream part of the blade suction surface. After the leading edge
of the blade passes the viewing cone of the pyrometer optics, one measurement point is received from the downstream portion of the pressure surface. Upon further progression of the rotor, the viewing area of the pyrometer 2 leaves the trailing edge region of the pressure surface and falls on to the downstream vane.

The extraction of useful surface temperature data from pyrometer No. 2 is more challenging than from pyrometer No. 1. Averaged data are recorded during the following three changeover phases:

1. From the suction surface leading edge on to the pressure surface trailing edge region;

2. from the pressure surface trailing edge on to the downstream vane;

3. from the downstream vane on to the suction surface at approximately -25% wetted distance (negative values for the suction surface).

The unbiased temperature data originating from the blade surface are marked by the black circles in Figure 2.9; data averaging two surfaces are indicated by the white circles and are rejected for the measurement of heat transfer coefficients. Clearly visible is the plateau of constant temperature caused by the downstream vane measurements.
The measurement locations on the blade surface are derived from the two-dimensional model which is based on the information of the pyrometer tip locations and the blade midspan contour. Thus, the indicated measurement locations are considered to be accurate to within 2 to 3% of the axial chord. For the measurement point on the pressure surface at 85% wetted distance, it is difficult to conclude with absolute certainty whether the viewing area is entirely located on the blade surface. A more precise assessment of the measurement locations could be achieved by using a three-dimensional model or by boroscopic inspection where the pyrometers are used to illuminate the viewing area on the blade surface.

In Figure 2.10, the temperature data originating from both pyrometers are joined together and plotted versus the wetted distance. Pyrometer No. 1 data are indicated by the black circles, pyrometer No. 2 data are indicated by the white circles. These temperature data were recorded with the engine running at 245 MW power. It should be noted that the agreement between the pyrometer channels at the leading edge is quite remarkable; the temperatures match to within a few degrees.
2.1 Experimental Setup and Instrumentation

2.1.4 Measurement of Thermocouple Time Constant

The reference gas temperature change is measured at the exit of the second combustor (SEV) using a rake of nine type B thermocouples, Figure 2.2. The thermocouple data are recorded at a frequency of 10 Hz. Since the measurement of the blade’s heat transfer coefficient depends on the transient behavior of the measured gas temperature, the time constants of the thermocouples are measured in a free jet experiment. In the absence of radiative and conductive heat fluxes, the thermocouple tip has a first-order system response [72]. The theoretical time constant is defined by Equation 2.3.

\[ \tau = \frac{\rho c D}{4h}, \]  

(2.3)

where \( \rho \) and \( c \) are the density and specific heat of the thermocouple material, respectively, \( D \) the tip diameter and \( h \) the convective heat transfer coefficient. Thus, in the free jet experiment, the Reynolds number and Prandtl number conditions of the turbine are duplicated, a temperature step is imposed, and the time constant determined from the thermocouple’s first-order response.
Experimental Setup. The measurements are conducted in LSM’s free jet facility that is used for the calibration of pressure probes. In the free jet facility, air is accelerated through a nozzle of exit diameter of 100 mm with velocities up to a Mach number of 0.6. Various grids and flow straighteners ensure stable flow conditions and a homogeneous flow field. The flow temperature can also be set between 22 and 32 °C.

The hot-gas rake is positioned at the center of the flow field. Of the 9 thermocouples of the rake, the time constants of the center thermocouple (No. 5) and the adjacent thermocouple (No. 6) are measured. A 3 kW industrial heater is mounted on a movable baffle. During the heating process of the thermocouple, the baffle fully extends into the flow. As the industrial heater heats up the thermocouple, the baffle shields it from the main flow. Temperatures between 200 and 350 °C are achieved, depending on the flow Mach number. Once the thermocouple reads a stable temperature, the baffle is manually retracted, exposing the thermocouple to the main flow. The hot thermocouple is then cooled by convective heat transfer. A micro-switch on the baffle records its position relative to the linear slide, on which the baffle is mounted. The signal from the micro-switch is used in the calculation of the ensemble average of the experimental data. The experimental setup is shown in Figure 2.11.

The thermocouple output signal is conditioned by a programmable signal transmitter. In order to get high temporal resolution, the resulting voltage
signal is recorded at a frequency of 100 kHz.

Two different sets of experiments are conducted: In the first experiment, the heater is operated at maximum power over the entire Mach number range considered. Thus, lower thermocouple temperatures result at higher Mach numbers. The second series of experiments is conducted so that by adjusting the heating power, the thermocouple temperature remains constant over the Mach number range.

The operating point of the free jet facility is set to match the estimated Reynolds number at the exit of the second combustor (SEV) of the GT26 with the Reynolds number in the test rig. The Reynolds numbers in the engine and in the test rig are determined according to Equations 2.4 and 2.5, respectively.

\[
Re_D = \frac{\dot{m} D}{\mu(T)} \tag{2.4}
\]

\[
Re_D = \frac{\rho a M D}{\mu(T)} \tag{2.5}
\]

Here, \( D \) is the diameter of the hot-gas rake, \( \dot{m} \) the engine mass flow rate, \( M \) the Mach number of the flow in the free jet and \( a \) the corresponding speed of sound. For the evaluation of the in-engine Reynolds number, air is assumed as the working fluid. The dynamic viscosity \( \mu \) of air is evaluated at the measured SEV exit temperature and the temperature of the free jet flow, respectively, using gas tables [81]. The variables used for the matching of the Reynolds numbers are summarized in Table 2.4

**Data Reduction.** Ignoring radiative and conductive heat transfer, the thermocouple can be modeled as a first-order element, where only heat transfer due to convection is considered. According to Murdock [72], the dynamic behavior of such a thermocouple is described by the time constant \( \tau \) that is defined by Equation 2.3.

According to the first-order system behavior, a linear temperature drop is observed on convective cooling if the temperature is plotted in semi-logarithmic scale. The time constant is then defined by the 1/e (63.2%) drop in temperature, starting from an arbitrary point within the linear regime, see Fig 2.12. In order to illustrate the dependence of the thermocouple response time on
Measurement of Heat Transfer Coefficients

Table 2.4: Flow conditions in the engine at design load and conditions in the free jet facility.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Engine</th>
<th>Free jet</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rake diameter (m)</td>
<td>D</td>
<td>0.02</td>
<td>0.02</td>
</tr>
<tr>
<td>Dyn. Viscosity (kg/s/m)</td>
<td>( \mu )</td>
<td>( 4.6 \times 10^{-5} )</td>
<td>( 1.8 \times 10^{-5} )</td>
</tr>
<tr>
<td>Massflow (kg/s)</td>
<td>( \dot{m} )</td>
<td>515</td>
<td></td>
</tr>
<tr>
<td>Flow area (m²)</td>
<td>A</td>
<td>1.771</td>
<td></td>
</tr>
<tr>
<td>Density (kg/m³)</td>
<td>( \rho )</td>
<td></td>
<td>1.3</td>
</tr>
<tr>
<td>Speed of sound (m/s)</td>
<td>( a )</td>
<td>346</td>
<td></td>
</tr>
<tr>
<td>Mach number</td>
<td>( M )</td>
<td>0.2</td>
<td></td>
</tr>
<tr>
<td>Reynolds number</td>
<td>( Re_D )</td>
<td>( 1 \times 10^5 )</td>
<td>( 1 \times 10^5 )</td>
</tr>
</tbody>
</table>

The measured time constants for the two thermocouples tested at different flow conditions are shown in Figure 2.14. Measurements are taken over a range of Mach numbers from 0.15 to 0.25. For every operating point, 10 repeat measurements are taken. The symbols show the mean value of the respective test, while the error bars represent one standard deviation. As seen, the results of constant temperature and constant heating power experiments agree well for both thermocouples tested up to a Mach number of 0.2. Above the free jet Mach number of 0.2, some experimental scattering seems to occur.

In the engine experiment, the free stream turbulence level is considerably higher than in the free jet facility. As reported by different authors (i.e. [61] and [92]), increased free stream turbulence intensities enhance the convective heat transfer. Thus, according to Equation 2.3, higher turbulence intensities cause the heat transfer coefficient \( h \) to increase and the thermocouple time
constant \( \tau \) to decrease. According to the work of Moss [70], turbulence intensities of 10% are typical for combustors such as those in the GT26. Therefore, a Nusselt number enhancement factor of 1.1 is assumed [71] and the measured time constants are then corrected for the effects of free stream turbulence.

These corrected time constants are then used to compensate the in-engine reference gas temperature data for the effects of the thermocouples’ dynamic behavior. This compensation is described in Section 2.2.3.
34 Measurement of Heat Transfer Coefficients

\[ T = \frac{q}{0.15} \]

Figure 2.13: Temperature response versus time of thermocouple in the free jet facility at different Mach numbers.

Figure 2.14: Measured time constants for thermocouples No. 5 and 6 of the hot-gas rake.
2.1 Experimental Setup and Instrumentation

2.1.5 Blade Surface Roughness

As described in Section 1.2.2, the surface roughness has a profound influence on the heat transfer rates of the engine components. In this experimental program, heat transfer coefficients are measured on TBC coated and uncoated turbine blades. Due to the lack of access to the turbine blades that were actually installed during the engine tests, the surface roughness of a set of different blades were measured in the laboratory. The blades tested are:

1. Second stage low-pressure blade that was in service for several hundred hours; unpolished TBC coating.
2. First stage high-pressure blade (new, production line); unpolished TBC coating.
3. Second stage low-pressure blade; used in service; uncoated; thermal paint finish.

In the roughness measurements, an optical profilometer MicroProf FRT (Fries Research & Technology GmbH) was used. The profilometer uses an optical sensor to illuminate the sample surface with focused white light. An internal measuring head splits the white light into different wavelengths. A miniaturized spectrometer detects the color of the light reflected by the sample and determines the vertical position of the sample surface by means of an internal calibration table. The measurement resolution is 10 nm with a minimum lateral spacing of 1 to 2 μm. The supplier states a measurement uncertainty of 0.3 to 1.1 % on a measurement of reference grooves of 0.987 to 8.914 μm depth.

Three repeat measurements were taken for every sample and measurement location. Since the sample surfaces are relatively rough, the tangential measurement distance along the surface was chosen to be 15 or 20 mm. The optical profilometer detects the vertical distance of the sample surface to the measuring head. Any surface curvature or slope of the sample is included in the measurement. Thus, the relevant data for the roughness analysis is extracted from the raw data trace by removing a calculated spline fit of that trace (Figure 2.15). With the shape of the sample removed, the traces were analyzed to give the centerline averaged roughness Ra, the rms roughness Rq, the skewness Sk and the kurtosis Ku (definitions see Equations 2.6 to 2.9 on page 37). Table 2.5 summarizes the roughness results of the four blades.
tested. The last two rows in Table 2.5 indicate ALSTOM’s specification for the surface roughness of production line blades. For the polished TBC coating, a Ra value of 4 – 6 μm is specified; for the uncoated as cast nickel alloy blades, the Ra roughness is 3.2 μm.

Comparing the roughness values of the production line thermal barrier coating (No. 7 and 8) with the coating that was in service (No. 1 to 6), an average increase in roughness of approximately 31% is found. On the uncoated blade, the roughness enhancement is more pronounced: Compared with the specified value (3.2 μm, No. 12), an average increase of 92% is observed. (It is assumed that the thermal paint does not significantly alter the surface roughness). Such changes in surface roughness are not unusual: On blades with long service hours, Bons [18] measured roughness levels 4 to 8 times greater than the levels of production line hardware.

The following processes are responsible for increasing the surface roughness levels of engine components in service:

- Foreign deposits such as fuel deposits and airborne contaminants
- Pitting/erosion due to hot corrosion
- Spallation (coated surfaces).
### 2.1 Experimental Setup and Instrumentation

Table 2.5: Summary of blade surface roughness measurements. Last two rows ALSTOM specifications for production line blades. Abbreviations used: PS Pressure Surface; SS Suction Surface; LE Leading Edge; MI Mid-Chord; TE Trailing Edge; TP Thermal Paint.

<table>
<thead>
<tr>
<th>No.</th>
<th>Blade</th>
<th>Surface finish</th>
<th>Location</th>
<th>$Ra, \mu m$</th>
<th>$Rq, \mu m$</th>
<th>$Sk$</th>
<th>$Ku$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>LPTB2</td>
<td>TBC used, rough</td>
<td>PS LE</td>
<td>20.5</td>
<td>25.9</td>
<td>-0.05</td>
<td>2.6</td>
</tr>
<tr>
<td>2</td>
<td>LPTB2</td>
<td>TBC used, rough</td>
<td>PS MI</td>
<td>18.1</td>
<td>22.7</td>
<td>0.19</td>
<td>2.9</td>
</tr>
<tr>
<td>3</td>
<td>LPTB2</td>
<td>TBC used, rough</td>
<td>PS TE</td>
<td>18.4</td>
<td>23.2</td>
<td>-0.12</td>
<td>3.0</td>
</tr>
<tr>
<td>4</td>
<td>LPTB2</td>
<td>TBC used, rough</td>
<td>SS LE</td>
<td>24.8</td>
<td>31.1</td>
<td>-0.23</td>
<td>3.0</td>
</tr>
<tr>
<td>5</td>
<td>LPTB2</td>
<td>TBC used, rough</td>
<td>SS MI</td>
<td>18.8</td>
<td>23.5</td>
<td>-0.12</td>
<td>3.0</td>
</tr>
<tr>
<td>6</td>
<td>LPTB2</td>
<td>TBC used, rough</td>
<td>SS TE</td>
<td>20.1</td>
<td>25.4</td>
<td>0.04</td>
<td>2.9</td>
</tr>
<tr>
<td>7</td>
<td>HPTB1</td>
<td>TBC new, rough</td>
<td>PS MI</td>
<td>15.5</td>
<td>19.7</td>
<td>-0.01</td>
<td>3.2</td>
</tr>
<tr>
<td>8</td>
<td>HPTB1</td>
<td>TBC new, rough</td>
<td>PS MI</td>
<td>15.1</td>
<td>19.1</td>
<td>0.09</td>
<td>3.0</td>
</tr>
<tr>
<td>9</td>
<td>LPTB2</td>
<td>Uncoated, TP</td>
<td>PS MI</td>
<td>6.9</td>
<td>8.9</td>
<td>0.60</td>
<td>3.8</td>
</tr>
<tr>
<td>10</td>
<td>LPTB2</td>
<td>Uncoated, TP</td>
<td>SS MI</td>
<td>5.4</td>
<td>6.9</td>
<td>0.34</td>
<td>3.0</td>
</tr>
<tr>
<td>11</td>
<td>LPTB2</td>
<td>TBC new, polished</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>LPTB2</td>
<td>Uncoated, as cast</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

\[ Ra = \frac{1}{N} \sum_{i=1}^{N} |y_i| \]  
\[ Rq = \sqrt{\frac{1}{N} \sum_{i=1}^{N} y_i^2} \]  
\[ Sk = \frac{1}{N \cdot Rq^3} \sum_{i=1}^{N} y_i^3 \]  
\[ Ku = \frac{1}{N \cdot Rq^4} \sum_{i=1}^{N} y_i^4 \]
While these mechanisms generally increase the surface roughness, thermal barrier coating spallation may at times cause a reduction in local roughness. In such cases, the spallation follows a cycle from pits to craters to completely exposed metal, which usually has a lower roughness than the original coating. Once the metal surface is exposed, it again becomes subject to the roughness inducing mechanisms.

For the TBC coated LPTB2 blade (No. 1 to 6), measurements were taken on different locations along the blade’s midspan. As seen from Table 2.5, the Ra roughness level at the suction surface leading edge (SS LE) is distinctly higher than on the other parts of the blade’s midspan. The increased surface roughness in this region of the blade is caused by fuel deposits and pitting, which is apparent on visible and tactile inspection of the blade. The same behavior was observed by Taylor [99] from roughness measurements on first stage rotor blades of two military engines. Taylor reported high roughness levels in the leading edge region of the suction surface. On one of the blades, however, high roughness values were also measured at the mid-chord and trailing edge of the pressure surface. Bons [18] observed the same trends: In his study of nearly 100 turbine components, the suction surface LE on average was rougher than the TE. On the pressure surface, the opposite was true: The TE was rougher than the LE.

The skewness (Equation 2.8) is negative for most of the TBC coated blades. Negative skewness means that the area below the meanline of the profile exceeds the area above the meanline. That is, the profile tends to have more deep valleys than high peaks. The thermal paint blade (No. 9 and 10) is the only one that shows distinct positive skewness values on both the pressure and the suction surface. This result is corroborated by Bons [18], who reported negative average skewness on coated blades, but positive average skewness on uncoated surfaces.

The average kurtosis is positive for all the samples. According to Boyle [19] a positive kurtosis indicates a spiky, rather than bumpy surface profile.
2.2 Data Reduction

2.2.1 Blade Surface Temperature

According to Wien’s approximation of Planck’s law, the spectral radiance of a blackbody surface is given by Equation 2.10.

\[ L_{\lambda,b} = \frac{C_1}{\lambda^5} \exp \left( \frac{-C_2}{\lambda T} \right) \]  

(2.10)

The spectral radiance of a real surface is less than that of a blackbody by the emissivity factor \( \varepsilon \) and is defined by Equation 2.11.

\[ L_{\lambda} = \varepsilon L_{\lambda,b} \]  

(2.11)

In the determination of the surface temperature, grey-body behavior is assumed. The emissivity of a grey body is constant with wavelength. The directional behavior of the emissivity of metals was observed to be essentially constant for approximately 50° away from the surface normal [87]. Therefore, the normal total emissivity is used in Equation 2.11, which is a valid assumption for oxidized turbine blades [57]. Since the LPT2 rotor is not directly exposed to radiative flux from the combustors, it can also be assumed that errors due to reflected radiation are small. Thus, Equations 2.10 and 2.11 can be combined following White and Nicholas [107] to yield an expression for the temperature corrected for surface emissivity:

\[ T_{sc} = \frac{C_2}{\lambda \ln(\varepsilon) + \frac{C_2}{T_{sm}}} \]  

(2.12)

where \( T_{sc} \) is the corrected surface temperature, \( T_{sm} \) is the surface temperature measured directly by the pyrometry system, \( C_2 \) is the second constant in Planck’s law, \( \varepsilon \) represents the normal total emissivity and \( \lambda \) the working wavelength of the pyrometer. Other sources of error such as lens contamination and non-luminous gases are not accounted for, since the determination of heat transfer coefficients require the changes in temperature rather than absolute temperature levels.
During the thermal transients generated by the sudden load changes of the engine, the pyrometry system continuously records the midspan surface temperatures. This continuous record includes surface temperatures of consecutive blades. The temperature-time histories for specific locations on the blade surface are then extracted from this record, see Figure 2.16. The extraction is accomplished by means of the key phaser signal, which is simultaneously recorded with the temperature data. The conditioning of the key phaser signal is described in Section 2.2.2.

In the generation of temperature-time histories, data sequences of one revolution length are stretched to a common time base. The stretching algorithm reduces revolution-to-revolution time differences caused by variations in rotational speed. However, the stretching algorithm only levels off these differences when the respective signals are in phase. This situation is demonstrated in Figure 2.17a by a simple sine wave signal. In Figure 2.17a, the
2.2 Data Reduction

![Graphs showing original and stretched signals](image)

(a) Signals of equal phase.

(b) Phase shifted signals.

**Figure 2.17:** Application of stretching algorithm to sine wave signals.

Original signals are in phase, but their periods are different. This example represents the recorded temperature data as the rotational speed changes. Using the stretching algorithm, the two signals are fitted to a common time base to become perfectly aligned. In Figure 2.17b, the two original signals are phase shifted. In this case, the stretching does not align the signals as successfully as in the previous case. The phase shift is still present after the stretching.

The above example demonstrates that the stretching algorithm works satisfactorily if the processed data have equal phase. In the engine environment, phase shifted data are acquired if the start of an engine revolution is not accurately detected for the generation of temperature-time histories.

The acquisition frequency of the pyrometry system is fixed at a rate of 100 kHz as described in Section 2.1.3. When the rotational speed of the engine changes slightly, the number of measurement points that are recorded per engine revolution varies. Thus, in order to have an equal number of measurement points in every data segment of one revolution length, the stretched data record is interpolated. At a rotational speed of 50 Hz, a theoretical number of 2000 data points are acquired during one engine revolution. This number of 2000 points per revolution corresponds to approximately 23.3 points per blade and blade passing. Using the interpolation procedure, a data set of 2000 points per revolution is increased to a number of 2150 (which corresponds to 25 points per blade). Linear interpolation is used for this process.
Thus, temperature-time histories for specific locations on the turbine blade are determined. The resulting temperature-time histories are smoothed by a three-point moving average procedure. The entire compilation process is shown schematically in Figure 2.16 on page 40 and a representative surface temperature history on the pressure surface of a nickel alloy blade is shown in Figure 2.18.

### 2.2.2 Key Phaser Signal Conditioning

This section describes the conditioning of the key phaser signal. The key phaser is a once-per-revolution signal that is recorded simultaneously with the blade surface temperature data. It is used to phase-lock a position on a specific blade for the generation of a temperature-time transient for that specific blade location.

Figure 2.19 shows an enlargement of the key phaser voltage signal versus time. The signal is recorded at a frequency of 100 kHz, at the same rate as the temperature data. The algorithm for the post-processing of the key phaser signal is such that it detects the start of the engine revolution when the voltage signal drops below a value of 10 mV. In Figure 2.19, this point
2.2 Data Reduction

Figure 2.19: Blowup of key phaser voltage signal. White circle indicates the detection of the start of one engine revolution.

is indicated by a white circle. The start of this engine revolution is set at this position in time. The detection of the start of the event is always done on the negative (falling) flank of the signal. Figure 2.20a shows the result of the detection process for a time period of 25 revolutions (0.5 seconds). At a nominal engine speed of 50 Hz, a theoretical number of 2000 data points are recorded per revolution. As can be seen from Figure 2.20a, the detection of the start of the revolution occurs at different voltage levels. In the case that is shown in Figure 2.20a, a sawtooth pattern can be observed, with the detection voltage abruptly rising after a few revolutions. Although the engine frequency is very nearly constant, it is not always an exact integer fraction of the acquisition frequency. In such a case, the theoretical number of key phaser data points per engine revolution is not 2000, but slightly less or more. Thus, the small time difference to the theoretical number of 2000 points per revolution is added to the following revolution. Once the sum of these small time differences add up to an additional data point, the respective revolution has one data point more than the ideal 2000 points, which causes the abrupt rising of the detection voltage.

The sawtooth pattern is not only observed in the key phaser signal, but also in the resulting surface temperature transient. The sawtooth is shown in Figure 2.21 by the white circles. This phenomenon has its largest effect in
areas on the blade where the surface temperature gradient is high, i.e. around the leading edge region of the blade.

In order to get a more precise phase-locking, the key phaser data is interpolated. The linear interpolation increases the number of data points of the key
phaser signal. The respective temperature data is also interpolated, before the continuous signal is divided into revolution-wise segments. Figure 2.21 shows the result of the interpolation. With the use of the original key phaser signal, a distinct saw tooth pattern is occasionally observed in regions of high thermal gradients. The application of a 10 point linear interpolation to both the key phaser and the temperature data reduces the amplitude of the saw tooth pattern by approximately 50%. A further increase in the number of interpolation points from 10 to 20 does not yield any significant improvement. Thus, the 10 point linear interpolation is used in the data reduction process.

From the above analysis, it can be concluded that the present data acquisition method may be improved. The acquisition frequency of the key phaser signal may be one order of magnitude higher than the acquisition of the temperature data. The high frequency key phaser signal may significantly reduce the blade surface temperature errors caused by the errors in the phase-locking of blade positions.

### 2.2.3 Driving Gas Temperature

The driving gas temperature change is measured at the exit of the second combustor (SEV). The temperatures from the nine individual thermocouples of the hot-gas rake are averaged and compensated to account for the thermocouples’ dynamic behavior. In the compensation, the experimentally determined time constant $\tau$ is used. The measurement of the thermocouples’ time constant was presented in Section 2.1.4. The compensation is described by Equation 2.13.

$$T_{gc} = T_{gm} + \tau \cdot \frac{dT_{gm}}{dt}$$  \hspace{1cm} (2.13)

where $T_{gc}$ is the gas temperature corrected for the thermocouple dynamics and $T_{gm}$ the uncorrected mean temperature. The corrected temperature is then translated from the absolute frame of reference at the exit of the combustion chamber to the relative frame of reference at the inlet of the second blade row. This translation is done by means of the results from an ALSTOM through-flow code. The relation between the SEV exit temperature and the LPT blade 2 inlet temperature is shown in Figure 2.22.
Figure 2.22: Relationship between SEV exit temperature and LPT blade 2 entry temperature from through-flow solver.

Figure 2.23 shows the corrections that are applied to the gas temperature data. The original temperature history is indicated by the white circles (A). After the original signal is compensated for the thermocouples’ dynamic behavior (B), the temperatures are translated to the relative frame of the LPT 2 rotor (C). Figure 2.23 demonstrates that of the two temperature corrections that are applied to the measured driving gas temperature, the correction for the thermocouples’ dynamic behavior has the dominant effect on the change in gas temperature.

2.2.4 Synchronization of Surface and Gas Temperature

The surface and gas temperature data used to determine the heat transfer coefficient are measured using two different data acquisition systems. The gas temperature data is stored in the central ALSTOM system, while for the high-frequency surface temperature data, a separate system is used.

Before the start of the experiments, the two systems were synchronized. The synchronization was accurate to within 1 second. During the data reduction process, a variable time lag was observed between the surface temperature signal and the gas temperature signal. Thus, in the evaluation of heat transfer
2.2 Data Reduction

Figure 2.23: Measured gas temperature variation vs. time. A: Temperature as measured at the exit of the combustion chamber. B: Temperature corrected for thermocouple dynamics. C: Temperature corrected for translating to the LPTB2 relative frame.

coefficients described in Section 2.2.5 below, the gas temperature history is shifted relative to the surface temperature (shown in Figure 2.24).

This synchronization of the temperature histories is achieved by the least-squares procedure that is used to determine the heat transfer coefficient distribution. Thus, the least-squares procedure simultaneously solves for the heat transfer coefficients and the time lag that minimizes the least-squares problem.

Typical time lags between the two signals are on the order of 0.5 seconds.

2.2.5 Evaluation of Heat Transfer Coefficients

A layered, semi-infinite, time-dependent heat conduction model is used to determine the heat transfer coefficients at the measurement points along the blades’ midspan. In the case of the uncoated blades, the conduction model has only one semi-infinite, nickel alloy layer, whose homogeneous thermal properties are evaluated at the blade surface temperature. The coated blades have a four-layered structure that is comprised of a ceramic top coat, a thermally
grown oxide layer, a bond coat and the nickel alloy blade. In the present work, the bond coat and oxide layer are assumed to be sufficiently thin such that they can be neglected. Thus, a two-layer model, comprised of a finitely thick ceramic layer and a semi-infinite nickel alloy layer, is used for the coated blades. The temperatures in the two layers are governed by

\[
\frac{\partial T_1}{\partial t} = \alpha_1 \frac{\partial^2 T_1}{\partial x^2} \quad 0 \leq x \leq x_i \quad (2.14)
\]

\[
\frac{\partial T_2}{\partial t} = \alpha_2 \frac{\partial^2 T_2}{\partial x^2} \quad x_i \leq x \leq \infty \quad (2.15)
\]

where \(x_i\) is the ceramic layer thickness. The thermal properties of the layers are evaluated at their respective upper boundary temperatures. The two interface conditions for the coated blades are given by Equations 2.16 and 2.17.

\[
T_1 = T_2 \quad x = x_i \quad (2.16)
\]
\[ k_1 \frac{\partial T_1}{\partial x} = k_2 \frac{\partial T_2}{\partial x} \quad x = x_i \] (2.17)

Note that for the uncoated blades, only Equation 2.14 is necessary with \( x_i = \infty \). At the upper blade surface, a heat convection boundary condition is specified as

\[ -k_1 \frac{\partial T}{\partial x} \bigg|_{x=0} = h(T_{gc}(t) - T_s(t)), \] (2.18)

where \( T_s(t) \) is the predicted surface temperature from Equation 2.14 and \( T_{gc}(t) \) is the corrected driving gas temperature (Equation 2.13). The second boundary condition is also of the convection type, but the modeled wall thickness is chosen to be large. The relatively thick wall ensures that over the time interval considered, the penetration depth of the thermal pulse is always less than the modeled wall thickness. The validity of the semi-infinite solid assumption was tested by changing the boundary condition of the internal wall. A change of the internal heat transfer coefficient of \( \pm 100\% \) results in a change of the external heat transfer coefficient of less than 0.4\%. Thus, the semi-infinite solid assumption seems to be justified.

Since the observed temperature variation at the blade surface is small (on the order of 10 K), the thermal properties of the blades are assumed to be constant with temperature.

The second blade row is not directly exposed to the radiation from the combustors, i.e. each cooled blade is surrounded by similarly cooled blades. As the surface temperature measurements show, the midspan region of the blades is nearly isothermal. Thus, it is assumed that the net radiative heat exchange between the surfaces is small compared with the convective heat transfer. Therefore, only the heat flux due to convection is considered in the external boundary condition, Equation 2.18. This simplifying assumption is supported by Siegel [88], who reported that for turbine blades away from the combustor, there is very little radiative exchange between the blades.

The Reynolds number variation through the thermal transient is estimated using the results of the mean line analysis, see Section 2.3. For the time period considered in the data fitting procedure (four seconds), the Reynolds number variation is less than 5\%. On a large portion of the blade surface,
the heat transfer coefficient changes as $\text{Re}^{0.8}$, in the leading edge region as $\text{Re}^{0.5}$ [59]. Therefore, the heat transfer coefficient is expected to vary less than 3.6% and thus, it seems justifiable to assume a constant heat transfer coefficient through the four seconds time interval.

The differential equations 2.14 and 2.15 are numerically solved, using an explicit finite difference scheme. The unknown variable in this set of equations is the time-invariant external heat transfer coefficient, $h$. It is determined by calculating the least-squares fit of the modeled change in surface temperature to the measured change in surface temperature, Figure 2.25. As described in Section 2.2.4, the time lag between the measured surface temperature and the measured gas temperature histories is also found by solving the least-squares problem.

On the TBC coated blades, the ceramic top-coat is deposited by air-plasma-spray (APS) deposition. The APS deposition procedure is such that the leading edge area of the blade is not entirely covered by the top-coat. It tapers off towards the leading edge on both the suction and the pressure surface, leaving the leading edge uncoated. Hence, a variable layer thickness, $x_i$, is integrated into the heat conduction model. The ceramic top-coat profile along the leading edge is shown in Figure 2.26, normalized by the average
2.3 Mean Line Analysis

The experimental program in the ALSTOM GT26 consists of sudden load changes at various initial loads, resulting in gas and surface temperature changes. From these temperature changes, the heat transfer coefficients are derived on the second blade of the low-pressure turbine. In order to relate the measured heat transfer coefficients to the flow condition, the Reynolds and Mach numbers are computed using a mean line calculation. According to Simoneau [90], the midspan region can be defined as the middle 50 to 75 percent of the blade span. In this region, the endwall boundary layer has little or no influence; the flow is nearly two-dimensional. Thus, a mean line analysis is sufficient to determine the Reynolds and Mach numbers of the flow. Since the GT26 is connected to the grid during the experiments, the rotational speed is constant. In turn, the incidence angle of the flow may change considerably.

Figure 2.26: Ceramic top-coat layer profile around the leading edge of the blade.

pressure side top-coat thickness. In the surface temperature measurement, the viewing area of the pyrometer on the blade surface extends over several millimeters. Therefore, the measured top-coat profile is smoothed and in the heat transfer model, these smoothed values (symbols in Figure 2.26) are used for the layer thickness $x_i$. 
at reduced temperature and massflow. Therefore, it is necessary to compute the incidence angle for every load step.

The ALSTOM through-flow solver Q263 is used to determine the relative total temperature at blade inlet from the measured SEV exit total temperature. Figure 2.22 gives the relationship between the SEV exit temperature and the LPT blade 2 inlet temperature. For measured SEV temperatures between the stated values, a linear interpolation is applied. The relative total temperature $T_{ord,m}$ is used to specify the off-design condition, which the engine is operated at. In the first step, the static temperature at station 1, $T_1$, is assumed to be known. (For the station labeling see Figure 2.27). With this assumed static temperature, the static pressure $P_1$, is iteratively found by calculating the capacity of the rotor and matching it to the design capacity. The Mach number at station 1, $M_1$, is determined from the isentropic relation given by Equation 2.19.

$$\frac{P_{\infty}}{P_1} = (1 + \frac{\gamma - 1}{2} M_1^2)^{\gamma/\gamma - 1},$$  \hspace{1cm} (2.19)
2.3 Mean Line Analysis

where $P_{0q}$ is the total pressure at vane 2 inlet and $P_1$ is the static pressure at blade 2 inlet. For the total pressure at station 0 it is assumed that the pressure ratio across stage 1 of the low pressure turbine is constant. Thus, the total pressure at stator 2 inlet (rotor 1 outlet) can be calculated from the measured SEV exit pressure, since the pressure ratio is known at design. The validity of this assumption will be verified later by determining the off-design pressure ratios across rotor 2. With the Mach number known, the absolute velocity at station 1 reads

$$V_1 = Ma_1 \sqrt{\gamma RT_1}. \quad (2.20)$$

The axial component of $V_1$ is found from the velocity triangles shown in Figure 2.28 and is given by Equation 2.21.

$$V_{1x} = V_1 \cos \alpha_1 \quad (2.21)$$

The relative flow angle $\beta_1$ is computed as a function of the constant absolute flow angle $\alpha_1$, the absolute velocity $V_1$ and the blade speed at midspan $U_1$ according to Equation 2.22.

$$\cot \beta_1 = \frac{U_1 \cos \alpha_1}{V_1 \sin^2 \alpha_1 - U_1 \sin \alpha_1} + \cot \alpha_1 \quad (2.22)$$
The relative flow velocity follows as

\[ V_{r1} = \frac{V_1 \sin \alpha_1 - U_1}{\sin \beta_1}. \] (2.23)

Now the relative total temperature \( T_{orel1} \) and the relative total pressure \( P_{orel1} \) can be determined, which are used to calculate the capacity of the rotor.

\[ T_{orel1} = T_1 + \frac{V_{r1}^2}{2c_p} \] (2.24)

\[ \frac{P_{orel1}}{P_1} = \left( \frac{T_{orel1}}{T_1} \right)^{\gamma/\gamma-1} \] (2.25)

Then the rotor capacity is

\[ Q_1 = \dot{m}_1 \frac{\sqrt{T_{orel1}}}{P_{orel1}} = A_1 V_{x1} \frac{P_1}{RT_1} \frac{\sqrt{T_{orel1}}}{P_{orel1}} \] (2.26)

By iteratively changing the static pressure \( P_1 \), the capacity that equals the design capacity is found. However, the resulting static pressure is based on the assumed static temperature at rotor inlet. Therefore, a second iteration is performed. This second iteration determines the static temperature for which the relative total temperature \( T_{orel1} \) equals the relative total temperature \( T_{orel,m} \) derived from the measured SEV exit temperature.

For the blade outlet parameters, an analogous calculation is performed. Here the static pressure at blade exit \( P_2 \), is iteratively adjusted to match the capacities. It is assumed that the capacity of stator 3 is constant for all off-design states. The stator 3 capacity is defined as

\[ Q_2 = \dot{m}_2 \frac{\sqrt{T_{oabs2}}}{P_{oabs2}} = A_2 V_{x2} \frac{P_2}{RT_2} \frac{\sqrt{T_{oabs2}}}{P_{oabs2}} \] (2.27)

Table 2.6 summarizes some results of the mean line calculation for different load cases. The absolute temperature at rotor inlet is given in Kelvin and the pressure unit is bar. The incidence angle, \( i \), is given in degrees, where case No. 1 is taken as the zero incidence case. The Reynolds numbers are based
2.3 Mean Line Analysis

Table 2.6: Results of the mean line calculation for different engine loads.

<table>
<thead>
<tr>
<th>Case</th>
<th>Power</th>
<th>$T_{oabs1}$</th>
<th>$P_{oabs1}$</th>
<th>$P_2$</th>
<th>$i$</th>
<th>$Ma$</th>
<th>$Re_C$</th>
<th>$\frac{P_{oabs1}}{P_2}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>275</td>
<td>1409</td>
<td>10.4</td>
<td>4.8</td>
<td>0</td>
<td>0.88</td>
<td>$2.4 \times 10^6$</td>
<td>2.19</td>
</tr>
<tr>
<td>3</td>
<td>220</td>
<td>1389</td>
<td>8.4</td>
<td>3.8</td>
<td>-0.5</td>
<td>0.88</td>
<td>$2.0 \times 10^6$</td>
<td>2.19</td>
</tr>
<tr>
<td>9</td>
<td>140</td>
<td>1296</td>
<td>6.5</td>
<td>3.0</td>
<td>-2.9</td>
<td>0.87</td>
<td>$1.6 \times 10^6$</td>
<td>2.18</td>
</tr>
<tr>
<td>14</td>
<td>70</td>
<td>1208</td>
<td>4.9</td>
<td>2.2</td>
<td>-5.6</td>
<td>0.87</td>
<td>$1.3 \times 10^6$</td>
<td>2.17</td>
</tr>
</tbody>
</table>

Figure 2.29: Inlet and outlet velocity triangles. Solid lines denote design case at 275 MW; dashed lines represent off-design case of 140 MW net engine power.

on the blade chord and rotor exit conditions. The last column in Table 2.6 shows the ratio of $P_{oabs1}$ and $P_2$. Here, the ratio is nearly constant throughout the power range considered. This result indicates that the assumption of a constant pressure ratio across the first turbine stage is valid.

Figure 2.29 shows a sketched cross-section of the LPT blade 2 including the resulting inlet and outlet velocity triangles of the approximate engine design condition (275 MW) and an off-design case (140 MW).
The mean line calculation described above can be summarized as follows:

- For every load step, the measured SEV exit total temperature and static pressure are used as an input for the calculation.

- Using the solution of the ALSTOM through-flow code Q263, the measured SEV exit total temperature is translated to the relative frame of reference of the LPT blade 2. The relative total temperature at blade inlet sets the condition of the blade flow.

- The static pressures at rotor inlet and outlet are iteratively computed by matching the flow capacities of blade 2 and vane 3 to their respective design capacities. The design capacities are defined in the Q263.

- Once the flow conditions are known, the blade Reynolds numbers (based on the blade exit variables) and incidence angles are computed.

The results of the mean line analysis described in this section can now be used to relate the measured heat transfer rates to the corresponding flow conditions. The temperatures and pressures also serve as input quantities for the boundary conditions of the CFD analysis that shall be presented in Chapter 3.

## 2.4 Experimental Results

In the following sections, the experimental results from the in-engine measurements for the blade heat transfer coefficient are presented. In Sections 2.4.1 and 2.4.2, the Nusselt number results are shown and discussed for a set of 10 uncoated nickel-alloy and TBC coated blades, respectively. Heat transfer data for the leading edge area are presented in Section 2.4.3. Following the heat transfer data, the uncertainty analysis for the heat transfer measurements is described in Section 2.5.1. Finally, Section 2.5.2 presents the assessment of the noise levels in the surface temperature measurements.

### 2.4.1 Nusselt Number Distribution on Uncoated Blades

**Pressure Surface.** For the uncoated blades, the Reynolds number effect on the streamwise distribution of Nusslet number is shown in Figure 2.30.
2.4 Experimental Results

Positive abscissa values indicate the pressure surface of the blade and negative values the suction surface. All the heat transfer data are normalized by $\text{Nu}_0$, the turbulent flat plate Nusselt number based on the exit Reynolds number and the blade chord that is given by Equation 2.28.

$$\text{Nu}_0 = 0.0296 \cdot R\text{e}_C^{4/5} P_r^{1/3}$$  \hspace{1cm} (2.28)

On the leading edge region on the blade pressure surface, the laminar flow develops on a convex surface curvature. The convex curvature and the favorable pressure gradient stabilize the boundary layer, which grows progressively. This thickening of the boundary layer leads to a drop in heat transfer. After a minimum at approximately 10% wetted distance, the Nusselt number distribution shows a sharp increase (point (a) in Figure 2.30). Figure 2.31 shows the isentropic relative Mach number distribution from the CFD prediction (Chapter 3). As seen, the flow is strongly accelerated and then diffused on the upstream part of the pressure surface. The sudden diffusion is likely to form a separation bubble on the pressure surface, which is followed by the reattachment of the flow. The visualization of a pressure surface separation bubble is shown in Figure 2.32 from the works of Brear et al. \cite{21}. The start of the flow separation is denoted by $S$, while $R$ indicates the mean reattachment
In the region of the separated shear layer, reduced heat transfer rates are observed, while at the point of reattachment, the heat transfer increases. The flow separation (feature (a) in Figure 2.30) on the blade pressure surface is not observed in all test cases. Cases with close to zero incidence (high Reynolds number) and high negative incidence (low Reynolds number) do not show the separation bubble, Figure 2.30. Close to design operation at low incidence,
there is a possibility that the flow separation may be absent or too small to be detected by the pyrometry measurements. For the cases that show separating flow, the location where the heat transfer rate increases, is manually determined from the Nusselt number distribution. The increase in Nusselt number is assumed to coincide with the onset of the reattachment behavior of the flow. In Figure 2.33, these locations are plotted versus the incidence angle of the different load cases. As seen, the reattachment point on the blade surface moves downstream as the incidence angle decreases. The same phenomenon was reported by Brear [21] in his study of pressure side separation in a linear cascade. Brear suggested that the extent of the separation increased with reduced incidence angle. In the engine experiment, the Reynolds number and the incidence angle are reduced simultaneously as the engine is operated off-design. However, the characteristics of pressure surface separation are primarily affected by the incidence [21]. At high negative incidence (low Reynolds number), the reattachment point may be shifted as far downstream as to the location where the flow reaccelerates (approx. 40% wetted distance). In this case, an increase in Nusselt number due to flow reattachment would coincide with an increase due to the concave surface curvature.

Downstream of the reattachment point, a turbulent boundary layer develops. The heat transfer rates are then mainly determined by the effects of the con-
cave surface curvature and free stream turbulence, counteracting the effects of the favorable pressure gradient [32]. Downstream of approximately 40% wetted distance (point (b) in Figure 2.30), an increase in Nusselt number is observed. In this part of the blade surface, the heat transfer varies as [59]

\[ Nu \sim Re^{0.8}, \]  

(2.29)

showing classical turbulent boundary layer trends. Nusselt number values for all test cases are shown in Figure 2.34 at 41% wetted distance. Equation 2.29 and the measurement standard deviation of 15.8% from the error assessment (Section 2.5) are also shown.

**Suction Surface.** Close to the leading edge, the boundary layer on the suction surface of the blade behaves in a similar way to that on the pressure surface. The growth of the laminar boundary layer causes the Nusselt numbers to decrease. Due to the acceleration and subsequent diffusion (Figure 2.31), the flow regime changes from laminar to transitional. The transition from laminar to turbulent is accompanied by an increased heat transfer level, point (c) in Figure 2.30 on page 57. The upstream portion of the suction surface is also subjected to vane wake interaction [32], which encourages transition at

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**Figure 2.34:** Measured Nusselt numbers versus blade Reynolds number. Solid line represents Equation 2.29. Dashed line shows the measurement standard deviation of 15.8% from the error assessment.
the relatively high Reynolds numbers of this large machine.

2.4.2 Nusselt Number Distribution on Coated Blades

**Pressure Surface.** For the TBC coated blades, the streamwise Nusselt number distributions for four engine operating conditions are shown in Figure 2.35. On the blade pressure surface, the flow separation previously described on the uncoated blades, does not appear. Blair [15] studied the heat transfer in a large-scale turbine rotor passage on rough and smooth turbine blades. He suggested that on the rough blades, the surface roughness may have suppressed the separation bubble effects that were observed on the smooth blades. Generally, a TBC coated surface is rougher than an uncoated one, but the ceramic top-coat of the present hardware used in the experiment is polished. Thus, in their respective production line condition, the surface roughness of the coated and the uncoated blades are of similar magnitude. ALSTOM specifies the roughness value Ra of the uncoated blades as cast as 3.2 μm and that of the polished TBC coating as 4 to 6 μm (see also Section 2.1.5). Bons [18] reported an increase in surface roughness on blade samples with high operating hours compared with the production line hardware, but roughness promoting
mechanisms apply to both coated and uncoated blades. As seen from Figure 2.36, the overall heat transfer levels of the TBC coated blades are of the same order as the levels of the uncoated blades. The similar heat transfer levels suggest a similar surface roughness on the coated and the uncoated blades. For these reasons, the absence of the flow separation on the pressure surface of the TBC coated blades can probably not be explained by the mechanism described by Blair.

As described in Section 2.2.5, the ceramic top-coat tapers off towards the leading edge of the blade. Thus, when the flow passes the wedge formed by the ceramic top-coat, it is first strongly diffused and then accelerated. The diffusion/acceleration may cause an immediate separation and reattachment of the flow and early transition to turbulent flow. Such a behavior would then suppress the large scale flow separation that is observed on the uncoated blades.

Towards the mid-chord of the blade pressure side, the boundary layer behavior is dominated by the concave surface curvature and acceleration with similar heat transfer rates on the coated and uncoated blades, Figure 2.36.

Also shown in Figure 2.36 is the measured blade-to-blade variation in heat transfer for a set of 10 coated and uncoated blades, respectively. The uncoated blades have higher blade-to-blade variation on the pressure surface than the TBC coated blades. The higher variation may indicate the sensitive nature of the flow separation on the uncoated blades. On the pressure surface of the coated blades, assuming the flow has experienced early transition, the turbulent boundary layer is more stable. Thus, less blade-to-blade variation is observed.

Suction Surface. On the blade suction surface of the TBC coated blades, the same heat transfer pattern is observed as on the uncoated blades. Figure 2.35 indicates that the heat transfer rates increase when the flow becomes transitional. Also, the general Reynolds number influence on heat transfer is clearly visible on the suction surface, Figure 2.35.

The comparison of the heat transfer levels on the upstream part of the suction surface between coated and uncoated blades shows higher heat transfer on the uncoated blades. However, the measured blade-to-blade variation is higher on the coated blades. The reason for this behavior is not yet fully understood. Compared with the TBC blades, the contiguous blocks of nickel blades in the
2.4 Experimental Results

Figure 2.36: Comparison of TBC coated and uncoated blades for test case No. 8 (170 MW). Error bars indicate measured blade-to-blade variation in heat transfer.

test engine have slightly higher throat areas due to the absence of the thermal barrier coating. Measurements of the throat areas on the coated and uncoated blades showed an increase of approximately 1.5%. The higher throat areas result in reduced blockage of the flow, which causes slightly positive incidence angles on the nickel blades compared with the TBC blades. Several authors reported increased heat transfer on the suction surface upon positive incidence (Blair [15], Arts [9], Giel [46]). Therefore, the difference in incidence angle on the nickel blades may explain the increased heat transfer rates on the blade suction surface.

2.4.3 Leading Edge Heat Transfer

The stagnation point heat transfer of all test cases is shown in Fig 2.37. The correlation of Lowery and Vachon, Equation 2.30, is also shown for comparison [61]. This correlation takes into account the influence of the free stream turbulence on the heat transfer in the stagnation region.
Figure 2.37: Measured Nusselt numbers on the leading edge versus blade Reynolds number. Correlation of Lowery and Vachon Equation 2.30 (L&V) shown for two turbulence intensities.

\[ Nu_d = Re_d^{1/2} \left( 1.01 + 2.624 \cdot \frac{TuRe_d^{1/2}}{100} - 3.07 \cdot \left[ \frac{TuRe_d^{1/2}}{100} \right]^2 \right) \]  

(2.30)

In the above equation, the Reynolds number is based on the rotor relative inlet velocity and the kinematic viscosity is evaluated at the static temperature of the inlet station of the blade. The leading edge of the blade is approximated by a circle of diameter \( d \). Predictions for two turbulence intensities are shown in Fig 2.37. The error bars represent measured blade-to-blade variation in Nusselt number.

Despite the highly unsteady nature of the flow around the stagnation point, the agreement between the experiment and the prediction is quite good. The Nusselt numbers for the coated and uncoated blades are also very similar. This is the expected result, since the leading edge of a TBC coated blade is uncoated, as described in Section 2.2.5.

The rotational speed of the engine is constant for the different load settings in the test matrix. When the power is significantly reduced, the gas temperature falls. Thus, reduced engine power causes a decrease in incidence angle, i.e. the flow turns towards the blade suction side. A range of just 11° incidence is predicted by a simple mean line analysis, see Table 2.2 on page 23. From the Nusselt number distribution of all the load cases, the location of the stagnation
2.4 Experimental Results

![Graph showing stagnation point location derived from Nusselt number distribution versus incidence angle.](image)

Figure 2.38: Stagnation point location derived from Nusselt number distribution versus incidence angle.

point on the blade surface is determined. This location is then plotted versus the flow incidence angle from the mean line analysis, Figure 2.38. As seen, the expected shift of the stagnation point location with the change in incidence does not occur in the experiment. With the exception of three load cases, the stagnation point is always detected at the same blade surface coordinate. A lack of spatial resolution in the surface temperature measurement at the leading edge of the blade may be the source of this measurement result.

2.4.4 Reynolds Number Scaling

As described in the introduction section of this thesis, the flow Reynolds number has a profound effect on turbine heat transfer rates. Scaling the heat transfer data using the Reynolds numbers at which the data were taken allows the comparison among experiments and the evaluation of the influence of the Reynolds number.

Figure 2.39 shows the heat transfer distributions at various Reynolds numbers scaled by $Re^{0.8}$. This scaling factor is used to compare the heat transfer to the isothermal flat plate correlation according to Equation 2.28. On the suction surface at approximately 25% wetted distance of the uncoated blades
(Figure 2.39a), the heat transfer rates collapse reasonably well to the same value. Here it can be argued that the flow becomes turbulent after the transition from the laminar boundary layer which is observed to occur close to the leading edge. The influence of the wake from the upstream vane shifts the transition point towards the leading edge. On the upstream part of the pressure surface, the $Nu \times Re^{-0.8}$ data differ due to the separation bubble found in some of the load cases. The last point on the pressure surface shows classical turbulent boundary layer behavior.

On the suction surface of the TBC coated blades (Figure 2.39b), the scaled heat transfer data also collapse within a quite narrow band. Although a turbulent boundary layer is expected on most of the pressure surface, the Reynolds scaling does not appear to collapse the data very well.

Similar turbine heat transfer behavior was reported by Abhari [2], who studied heat transfer processes in a fully scaled transonic turbine stage. Abhari showed that on the suction surface, the $Re^{0.8}$ scaling worked well, with the exception of the the last 30% of the wetted distance. On the pressure surface, however, an empirical correlation of the form

$$Nu = \text{const. } Re^{0.8} \left[ 1 + a_1 Re_{n1} \left( \frac{x}{s} \right)^{0.8} \right]$$

(2.31)
yielded better results in collapsing the data. In Equation 2.31, \( a_1 \) and \( n_1 \) are experimentally determined constants and \( x/s \) is the fractional wetted distance. This expression is similar to the flat plate correlation with the addition of an enhancement term in the square brackets. It was argued that on the concave pressure surface, shear instabilities may have produced Taylor-Görtler vorticity. This vorticity in turn increased the mixing in the boundary layer and caused enhancements in heat transfer.
2.5 Uncertainty Analysis

2.5.1 Heat Transfer Measurement Uncertainty

Nickel Blades. In the uncertainty analysis, the individual and combined effects of several variables on the measured heat transfer coefficients are shown. This error analysis is conducted on one blade of load case No. 8. (see Table 2.2 on page 23). The parameters considered to affect the measurement of heat transfer coefficients are the thermal diffusivity \( \alpha \) and conductivity \( k \) of the blade material, the time constant \( \tau \) of the thermocouples, and the emissivity \( \varepsilon \) of the blade surface. The effects of a 10% change in the nominal value of each parameter are summarized in Table 2.7.

Table 2.7: Percentage change in heat transfer upon 10% change of input variables of nickel blades.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Nominal</th>
<th>Change, %</th>
<th>( \Delta \text{Nu}, % )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \alpha (m^2/s) )</td>
<td>( 4 \times 10^{-6} )</td>
<td>+10</td>
<td>-4.3</td>
</tr>
<tr>
<td>( k (W/m/K) )</td>
<td>20</td>
<td>+10</td>
<td>+10.1</td>
</tr>
<tr>
<td>( \tau (s) )</td>
<td>2.5</td>
<td>+10</td>
<td>-9.7</td>
</tr>
<tr>
<td>( \varepsilon )</td>
<td>0.8</td>
<td>+10</td>
<td>-2.0</td>
</tr>
</tbody>
</table>

It is seen that the thermal conductivity and the thermocouple time constant have the strongest influence on the measured heat transfer, with respective changes to the Nusselt number of 10.1% and -9.7%. Modifying the emissivity by 10% changes the Nusselt number by only 2%.

In the uncertainty analysis, special attention is given to the surface emissivity. As mentioned earlier, this parameter highly affects the absolute accuracy of infrared surface temperature measurements. Large errors were reported in measuring the surface emissivity [107]. Also, the emissivity can change as the surface conditions of the blades alter. Figure 2.40 shows the midspan Nusselt number distribution for three values of the surface emissivity. The three emissivity values, 0.8 ± 0.05, are typical for nickel alloy blades. Generally, the assumption of a higher emissivity value in the data reduction procedure causes the processed absolute surface temperature to decrease. As Figure 2.40 shows, the effect of the emissivity variation on the Nusselt number is small,
2.5 Uncertainty Analysis

Figure 2.40: Effect of emissivity variation on measurement of Nusselt numbers.

in this case on the order of 1.3%. This value is lower than those reported by Boyle [19], who measured a variation in rough vane Nusselt numbers of approximately 5% on the same variation of surface emissivity. This difference is not surprising, since Boyle relied on the absolute surface temperature data, whereas in the present transient technique, the temperature variation is used to determined the heat transfer.

The combined effects of these parameters are examined using a Monte Carlo simulation. A standard uncertainty is assigned to the parameters, Table 2.8. For the thermal diffusivity and conductivity a standard deviation of 10% is applied. This value is typical for a thermal calibration in a laboratory experiment [62]. The error in the thermocouple time constant of ±0.1 seconds is experimentally determined. However, since the absolute value of the time constant also accounts for the estimated turbulence level in the combustion chamber, a more conservative value for the standard deviation of ±0.2 is used. According to White [107], estimates of emissivity are rarely better than 10%. In the emissivity measurements that are discussed in Chapter 6, a mean uncertainty of 16% is found. Therefore, a value of 16% is used, which corresponds to an emissivity uncertainty of ±0.13. This relatively large value is also considered to include the effects of errors in apparent emissivity. The main sources of reflected radiation in a turbine are the combustors and nozzle guide vanes. For a first stage rotor blade, DeLucia et al. [28] reported an
Table 2.8: Input variables standard deviation for the Monte Carlo simulation of the nickel blades.

<table>
<thead>
<tr>
<th>Variable</th>
<th>$\sigma$</th>
<th>% of nominal</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$ (m$^2$/s)</td>
<td>$4 \times 10^{-7}$</td>
<td>$\pm 10$</td>
</tr>
<tr>
<td>$k$ (W/m/K)</td>
<td>$\pm 2$</td>
<td>$\pm 10$</td>
</tr>
<tr>
<td>$\tau$ (s)</td>
<td>$\pm 0.25$</td>
<td>$\pm 10$</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>$0.13$</td>
<td>$16$</td>
</tr>
</tbody>
</table>

increase in emissivity of up to 20% due to apparent emissivity effects. On the second stage rotor blade, however, these effects are considerably lower. Thus, a combined emissivity uncertainty of $\pm 0.13$ is deemed to be appropriate.

In the Monte Carlo method, a value within the assigned error band is randomly selected for every one of the input parameters. For these random values, a normal distribution is applied. The heat transfer coefficient is computed on one representative blade for a population of 1000 input quantity groups and the resulting total error distribution of the heat transfer coefficient is determined.

![Figure 2.41: Histogram of the Nusselt number distribution from the Monte Carlo analysis at 34% wetted distance.](image)

A typical histogram of the Nusselt number distribution from the Monte Carlo
2.5 Uncertainty Analysis

analysis is shown in Figure 2.41 for the pressure side location at 34% wetted distance. The distribution is divided into 12 classes and the Nusselt numbers are scaled by the turbulent flat plate Nusselt number. The resulting probability distribution shows a Gaussian behavior. This result is as expected, since the input parameters also have a normal distribution.

Figure 2.42 summarizes the Monte Carlo analysis for the heat transfer coefficient. While the symbols show the Nusselt number distribution with the nominal parameter values, the dashed lines represent one standard deviation. The percentage error in Nusselt number is approximately constant along the wetted distance of the blade and has a mean value of ±15.8%.

Figure 2.42 also shows that the observed variation in heat transfer between different blades is always smaller than the measurement uncertainty.

TBC Blades. For the coated blades, an analogous uncertainty analysis is performed. The Monte Carlo simulation is applied to one blade of the TBC coated blade set for the same load case as for the uncoated nickel blades. The individual influence of the input variables on the heat transfer measurement error is shown in Table 2.9. The assigned error band for all the variables was 10%. The nominal value of the TBC top-coat thickness in Table 2.9 is
Table 2.9: Percentage change in heat transfer upon 10% change of input variables of TBC blades.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Nominal</th>
<th>Change, %</th>
<th>ΔNu, %</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha_{TBC}$ $(m^2/s)$</td>
<td>$5 \times 10^{-7}$</td>
<td>+10</td>
<td>-1.8</td>
</tr>
<tr>
<td>$k_{TBC}$ $(W/m/K)$</td>
<td>2</td>
<td>+10</td>
<td>+8.4</td>
</tr>
<tr>
<td>$x_i$</td>
<td>1</td>
<td>+10</td>
<td>-4.0</td>
</tr>
<tr>
<td>$\alpha_{Ni}$ $(m^2/s)$</td>
<td>$4 \times 10^{-6}$</td>
<td>+10</td>
<td>-0.6</td>
</tr>
<tr>
<td>$k_{Ni}$ $(W/m/K)$</td>
<td>20</td>
<td>+10</td>
<td>+1.4</td>
</tr>
<tr>
<td>$\tau$ $(s)$</td>
<td>2.5</td>
<td>+10</td>
<td>-10.8</td>
</tr>
<tr>
<td>$\epsilon$</td>
<td>0.8</td>
<td>+10</td>
<td>-2.1</td>
</tr>
</tbody>
</table>

normalized by the average top-coat thickness on the pressure surface of the blade.

As seen from Table 2.9, the strongest impacting factors on the measurement uncertainty for the TBC blades are the thermal conductivity of the ceramic top-coat and the time constant of the thermocouples. This result reflects the same trend found in the nickel blades, where the thermal conductivity of the blade wall and the time constant were the factors that were most influential on the measurement uncertainty.

To study the combined effect of the input variables, an error band of 10% is applied to all the variables, with the exception of the emissivity value. For this value, the error band applied is 16%. The mean error found for the TBC blades is 15.7%, which is slightly lower than for the uncoated blades.

In the author’s opinion, the potential future uncertainty of the proposed measurement technique could be close to ±10%. This minimized value could be achieved by assessing the following with greater accuracy: The material properties of the blade, the thermocouple time constant and the surface emissivity. The application of higher response thermocouples for the measurement of the reference gas temperature would also help to reduce the measurement uncertainty.
2.5.2 Surface Temperature Measurement Noise

In this section, the noise level of the blade surface temperature measurements using an infrared pyrometer is assessed. Firstly, the white noise level of the pyrometer is measured at typical engine temperatures using a calibration furnace. Secondly, the in-engine measurement noise is determined by investigating data segments of nearly constant temperature from the engine experiments.

Table 2.10: Pyrometer white noise standard deviation from calibration furnace measurements.

<table>
<thead>
<tr>
<th>Blackbody temperature (°C)</th>
<th>800</th>
<th>900</th>
<th>1000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Noise standard deviation (K)</td>
<td>0.012</td>
<td>0.004</td>
<td>0.002</td>
</tr>
</tbody>
</table>

Calibration Furnace. For the measurement of the pyrometry system's white noise, the calibration furnace containing a blackbody radiator is set to a constant temperature. The blackbody temperature is measured using the single point pyrometer. The acquisition frequency of the pyrometer is 100 kHz and its corresponding resolution 0.1 K. Data is recorded through periods of 30 seconds and the standard deviation of the measurement sample from its mean value is determined. The standard deviation of the sample is defined by Equation 2.32.

\[
y = \sqrt{\frac{1}{n-1} \sum_{i=1}^{n} (x_i - \bar{x})^2}
\]  

(2.32)

The average value of the sample is given by Equation 2.33.

\[
\bar{x} = \frac{1}{n} \sum_{i=1}^{n} x_i
\]  

(2.33)

Table 2.10 shows the resulting standard deviation for different blackbody temperatures. In the steady state case, the noise level is low. As expected, the standard deviation decreases with increasing blackbody temperature. This decrease is attributed to the increase in radiation intensity with temperature as described by Planck’s law (Equation 2.10 on page 39).
In-Engine Measurement. In the engine experiment, the pyrometer records temperatures on rotating blades. From the continuous data record, temperature-time histories are extracted for specific blade locations using a key phaser signal. Measuring a stationary object in the engine, the measurement noise comprises of in-engine effects and the noise originating from the measurement system. In addition to these error sources, noise is introduced by the procedure that generates the temperature-time histories for specific locations on the rotating blades. The following sources for in-engine measurement noise are identified:

- Spatial averaging: The size and shape of the defined area of the pyrometer depends on the measured location on the blade surface. As described in Section 2.1.3, the sizes of the viewing areas range from 3 to 16 mm and are of circular or elliptical shape.

- Phase locking: For the generation of a temperature transient for a specific blade location, the key phaser signal is used. The procedure extracts temperature data from the continuous data record at the phase-locked position on the rotor. The measurement noise depends on the quality of the phase-locking procedure.

- The pyrometer system’s white noise. As shown above, at the resolution of 0.1 K, this source of error is small.

- Real engine effects such as radiating objects in the hot-gas path or lens contamination.

The combined effects of the above described sources of error are studied. Sequences of constant blade temperature (Figure 2.43) at different temperature levels are analyzed in order to determine the standard deviation of the measurement noise. The surface temperature signal is generated from a specific location on a single blade using the phase-locking procedure described in Section 2.2.2. Thus, it is a 50 Hz signal. The moving average of this signal is computed and illustrated in Figure 2.43. This curve represents the underlying temperature response of the blade surface. The standard deviation of the temperature history from the respective moving average is calculated. Different window sizes for the computation of the moving average were tested. The results of these tests are shown in Figure 2.44. Consequently, a window size of 1 second is used.

In the above analysis, the mean temperature history is determined by using
2.5 Uncertainty Analysis

a moving average procedure on a single blade. As an alternative, the mean temperature history could be calculated by averaging the original temperature history of a number of blades. However, such an approach would suppress any effect of the phase-locking on the measurement noise, since the phase-locking of all the blades is based on the same key phaser signal.

The procedure of determining the standard deviation of the measurement noise is applied to every measurement location on the blades for both pyrometer channels. The noise level is determined for different engine operating conditions (different surface temperatures). For the uncoated and coated blades of load case No. 2, the results are shown in Figures 2.45 and 2.46, respectively. (The results for the load cases No. 8 and 13 are included in Appendix A). The measured surface temperature distribution is compared with the noise level and the Nusselt number distribution. While the open symbols show the average value over the blade set, the error bars represent blade-to-blade variation. Comparing the blade surface temperatures and the Nusselt number distributions at different temperatures for coated and uncoated blades, the following observations are made:

- For both the coated and the uncoated blades, the blade-to-blade variation in Nusselt number scales with the blade-to-blade variation observed in the surface temperature measurement. This result is as expected,
Figure 2.44: Standard deviation of the measurement noise for one blade location depending on the window size used to calculated the average temperature.

since it follows the trend that high heat transfer rates cause high surface temperatures. In the data reduction process that was described in Section 2.2.5, the semi-infinite solid assumption is made for the one-dimensional heat conduction model. The fact that variations in surface temperature and heat transfer show equal trends seems to corroborate that assumption.

- On the uncoated blades, blade-to-blade variations in heat transfer and surface temperature are higher on the pressure surface than on the suction surface. As described in Section 2.4, a flow separation seems to occur on the blade pressure surface. The sensitive behavior of the separation bubble might be the cause for higher pressure surface blade-to-blade variation.

- On the the TBC coated blades, measured blade-to-blade variation is higher on the suction surface. Here it is assumed that the flow on the pressure surface immediately becomes turbulent after the leading edge area. Thus, stable flow behavior is expected. The high blade-to-blade variation on the suction surface may be explained by the strong wake interaction that occurs in this region of the blade.

- As seen from Figures 2.45 and 2.46, the pyrometer noise level (standard
deviation from the time mean at steady state conditions) is high where the temperature gradients on the blade surface are high. This behavior seems to reflect the influence of the phase-locking of individual blade locations. An error in the determination of the blade location leads to a higher temperature error in a region of high thermal gradient (along the blade surface) than in a region of low thermal gradient. Thus, in the leading edge region, the standard deviation is high compared with the downstream part of the pressure surface, where thermal gradients generally are low.

- Table 2.11 summarizes the pyrometer standard deviation of the load cases considered. The mean value of the noise level distribution along the wetted distance of the blade is listed for coated and uncoated blades. The coated blades show a higher average standard deviation deviation than the uncoated blades. At the same time, the temperature distribution seems to be more isothermal on the coated blades. Thus, this difference can not be explained by the effects of the phase-locking procedure.

- The comparison of the pyrometer standard deviations at different engine loads (different temperatures) does not yield a clear trend. The coldest case shows higher pyrometer standard deviation than the next warmer case, for both coated and uncoated blades. Two different conditions affect the measurement accuracy: Firstly, the accuracy of infrared measurements increases with rising temperature because of the general increase in signal-to-noise ratio. Secondly, as shown above, the temperature gradients along the blade surface increase at higher temperatures. This in turn yields high noise in the generation of temperature-time histories due to the errors introduced in the phase-locking of measurement locations.

### Table 2.11: Average blade surface temperature noise levels (K).

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Uncoated</th>
<th>Coated</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>0.229</td>
<td>0.353</td>
</tr>
<tr>
<td>8</td>
<td>0.183</td>
<td>0.226</td>
</tr>
<tr>
<td>13</td>
<td>0.266</td>
<td>0.272</td>
</tr>
</tbody>
</table>
Thus, the following two conclusions can be drawn from the above analysis: Firstly, on the uncoated blades, the mean standard deviation in the surface temperature measurement noise is 0.23 K. Secondly, on the TBC coated blades, the noise standard deviation is slightly higher and has a mean value of 0.28 K.
Figure 2.45: Top: Blade-to-blade variation in surface temperature; Center: Noise level (standard deviation from moving average); Bottom: Nusselt number distribution versus wetted distance. Uncoated blades of load case No. 2 (245 MW).
Figure 2.46: Top: Blade-to-blade variation in surface temperature; Center: Noise level; Bottom: Nusselt number distribution versus wetted distance. TBC coated blades, load case No. 2 (245 MW).
Chapter 3

CFD Predicted Heat Transfer

3.1 Introduction

In Chapter 2, the heat transfer distributions on coated and uncoated rotating blades were measured. This chapter presents the results of the respective three-dimensional, steady state Navier-Stokes simulations of the rotor blade flow. From the simulation results, heat transfer distributions are derived for a subsequent comparison with the experimental data.

For all the computations described in this chapter, the flow solver MBStage3D was used [7]. This code is a three-dimensional, structured Navier-Stokes solver developed for multistage turbomachinery applications. The time marching algorithm used in MBStage3D is a Jameson-type algorithm [52, 98] with a residual-averaging technique for improved stability. The time discretization is accomplished by a five-stage Runge-Kutta technique, which is fourth-order accurate. All computations were conducted with the algebraic Baldwin-Lomax turbulence model [11] together with a logarithmic wall function developed by Sommer [93], to compute the turbulent viscosity at the wall.

The post processing necessary to gain an understanding of the flow physics was achieved through three-dimensional and scalar investigations of the flow fields of interest. The 3D visualization was created using TECPLOT. Many routines used for the post processing were developed by members of LSM’s
3.2 CFD Load Cases

In the CFD study, the flow solutions of six different load cases are computed. The six cases reflect the different engine load settings of the experimental program. Table 3.1 summarizes some salient parameters of the CFD cases. The engine power, $P_{net}$, and the average surface temperature, $T_{wall}$, are derived from the experiments. The corresponding Reynolds number was determined by the mean line analysis described in Section 2.3. Air is assumed as the working fluid and the value of its dynamic viscosity, $\mu$, is evaluated at the temperature of the trailing edge station. The dynamic viscosities are determined using tables of gas properties [81].

<table>
<thead>
<tr>
<th>No.</th>
<th>$P_{net}$ (MW)</th>
<th>$Re_C/Re_{C_0}$</th>
<th>$T_{wall}$ (K)</th>
<th>$\mu \times 10^5$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>275</td>
<td>1</td>
<td>1148</td>
<td>4.61</td>
</tr>
<tr>
<td>2</td>
<td>245</td>
<td>0.88</td>
<td>1145</td>
<td>4.58</td>
</tr>
<tr>
<td>3</td>
<td>170</td>
<td>0.72</td>
<td>1113</td>
<td>4.47</td>
</tr>
<tr>
<td>4</td>
<td>140</td>
<td>0.68</td>
<td>1083</td>
<td>4.36</td>
</tr>
<tr>
<td>5</td>
<td>100</td>
<td>0.61</td>
<td>1043</td>
<td>4.26</td>
</tr>
<tr>
<td>6</td>
<td>70</td>
<td>0.55</td>
<td>1004</td>
<td>4.16</td>
</tr>
</tbody>
</table>

3.3 Computational Domain

The computational domain for the second low-pressure turbine blade (LPTB2) models a single pitch of one rotor blade row. Although the LPTB2 blade has a half shroud, a full shroud is assumed in the simulation. No leakage or cooling flows are modeled. Two different H grids are used for the computations. The first grid is relatively coarse and has 48x116x64 cells in circumferential, axial and radial directions, respectively. The second finer grid has 88 instead
3.3 Computational Domain

Figure 3.1: Fine straight H-grid used for the CFD calculation of the GT26 LPT2 blade. Distorted cells cause over-predicted heat transfer rates at the leading edge.

of 48 pitch-wise cells. Thus, the two grids have 356’352 and 653’312 cells, respectively. A section of the grid at blade midspan is shown in Figure 3.1. In order to correctly resolve the temperature gradient at the wall, sufficient grid resolution is requisite in the boundary layer. Small $y+$ values are required, where $y+$ is defined by Equation 3.1.

$$y^+ = \frac{y\sqrt{\tau_0/\rho}}{\nu}$$ (3.1)

Here, $y$ is the distance from the wall, $\tau_0$ the wall shear stress and $\rho$ and $\nu$ the density and viscosity of the fluid, respectively. The grid generator used (Bladegrid V1.6) allows the specification of the grid step ratio, which then enables the application of smaller cell spacing close to the blade surface. In
the center of the blade passage, the grid becomes more coarse. Thus with the finer grid, the mean $y+$ values on the pressure and suction surfaces of case No. 3 are 6.1 and 7.1, respectively. Whereas with the coarse grid, the mean $y+$ values are 79.7 and 99.7, see Figure 3.2.

3.4 Boundary Conditions

For the six cases considered in the CFD study, inlet and outlet boundary conditions are specified. At the inlet of the computational domain, the absolute total temperature and pressure profiles are set. The hub-to-tip profiles are shown in Figure 3.3 in non-dimensional form. Also, the pitch and yaw angles in the absolute frame of reference are given at the inlet boundary of the computational domain. The pitch angle over the blade span is shown in Figure 3.3c.

At the outlet of the computational domain, MBStage3D offers several options to specify the exit pressure boundary condition. The default setting is to apply the radial equilibrium with the static pressure fixed at the turbine casing. In these computations, however, the exit static pressure is defined by the profile shown in Figure 3.3d. All the boundary conditions were provided.
3.4 Boundary Conditions

For the calculation of the isothermal blade flow, the wall temperature is set as an additional boundary condition. The pressure and suction surface temperatures are specified to be the respective average surface temperatures measured in the experiments. In the absence of experimental data, the endwall surface temperatures are specified to be the same as those on the blade pressure surface.

For the specification of the fluid properties used in the simulation, different options are given in the flow solver. The fluid can be modeled as a perfect, ideal or real gas. For the perfect gas assumption, the specific heat at constant
pressure, $c_p$, is kept constant throughout the computation. With the ideal gas option, polynomial functions are used to adjust the specific heat with changing temperature. Thus, $c_p = f(T)$. If the fluid is modeled as a real gas, gas tables instead of polynomial functions are used to account for temperature changes.

The computations described in this chapter assume perfect gas behavior with air as the working fluid. The dynamic viscosity of the gas is evaluated at the temperature of the blade trailing edge station. The physical properties of case No. 3 are summarized in Table 3.2.

With this set of boundary conditions and fluid properties, the CFD problem is completely defined and can be solved by the numerical solver.

### 3.5 Wall Functions

The wall functions used in the present study were developed by Sommer [93]. The wall functions apply the logarithmic law of the wall and a blending function for the viscous sublayer. These are scaling wall functions, i.e. they are valid throughout the entire range of $y+$ values from 0 to 500. The heat transfer rate at the wall is determined from the enthalpy gradient in the near-wall region and then scaled by a factor from the wall functions. For grids with small $y+$ values, this scaling factor converges toward unity. In this case, the correct values from the turbulence model are effectively used.

In a recent work, Mokulys [68] used the MBStage3D solver to study the influence of different grid types on the prediction of heat transfer. The wall functions by Sommer were used together with the turbulence model of Spalart
3.6 Flow Results

In this section, the results of the CFD analyses are presented. Predicted heat transfer distributions along the midspan of the blade for the coarse and fine computational grid are shown. Full surface distributions of static pressure and predicted heat transfer coefficients are given for the fine computational grid. Depending on the grid size, typical run times for these CFD cases ranged from 24 to 72 hours until convergence was achieved. A typical Stage3D control file is shown Appendix B. For every load step, an initial guess from the pre-processor (Stagelink) was used as an input for Stage3D. This pre-processor determines an initial guess of the flow field using a coarsened version of the original computational grid.

A typical convergence history of case No. 3 (see Table 3.1 in Section 3.2) is shown in Figure 3.4. As seen, the residual decreases about one order of magnitude until convergence is achieved. Although usually a reduction of the mean residuals of approximately two orders of magnitudes is desired, a smaller decrease is found here. This relatively small reduction is caused by an unsteady mode found at the trailing edge of the blade. Therefore, an oscillating flow solution is found. In heat transfer simulations, it may be the case that the aerodynamic field looks converged although the thermal field has
not yet completely converged. Therefore, an additional approach is chosen to check the convergence of the thermal field. The heat transfer rates are examined every 500 iterations and the simulation is stopped when no further change in heat transfer is observed. For the present load cases, fully converged solutions of the thermal field are reached after 15'000 to 20'000 iterations.
The percentage error in massflow of the CFD case No. 3 is shown in Figure 3.5. The massflow error is computed according to Equation 3.2.

\[ \Delta \dot{m} = \frac{\dot{m}_{\text{outlet}} - \dot{m}_{\text{inlet}}}{\dot{m}_{\text{inlet}}} \]  

(3.2)

Over the first few hundred iterations, the error is reduced from approximately 25% to below 5%. When the calculation has fully converged, a constant error in massflow of 0.6% is observed.

The predicted Nusselt number distributions are based on the simulated isothermal blade flow. The calculated heat flux \( \dot{q} \), the isothermal wall temperature \( T_w \) and the relative total inlet temperature at midspan \( T_{\text{rel}} \) are then combined to yield the heat transfer coefficient according to Equation 3.3.

\[ h = \frac{\dot{q}}{T_{\text{rel}} - T_w} \]  

(3.3)

The heat transfer distributions from the CFD prediction for case No. 3 are shown in Figure 3.6 for the coarse and the fine grid. The Nusselt numbers shown are normalized by the turbulent flat plate Nusselt number, as described in Chapter 2. As is typical of straight H-grids, high Nusselt numbers are ob-
served at the leading edge of the blade. The high Nusselt numbers are caused by distorted cells at the leading edge that are unavoidable when straight H-grids are used (see Figure 3.1). As seen in Figure 3.6, the Nusselt number distributions of the fine and coarse grids are almost identical on the blade pressure surface. On the suction surface, however, higher heat transfer rates are predicted by the fine grid. Since the $y^+$ values of the fine grid are one order of magnitude lower than those of the coarse grid, the fine computational grid is used in the following computations. As described in Section 3.5, the heat transfer predictions are considered to have an accuracy of ±20%.

Figure 3.7 shows the stream traces in the midspan region of the blade. The flow is attached to the wall along the entire blade surface. In Section 2, the heat transfer distribution was shown at the blade midspan. It was argued that a drop and subsequent increase in the heat transfer rate possibly indicated the existence of a laminar separation bubble on the uncoated blades. However, the flow field that is shown in Figure 3.7 does not show any flow separation. This is as expected, since a turbulent boundary layer is assumed on the entire blade in the CFD calculation. A small steady recirculation is found downstream of the square trailing edge, where in reality there will be unsteady eddy shedding. The square trailing edge geometry in the CFD model is also existent on the real blade, where a slot along the span is used for coolant ejection.

Figure 3.8 shows a contour plot of the static pressure distribution on the airfoil of load case No. 3. The pressure is given in bar. Clearly visible are the high pressure regions at the leading edge (LE) of the blade as well as on the pressure surface. The radial pressure distribution is close to uniform, with some slight variation observed on the pressure surface of the blade.

Figure 3.9 shows a full surface predicted Nusselt number distribution for load case No. 3. High values of heat transfer are observed in the stagnation region at the leading edge (LE) of the blade. On the upstream portion of the blade, the Nusselt numbers are higher on the suction surface than on the pressure surface (see also Figure 3.10). Towards the trailing edge (TE), increasing heat transfer is observed on the pressure surface, while on the suction surface, the heat transfer levels remain approximately constant. In the experimental part of this study, the midspan heat transfer coefficients were measured. Hence in the CFD work, the evaluation of the heat transfer rates also concentrated on the midspan region of the blade. For this reason, no tip gap was included in the CFD model. A full shroud was assumed, where in reality the blade
3.6 Flow Results

Figure 3.7: Stream traces from the CFD prediction in the midspan of the second LPT blade passage.

Figure 3.8: Predicted static pressure contours on airfoil pressure and suction surfaces for load case No. 3. Pressure is given in bar.
features a half shroud. Therefore, the tip leakage flows, which highly affect the heat transfer rates at the blade tip, are not modeled in the CFD. Thus, the predicted tip heat transfer rates (Figure 3.9) are not considered to reflect those encountered in the engine.

In Figure 3.10, a comparison of the heat flux variation along the midspan is shown for all the CFD cases. As expected, cases of high Reynolds number have high heat transfer rates. The established effect of Reynolds number on heat transfer is also shown in Figure 3.11, where scaled Nusselt numbers at 30% wetted distance are plotted versus the Reynolds number of the flow. According to Lakshminarayana [59], the heat transfer coefficient of a fully turbulent boundary layer scales as $Re^{0.8}$. Thus, this correlation is also indicated in Figure 3.11 by the dashed line. As seen, the agreement between this correlation and the predicted heat transfer rates is good.

Figure 3.12 shows the effect of incidence angle on the blade loading for the CFD cases No. 2, 4 and 6. The isentropic Mach number is calculated from the ratio of the relative total pressure at the inlet to the static pressure, Equation 3.4. The Mach numbers are scaled by the trailing edge (TE) value of case No. 1.
3.6 Flow Results

### Figure 3.10: Predicted heat flux distribution versus wetted distance at the blade midspan for different load cases.

![Graph showing predicted heat flux distribution](image)

\[
M^2 = \frac{2 \left[ \left( \frac{P_{\text{rel}}}{P_s} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]}{\gamma - 1} \tag{3.4}
\]

Close to the leading edge, the incidence angle has a strong influence on the acceleration of the flow. With increasingly positive incidence, a peak develops on the suction surface of the blade. This peak causes a diffusion of the flow at about 10% wetted distance for the high Reynolds number cases, which according to the mean line analysis have incidence angles close to zero. On the pressure surface, the opposite effect is observed. With increasingly positive incidence, the peak at 10% wetted distance of the pressure surface is reduced. However, the subsequent diffusion is very strong for all the cases. In Section 2.4.1, this diffusion was identified to cause a separation bubble on the upstream part of the pressure surface. On the downstream part of the blade, the effect of the incidence angle is small. The same pressure and suction surface behavior was reported by Jouini et al. [54], who measured the aerodynamic performance of a transonic turbine cascade at off-design conditions.
3.7 Comparison of CFD and Experiment

For the CFD case No. 3 (170 MW), the experimentally determined Nusselt number distributions from coated and uncoated blades are compared with the
CFD predicted Nusselt numbers in Figure 3.13. The CFD predicted Nusselt numbers are calculated as described in the previous sections, based on the fine computational grid.

As seen in Figure 3.13, the predicted Nusselt numbers agree well with the experiment on the blade suction side of the uncoated blades. In the upstream portion of the suction surface, the flow is laminar. However, strong wake interactions occur in this area. While this interaction does not affect the time-averaged heat load to the blade considerably, it causes the laminar boundary layer to become turbulent [3]. This may be the reason for the accurate suction side prediction by the CFD calculation, which assumes a turbulent boundary layer along the entire blade surface. The heat transfer rates on the suction surface of the TBC blades are lower than those of the uncoated blades. As described in Section 2.4.2, these lower Nusselt numbers may be caused by a slightly negative incidence angle due to increased blockage on the TBC blades.

On the blade pressure surface near the leading edge, the boundary layer develops on a convex surface. This causes the Nusselt numbers to decrease on both the coated and uncoated blades. On the uncoated blades after the leading edge region, the experimental Nusselt number distributions first show a drop and then a subsequent increase in Nusselt number. This may indicate
the formation of a separation bubble, which is usually observed when the surface curvature changes from convex to concave (see Section 2.4.1). The reattachment of the flow induces transition followed by an increase in heat transfer. However, the separation bubble is not observed in the CFD calculation. The heat transfer rates in the experiment may be enhanced by: The influence of free stream turbulence, surface roughness and the concave surface curvature. The effects of these phenomena are not modeled in the CFD code and could explain the underpredicted Nusselt numbers downstream of 20% wetted distance on the pressure surface.

On the TBC blades, no indication of flow separation is found on the pressure surface of the blade. In Section 2.4.1 it is hypothesized that the TBC layer which starts near the leading edge may trip the boundary layer and cause an early transition to turbulent flow. Hence, the measured heat transfer distribution on the TBC blades agrees better with the CFD turbulent prediction than that on the uncoated blades. This behavior is clearly shown in Figure 3.13. However, downstream of 30% wetted distance, the experimental Nusselt numbers of the coated blades are also under-predicted by the CFD prediction.

Dunn et al. [35] observed a similar behavior in a study on time-averaged heat transfer to rotating turbine blades. They measured Stanton numbers in a shock tunnel facility and compared the results with Navier-Stokes predictions. With the smooth heat transfer prediction at the high Reynolds number, the numerical results underpredicted the heat transfer on the pressure surface. On the suction surface, the heat transfer was over-predicted. The same trend was shown by Nealy et al. [75] on the pressure surface of a cascade blade.

Figure 3.14 shows the measured and predicted Nusselt numbers versus Reynolds number on the suction and pressure surfaces of the uncoated blades. As also seen from the single load case illustrated by Figure 3.13, good agreement is found between the predicted and the measured Nusselt numbers on the suction surface (Figure 3.14a). On the pressure surface, however, the CFD under-predicts the measurement quite strongly (Figure 3.14b). Also, the under-prediction is more pronounced at high Reynolds numbers.

Another source of the discrepancy between the CFD and experiments may be attributed to the differences in the temperature flow fields. In the gas turbine, the temperature field around the blade is non-uniform, as the flow along the blade pressure surface is hotter than along the suction surface. However,
3.7 Comparison of CFD and Experiment

Figure 3.14: CFD predicted and measured Nusselt numbers versus scaled Reynolds number on the suction and pressure surface of the uncoated blades. Error bars indicate measured blade-to-blade variation.

A uniform temperature distribution is assumed in the determination of the heat transfer coefficient. Since for that process, only the relative temperature change is used, the effect of non-uniform driving gas temperature distribution is considered to be small.
Chapter 4

Gas Recovery
Temperature Measurement

4.1 Introduction

According to a study by Rose [82], the single most important parameters to predict the metal temperatures of turbine components are the convective heat transfer coefficient and the local adiabatic gas recovery temperature. In the preceding chapters, the measurements of in-engine heat transfer coefficient in the midspan region of rotating blades were described and compared with CFD predictions. The present chapter addresses the measurement of the adiabatic gas recovery temperature.

Measurements of the adiabatic wall temperature (recovery temperature) are commonly conducted in film cooling heat transfer studies. The performance of film cooling schemes is given in terms of adiabatic effectiveness. The adiabatic wall temperature is part of the definition of the adiabatic effectiveness. In this field of research, various scientists have measured the adiabatic wall temperature, e.g. Burd [24], Van Teuren [104], Chambers [25]. Burd measured the local adiabatic wall temperature using a traversing thermocouple by extrapolation of the near-wall fluid temperature profile to the wall. Van Teuren and Chambers used liquid crystals to measure the adiabatic wall temperature. Apart from these applications in film cooling test facilities, no references to measurements of the adiabatic wall temperature were to be found in the open literature. To date, the measurement of the adiabatic wall temperature in a
rotating machine has not yet been attempted.

This chapter presents a novel method of determining the recovery temperature in an operating engine. The method uses the measured surface temperatures from steady state as well as the measured heat transfer coefficients that were presented in Section 2.4. The measurement technique is based on matching the conductive heat flux to the convective heat flux at the surface of a set of blades. This matching is achieved by means of a least-squares fitting procedure.

The present in-situ measurement method for the local adiabatic gas recovery temperature is a novel concept. As will be shown in the present chapter, this in-situ measurement method currently still has some limitations but also promises high potential for further development. Although the presented data show relatively large errors, to date, they are the first ones to be acquired on rotating blades of an operating engine.

4.2 Theory

In a high velocity thermal boundary layer, heat is transferred from the fluid to the wall if the adiabatic wall temperature $T_{aw}$ is higher than the body temperature $T_0$. For a fluid with Pr=1, the adiabatic wall temperature is defined by Equation 4.1

$$T_{aw} = T_\infty + \frac{u_\infty^2}{2c_p}.$$  \hspace{1cm} (4.1)

where $T_\infty$ and $u_\infty$ are local values of the static temperature and velocity of the free-stream, respectively. It reflects the influence of high-velocity viscous dissipation on the convective heat transfer. As a consequence of the dissipation in the boundary layer, a thermal boundary layer forms on a body, even if there is no heat transfer between the body and the fluid.

For a fluid with Pr$\neq$1, the adiabatic wall temperature is defined by means of the recovery factor, $r_c$, as

$$T_{aw} = T_\infty + r_c \frac{u_\infty^2}{2c_p}.$$  \hspace{1cm} (4.2)
4.2 Theory

For $Pr=1$, the recovery factor equals unity and Equation 4.2 reduces to Equation 4.1. The adiabatic wall temperature then equals the total temperature.

Measurements of the recovery factor have shown that, to a good approximation, it can be defined as

$$r_c \approx Pr^{1/2}. \quad (4.3)$$

From Equation 4.2, the recovery factor can be rewritten as

$$r_c = \frac{T_{aw} - T_\infty}{u_\infty^2/2c_p}. \quad (4.4)$$

The dimensionless temperature increase of an adiabatic wall as a consequence of dissipation given by Equation 4.4 is also called the recovery factor, since the denominator of Equation 4.4 is the temperature increase due to adiabatic compression of an ideal gas [86]. Thus, the adiabatic wall temperature is often called the adiabatic recovery temperature.
Figure 4.1 shows the temperature distribution in the high-velocity boundary layer for a fluid with \( \text{Pr}=1 \) [55]. In this figure, \( t^*_\infty \) denotes the total temperature of the free stream, \( t^* \) the total temperature in the boundary layer and \( t_\infty \) the static temperature. In the adiabatic case \( (q_0''=0) \), the wall temperature equals the free stream total temperature \( (\text{Pr}=1) \). If the wall temperature is above the adiabatic wall temperature, heat is transferred from the wall to the fluid.

### 4.3 Analytical Method

For the measurement of the recovery temperature, the blade surface temperature data acquired at constant engine load (steady state) are used. In addition to the steady state surface temperatures, the previously measured heat transfer coefficients are also required in the method.

The determination of the recovery temperature along the midspan region of the blade is based on the calculated heat transfer at the surface due to convection and conduction, respectively. The convective heat flux is described by

\[
\dot{q}_1 = h(\text{T}_{aw} - \text{T}_w),
\]

where \( \text{T}_{aw} \) is the adiabatic wall temperature (recovery temperature), \( \text{T}_w \) the wall temperature and \( h \) the heat transfer coefficient. In Equation 4.5, \( \dot{q}_1 \) and \( \text{T}_{aw} \) are unknowns, \( \text{T}_w \) is measured in the experiment and \( h \) is determined as described in Chapter 2. Assuming one-dimensional heat transfer, the conductive heat flux through the blade wall is defined by the driving temperature and the thermal resistance given by Equation 4.6.

\[
\dot{q}_2 = \frac{\text{T}_w - \text{T}_c}{R},
\]

where \( \text{T}_c \) is the coolant temperature and \( R \) the total thermal resistance of the blade wall. The total thermal resistance is defined by

\[
R = \frac{1}{h_{ext}} + \sum_i \frac{L_i}{k_i} + \frac{1}{h_{int}},
\]
where \( h_{\text{ext}} \) and \( h_{\text{int}} \) are the convective heat transfer coefficients of the external and internal boundary layer, respectively. For the TBC coated blades, the conduction resistance on the right hand side of Equation 4.7 consists of two layers: The metal substrate and the ceramic coating. Matching the convective and conductive fluxes, \( \dot{q}_1 \) and \( \dot{q}_2 \), respectively, yields one equation for the two unknown variables, \( T_{aw} \) and \( T_c \). Solving this equation for \( n \) blades gives \( n \) equations, with \( n + 1 \) unknown variables. These unknown variables are the \( n \) coolant temperatures and the recovery temperature, which to a good approximation is the same for all the blades at a specific blade location. The number of unknowns is reduced by also assuming an equal coolant temperature for all the blades in the set. Hence, a set of \( n \) equations remains and is solved for the two unknown variables (the coolant temperature and the recovery temperature). The solution is found by minimizing Equation 4.8.

\[
SSR = \sum_{i=1}^{n} (\dot{q}_1 - \dot{q}_2)^2, \tag{4.8}
\]

where \( n \) is the number of blades that are considered. The system of equations is solved for every point on the blades' surfaces by finding the coolant temperature and the recovery temperature that minimize the respective SSR.

In the least-squares procedure, a search domain is attributed to the recovery temperature and the coolant temperature. Within this search domain, the recovery temperature and the coolant temperature that minimize the SSR are determined. However, a solution is not always found, because the minimum of the SSR is not very prominent. A criterion is established that defines whether a solution is accepted. The criterion is given by Equations 4.9 and 4.10.

\[
A_l + d_T \leq T_c \leq A_u - d_T \tag{4.9}
\]

\[
B_l + d_T \leq T_{aw} \leq B_u - d_T \tag{4.10}
\]

Here, \( A_l \) and \( B_l \) are the lower bounds of the search domain of the coolant and recovery temperature, respectively. \( A_u \) and \( B_u \) are the upper bounds. The parameter \( d_T \) is used to exclude solutions that are too close to the boundaries of the search domain.

The wall temperature that is used in Equation 4.5 and 4.6 is extracted from the surface temperature data for every load step (Table 2.2 on page 23) at
steady state. The measured external heat transfer coefficients are used as an input variable in Equation 4.5. For the internal heat transfer coefficient, a typical value at design load was provided by ALSTOM. For the off-design load cases, the internal heat transfer coefficient is scaled according to the coolant massflow measurement performed during the experiments.

Thus, the system of equations is solved for every measurement point on the blades surface for a set of TBC coated and uncoated blades. Solutions for the coolant temperature $T_c$ and the recovery temperature $T_{aw}$ are valid if they fulfill the criterion given by Equations 4.9 and 4.10.

### 4.4 Results and Discussion

In order to ascertain the accuracy of the novel measurement method, the data reduction procedure was thoroughly tested. In the least squares fitting procedure, the recovery temperature is determined from a given search domain. The method was tested by calculating the distribution of the recovery temperature of load case No. 8 (Table 2.2 on page 23), based on different input search ranges. As Figure 4.2 shows, identical results are achieved for different search ranges, which establishes the independence of the result of the search domain. The same tests were performed for the search domain of the coolant temperature, with a fixed range for the recovery temperature. For blade locations that have a solution according to the criterion of Equations 4.9 and 4.10, identical results for the recovery temperature were found.

Figure 4.3 shows the recovery temperature distribution along the pressure surface at the midspan of the blade. The results are shown for the TBC coated and uncoated blades of load case No. 9 (140 MW). The temperatures are scaled by the relative total inlet temperature derived from the measured SEV exit temperature as described in Section 2.3.

The relative total temperature of the blade flow is indicated by the dashed line in Figure 4.3. As described in Section 4.2, the relative total temperature equals the recovery temperature if the recovery factor equals unity, i.e. for fluids with Prandtl=1. For a Prandtl number of 0.7, the adiabatic wall temperature is calculated from the results of the mean line analysis (Section 2.3) using the following variables:
• The relative inlet velocity $v_r = 318 \text{ m/s}$.

• The specific heat at constant pressure $c_p = 1260 \text{ kJ/kg/K}$.

With these variables and the relative total temperature, the adiabatic wall temperature is determined and shown in Figure 4.3 by the dash-dot line. The discrepancy of the measured data to the adiabatic wall temperature based on Equation 4.2 and a Prandtl number of 0.7 is small. The average percentage difference is 2.9%.

As seen in Figure 4.3, the recovery temperature level of the TBC blades is higher than that of the uncoated blades. Due to the absence of the insulating ceramic layer, the uncoated nickel alloy blades experience a higher surface heat flux than the TBC blades. Thus, the cooling of the hot gas is greater along the nickel blades, which in turn results in a decreased gas recovery temperature (adiabatic wall temperature). This effect is demonstrated in Figure 4.3. However, only 60% of the load cases follow this trend. In some of the cases, the noise level in the resulting recovery temperature data is too high to see a clear trend. In very few cases, the reverse effect is seen, i.e. the nickel alloy blades have higher recovery temperature than the TBC blades.

In determining solutions for the recovery temperature on the suction surface, a distinct difference is observed between the nickel alloy blades and the TBC
blades. On the TBC blades, a solution for the recovery temperature is found on virtually all blade locations. On the nickel alloy blades, however, a solution is found on only a few locations on the blade surface. There is, at this point, no ready explanation for the relatively sparse number of solutions found on the suction surfaces of the nickel alloy blades.

Figure 4.4 shows the variation of the measured recovery temperature of load case No. 5 (200 MW) for the coated blades. A CFD solution of the adiabatic wall temperature distribution at blade midspan is also shown. This solution was derived from the adiabatic solution of the respective CFD simulation. (For the CFD predicted heat transfer rates of Chapter 3, the isothermal solutions of the blade flows were used.) The temperature data drop to a minimum at 20% wetted distance on the pressure surface, followed by an increase in recovery temperature. The predicted recovery temperature history also shows the same trend, although the minimum is located closer to the leading edge. On the suction surface, the measured minimum in recovery temperature is also located further downstream than as predicted by the CFD solution. As seen from the CFD solution, the temperature levels on the suction surface are lower than on the pressure surface. Such a temperature distribution is caused by the tendency of the hot fluid to drift towards the high pressure region on
the pressure surface of the blade. This trend is correctly reproduced by the measured recovery temperature data shown in Figure 4.4. Excluding the high value at the leading edge of the blade, the measured data differs from the CFD prediction by an average of -2.6%.

According to the CFD prediction, the highest values of the adiabatic wall temperature are observed at the leading edge and mid-chord of the pressure surface. In the measurement, however, exceptionally high values often occur at the leading edge, as shown in Figure 4.4. In the leading edge region, high temperature gradients are observed along the blade surface, see Figure 2.10 on page 29. Also, as the blade wall has high surface curvature in this region of the blade, the one-dimensional model used in the data reduction method may not be suitable. Inadequate modeling may be the reason for the high values that are measured at the leading edge of both the coated and uncoated blades.

Figure 4.5 shows the measured recovery temperature of all the load cases from Table 2.2 (page 23) at 40% wetted distance of the pressure surface. With the exception of the two data points at $Re_C = 1.5 \times 10^6$, the distribution of recovery temperature of both the coated and uncoated blades produces the trend that is shown by the dashed line. This line indicates the relative
Figure 4.5: Recovery temperature versus blade Reynolds number at 40% wetted distance. Dashed line shows the relative total temperature ($T_{rel}$) determined from the measured SEV exit temperature.

total inlet temperature derived from the measured SEV exit temperature. As described in Section 4.2, the relative total temperature equals the adiabatic recovery temperature if the recovery factor $r_c$ equals unity.

### 4.5 Uncertainty Analysis

The influence of all the parameters in Equation 4.8 for the determination of the recovery temperature is studied. The results of this sensitivity analysis are shown in Figure 4.6. The resulting variation of the recovery temperature is plotted versus the percentage variations of the input parameters. In the uncertainty analysis of the measurement of heat transfer coefficients (Section 2.5.1), the minor influence of the surface emissivity on the heat transfer measurement, compared with the influence of the other parameters, is demonstrated. However, as Figure 4.6 shows, the surface emissivity has the most profound influence on the measurement of the recovery temperature of all the parameters involved. This profound effect is due to the direct influence of the emissivity on the absolute value of the surface temperature, which in turn determines the driving temperature difference of the heat conduction
4.5 Uncertainty Analysis

The results of the sensitivity analysis are summarized in Table 4.1. The percentage variation in recovery temperature on a 20% change of the variables is presented. The absolute nominal values of the external and internal heat transfer coefficients and the blade wall thickness are not indicated.

For the measurement of the recovery temperature, an analytical error assessment is performed. The estimated maximum measurement error is taken as the sum of the maximum uncertainty of the individual sources. Hence, the maximum measurement error in the recovery temperature is defined by Equation 4.11.

\[
|\Delta T_{aw}| = |\Delta h_{ext}| + |\Delta h_{int}| + |\Delta L| + |\Delta k| + |\Delta \varepsilon| \tag{4.11}
\]

The individual measurement uncertainties of the variables affecting the error in the recovery temperature measurement (right hand side of Equation 4.11) are specified as follows:

- External heat transfer coefficient \(h_{ext}\): The measurement uncertainty for the external heat transfer coefficient is assessed in Section 2.5.1. In the HTC measurement, the error analysis is conducted on one blade of load case No. 8. The mean percentage uncertainty in HTC for the nickel
Table 4.1: Percentage change in recovery temperature upon 20% change of input variables.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Nominal</th>
<th>Δ, %</th>
<th>ΔT_{aw}, %</th>
</tr>
</thead>
<tbody>
<tr>
<td>(h_{ext})</td>
<td>1</td>
<td>+20</td>
<td>-1.3</td>
</tr>
<tr>
<td>(h_{int})</td>
<td>1</td>
<td>+20</td>
<td>+0.5</td>
</tr>
<tr>
<td>(L)</td>
<td>1</td>
<td>+20</td>
<td>-0.5</td>
</tr>
<tr>
<td>(k\ (W/m/K))</td>
<td>20</td>
<td>+20</td>
<td>+0.23</td>
</tr>
<tr>
<td>(\varepsilon)</td>
<td>0.8</td>
<td>+20</td>
<td>-2.7</td>
</tr>
</tbody>
</table>

alloy blades is 15.8%. For the TBC coated blades the mean percentage error is 15.7%. Thus a value of 16% is applied here.

- Internal heat transfer coefficient \(h_{int}\): The value of the internal heat transfer coefficient relies on a typical number for ribbed channel flows provided by ALSTOM. For the different load cases, the value is adjusted according to the coolant massflow change. Thus, a relatively conservative uncertainty of 30% is assumed.

- Blade wall thickness \(L\): In the data reduction procedure, a constant value for the wall thickness is used. However, the blade wall thickness varies along the blade contour. Also, the blade has ribbed cooling channels. Therefore, an uncertainty value of 20% is deemed appropriate.

- Thermal conductivity \(k\): In a clean laboratory experiment, the thermal conductivity can be measured to an accuracy of 4 to 5% [63]. In the present discussion of uncertainties, a more conservative value of 10% is used.

- Surface emissivity \(\varepsilon\): According to White [107], emissivity uncertainties are typically on the order of 10%. In the emissivity measurement of TBC coated samples described in Chapter 6, an average uncertainty of 16% is reported. Thus, a value of 16% is used. This value is also considered to include the effects of errors in apparent emissivity.

The uncertainty in the recovery temperature measurement due to the uncertainties of the individual sources is evaluated according to Figure 4.6. Then the estimated maximum uncertainty is calculated using Equation 4.11. Thus,
the maximum measurement uncertainty for the recovery temperature on the nickel blades is 5.9%.

At an assumed recovery temperature of 1200 K, the measurement method has an absolute uncertainty of 70 K.

4.6 Concluding Remarks

A unique measurement method for the local adiabatic gas recovery temperature is presented. The method extracts recovery temperature data on turbine blades from measurements of the surface temperature and the external heat transfer coefficient. The measurement method equates the conductive and convective heat fluxes at the blade surface using the one-dimensional solution of the heat equation. In the data reduction method, an assessment of the internal heat transfer coefficient for the off-design load cases is used.

From the error assessment, it is concluded that the external heat transfer coefficient and the surface emissivity have a profound influence on the measurement of the recovery temperature. A good understanding of the surface emissivity is important to achieve a reliable measurement of the surface temperature, as it directly affects the resulting recovery temperature. The other variables (internal heat transfer coefficient, wall thickness, thermal conductivity) have a smaller effect on the recovery temperature measurement. The maximum measurement uncertainty is found to be 5.9%. This value corresponds to 70 K at a typical temperature level of 1200 K. This is a considerable error. For the current blade, such an error in the gas temperature assessment would cause an error in the metal temperature prediction of approximately 45 K. For a high-pressure turbine blade of an aero-engine, Rose [82] estimated that a metal temperature variation of 30 K would reduce the blade life by a factor of two.

Regarding the extent of the validity of a one-dimensional approach to measure the recovery temperature, the following conclusions are drawn. On the pressure surface of both the coated and uncoated blades, the measurement method yields reasonable recovery temperature data for all the load cases considered. On the suction surface of the TBC blades, the measurement method also works satisfactorily. On the suction surface of the uncoated blades, however, the data reduction procedure fails to deliver consistent data; in only
a few of the load cases, a solution is found. In the leading edge region of the blade, high local values of recovery temperature data are found. In this part of the blade, the one-dimensional approach used in the data reduction is prone to errors. The application of a two-dimensional heat transfer model in the data reduction may help to address this problem.

The present approach does not claim to be a technically mature measurement method. Further work is required to refine the method. However, the simplicity of the present measurement method in combination with the measurement of heat transfer coefficients may promise a future application in the development process of gas turbines.
Chapter 5

Thermal Diffusivity

5.1 Introduction

The measurement of heat transfer coefficients on turbine blades involves the application of mathematical models that describe the physics of the heat transfer process. In this work, the one-dimensional heat equation is solved to determine the convective heat transfer coefficient. In its numerical form, the heat equation contains the variables for the thermal diffusivity and conductivity of the blade material.

Many different methods for the measurement of thermal diffusivity are described in the literature. According to Maglic [62], thermal diffusivity measurement methods can be divided into the following two groups: Transient heat flow methods and periodic heat flow methods. In the transient heat flow methods, the thermal diffusivity is determined by measuring a temperature perturbation or temperature history on the sample. The most commonly used method of this group is the laser flash method. In this technique, the front face of a small sample is heated by radiant energy from a laser. The thermal diffusivity is then computed from the temperature response on the rear surface of the sample. Vozar has recently reviewed the state of the art of the laser flash method and its derivatives [106]. The periodic heat flow methods are based on the measurement of the attenuation or the phase shift of temperature waves along the path of their propagation. One of the widely applied periodic methods is the temperature wave method. A large variety of variants of this technique exists, which can be classified depending on whether the
thermal waves are generated by electrical [13] [73] [74] or optical [27] [39] [40] heating.

This chapter describes a periodic heat flow method for the measurement of thermal diffusivity. It is based on the dynamic detection of an alternating temperature distribution that is induced by the application of periodic optical heating. Specifically, the phase relation of the detected temperature signal to the excitation signal is measured, which correlates with the thermal diffusivity of the material. The excitation and wave detection are applied on the same face of the sample, since only one face of a turbine blade is accessible without destroying it. Optical fibers are used instead of open laser beam paths. Thus, the proposed method is a non-destructive technique and has the future potential of field application. The diffusivity of selected materials is measured on cylindrical samples and compared with diffusivity data from the literature.

In the measurement of the thermal phase relationship of the periodic signal, the periodic temperature variation of the blade surface is evaluated. The absolute temperature of the sample is only used for the comparison with the reported diffusivity values, which in general are varying with temperature. The measurement of the phase instead of the amplitude of the surface temperature response has the distinct advantage of being less affected by local variations of surface emissivity.

5.2 Theory and Data Reduction

The method for measuring the thermal diffusivity follows an approach that is described by Fabbri [39] and Aamodt [1]. In their work, they describe photo-thermal measurements using localized excitation sources. They use lenses and mirrors to focus the open laser beam, whereas in this work, optical fibers are used. However, the theoretical background of the two approaches is similar.

The spatial temperature distribution that is induced by a harmonic optical excitation can be described by the time-invariant and the periodic part as [1]
\[ T(x, t) = T_{dc}(x) + T_{ac}(x) \exp(-i\omega t), \quad (5.1) \]

where \( T \) is the temperature rise of the sample due to the optical excitation. The periodic part \( T_{ac} \) satisfies Helmholtz's equation that is given by

\[ (\nabla^2 + q^2)T_{ac} = -\frac{A_{ac}(x)}{k} \quad (5.2) \]

In the above equation, \( q \) is the complex wave number that is given by

\[ |q| = (1 + i)/\mu, \quad (5.3) \]

where \( \mu \) is the thermal diffusion length that is defined by

\[ \mu = \sqrt{2\alpha/\omega}. \quad (5.4) \]

In this equation, \( \alpha \) is the thermal diffusivity and \( \omega \) the angular frequency of the excitation source. The term \( A_{ac} \) in Equation 5.2 represents the periodic part of the heat source. If the heat is supplied by a laser beam of power \( P \) that has a Gaussian power distribution according to Equation 5.5, the periodic part of the heat source is given by Equation 5.6

\[ I(r) = I_0 \exp\left(-\frac{r^2}{R^2}\right) \quad (5.5) \]

\[ A_{ac}(r, z) = \frac{P(1 - \rho)}{\beta} 2\pi R^2 I(r) \exp(-\beta z). \quad (5.6) \]

Here, \( R \) is the excitation-beam radius, \( r \) the radial coordinate from the center of the beam and \( z \) the coordinate normal to the surface. \( I(r) \) can be expressed as the inverse Hankel transform,

\[ I(r) = \int_0^\infty \frac{I_0 R^2}{2} \exp\left(-\frac{\lambda^2 R^2}{4}\right) J_0(\lambda r) \lambda d\lambda \quad (5.7) \]
where \( J_0 \) is the zero-order Bessel function of the first kind and \( \lambda = 1/r \). Using the zero-order Bessel function for the profile of the laser beam, the periodic part of the temperature profile on the surface is given by:

\[
T_{ac}(r) = \frac{P(1 - \rho)}{2\pi} \int_{0}^{\infty} \exp\left(-\frac{\lambda^2 R^2}{4}\right) E(\lambda) J_0(\lambda r) \lambda d\lambda
\] (5.8)

where \( E(\lambda) = \frac{1}{k\sigma(\lambda)} \) for a thermally thick sample with \( L >> \mu \). The parameter \( \sigma \) is defined by Equation 5.9.

\[
\sigma(\lambda) = \sqrt{\lambda^2 + \frac{2i}{\mu^2}}
\] (5.9)

In the experiment, excitation frequencies as low as 0.1 Hz are used. For a typical thermal diffusivity of a nickel alloy on the order of \( 4 \times 10^6 \) m²/s, the criterion of a thermally thick sample is satisfied, if the sample thickness exceeds a value of 4 mm. For the samples used in this study, this criterion is fulfilled.

For a thermally thick sample, the amplitude of the periodic temperature signal is calculated using Equation 5.8. The temperature distribution is normalized by the characteristic temperature that is defined by

\[
T_e = \frac{P}{\pi k R}
\] (5.10)

and is plotted versus the radial offset \( r \) in Figure 5.1. The three curves are obtained for different radii of the laser beam. As expected, the temperature in the center of the beam increases with decreasing beam radius, because the laser energy is fixed.

The phase angle of the periodic temperature at the sample surface is shown in Figure 5.2. The asymptotic behavior of the phase angle has a linear dependence that is related to the thermal diffusion length \( \mu \). If the phase angle is plotted versus the radial offset normalized by the thermal diffusion length, the slopes of the curves asymptotically approach a value of -1. Thus, the phase profile can be described by Equation 5.11.
Figure 5.1: Temperature distribution induced by laser beam versus radial offset. Different curves arise from variation in laser beam radius. Theoretical amplitudes at 2.5 mm radial offset marked for (a) 0.1 Hz, (b) 0.2 Hz, (c) 0.5 Hz.

\[
\phi(r) = -\frac{r}{\mu} = -\frac{r}{\sqrt{2\alpha/\omega}} 
\]

(5.11)

The linear behavior is also explained by the Green solution of Equation 5.1. In this case, the Green function represents a point-like laser source that is described by

\[
G(r, 0) = \frac{T_0}{r} \exp \left[-\frac{(1 + i)r}{\mu}\right]. 
\]

(5.12)

From the above considerations, the following conclusions can be drawn for the setup of the experiment for the measurement of thermal diffusivity:

- The phase of a point-like laser source has a constant slope according to Equation 5.12.
- For a finite laser beam radius, the phase is measured at a certain distance from the center of the beam, where the phase behavior is governed by the linear relation.
Figure 5.2: Theoretical phase angle distribution versus radial offset. Phase at 2.5 mm radial offset marked for (a) 0.1 Hz, (b) 0.2 Hz, (c) 0.5 Hz.

- From Figure 5.1 it is apparent that if the signal-to-noise ratio in the experiment is low, a small laser beam radius and a small radial offset are advantageous. Thus, the linear behavior of the phase (Figure 5.2) extends close the center of the beam, where the amplitudes are high.

- High signal to noise ratio of the periodic temperature variation at the offset position is achieved by using high laser power.

In the data reduction procedure, the phase lag between excitation and temperature response is measured for different pulsation frequencies. The phase lag is determined by cross-correlation of the two signals. Once the phase is measured, Equation 5.11 is used to determine the phase lag for different frequencies. The thermal diffusivity is then calculated by determining the least squares fit of the experimental phase-frequency data to the modeled phase-frequency correlation.

### 5.3 Experimental Setup and Instrumentation

The diffusivity measurements are conducted on cylindrical metal samples, heated in an Isotech Pegasus calibration furnace. This furnace has a cylindri-
5.3 Experimental Setup and Instrumentation

Figure 5.3: Schematic of the experimental setup for the diffusivity measurement.

cal cavity that is open to the front and allows temperatures of up to 1200°C. For the surface temperature measurements, infrared pyrometers are used. The specifications of the pyrometers are described in Chapter 2.1. A probe head is designed that holds three optical fibers. The first optical fiber, the feeding fiber, is used to heat the sample by means of a chopped laser beam. A fiber coupler is used to couple the laser beam into the fiber. In this process, approximately 20% of the laser power are lost. Another 40% of the initial laser power are lost in the optical fiber. Thus, 40% of the available laser power are effectively used to heat the sample. As a laser source, a Coherent Verdi V5 laser is used. This is a diode-pumped, solid state (DPSS) laser of maximum 5 W continuous-wave output power at a wavelength of 532 nm. A programmable laser shutter (nm Laser Products, Inc.) is used in this study for chopping the laser beam at different frequencies, ranging from 0.1 to 5 Hz. The shutter is controlled by an analog function generator. The operating frequency is manually set and monitored using a digital multimeter. Two additional fibers, the reading fibers, serve as the optical guides for the pyrometers. A schematic of the experimental setup is shown in Figure 5.3.

In setting up the experiment, several boundary conditions are taken into account. The determination of thermal diffusivity is based on the measurement of the phase angle of the surface temperature response according to Equation 5.11. Thus for the experiment, the radial offset of the reading fiber, the pulsation frequency and the laser output power are variables. In order to achieve high signal-to-noise ratio, the highest possible laser power is chosen. The lowest pulsation frequency in the experiment was set to 0.1 Hz. This
low frequency yields a clear signal on the sample surface while maintaining a reasonably short experimental duration. The optical fibers that are available for these experiments have a tip diameter of 2.2 mm. The beam diameter is determined by the lens diameter of the fiber, which in this case is 1.4 mm. Considering the hardware constrains as well as the theoretical discussion of Section 5.2, the radial offsets of the reading fibers are set to 2.5 and 5 mm, respectively.

A prototype probe of 18 mm diameter holding the three optical fibers is developed and used for the present set of experiments. The probe head is designed in such a way as to hold several fibers adjacently aligned in parallel, see Figure 5.4. The feeding fiber is placed in the center of the probe, while the reading fibers are positioned at different radial offsets. The prototype probe head has six holes to hold fiber tips, to allow some flexibility in the probe setup. The large hole on the upper right hand side of the probe tip is used for the assembly of the probe.

Table 5.1 summarizes the ratio of laser beam and offset distances to diffusion length, respectively, for the excitation frequencies that are used in the experiment. A thermal diffusivity of $4 \cdot 10^{-6} \, \text{m}^2/\text{s}$ is assumed, a typical value for metals. Some of the values from Table 5.1 are indicated in Figure 5.1 and 5.2. As seen, high amplitudes result only at low frequencies. However, for the small radial offset distance, clear signals are also achieved in the experiment for a pulsation frequency of 1 and 2 Hz, respectively. For the large offset distance, the signal-to-noise ratio is sufficient only at low frequencies.
Table 5.1: Ratio of laser beam and offset distances to diffusion length for the experiments of thermal diffusivity for laser beam radius of 1.4 mm and thermal diffusivity of $4 \cdot 10^{-6}$ m²/s.

<table>
<thead>
<tr>
<th>$f$ (Hz)</th>
<th>$R/\mu$</th>
<th>$r_1/\mu$</th>
<th>$r_2/\mu$</th>
<th>$\mu$ (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>0.4</td>
<td>0.7</td>
<td>1.4</td>
<td>3.6</td>
</tr>
<tr>
<td>0.2</td>
<td>0.6</td>
<td>1.0</td>
<td>2.0</td>
<td>2.5</td>
</tr>
<tr>
<td>0.5</td>
<td>0.9</td>
<td>1.6</td>
<td>3.1</td>
<td>1.6</td>
</tr>
<tr>
<td>1</td>
<td>1.2</td>
<td>2.2</td>
<td>4.4</td>
<td>1.1</td>
</tr>
<tr>
<td>2</td>
<td>1.8</td>
<td>3.1</td>
<td>6.3</td>
<td>0.8</td>
</tr>
<tr>
<td>5</td>
<td>2.8</td>
<td>5.0</td>
<td>9.9</td>
<td>0.5</td>
</tr>
</tbody>
</table>

5.4 Experimental Results

The thermal diffusivity is determined on selected material samples by the approach described above. The calibration probe that holds the feeding fiber and the two reading fibers is used. The samples are periodically excited and the temperature response on the surface is measured at offset distances of 2.5 and 5 mm, respectively. Three different materials are tested: Inconel 600, NiCr 80/20 and ARMCO iron. The former two high temperature nickel alloys are often used in turbomachinery applications. The third material, ARMCO iron, is widely used as a reference material.

Figure 5.5 shows a typical signal of the surface temperature response and the laser excitation. The square-wave signal is received from the function generator controlling the optical chopper. The temperature signal of the surface response is measured using the pyrometer. In Figure 5.5, the amplitudes of the signals are scaled by a suitable reference amplitude for the sake of clarity.

The phase diagrams of the Inconel 600 and Armco iron samples for the small offset distance of 2.5 mm are shown in Figures 5.6 and 5.7, respectively. The results of all the tested materials are summarized in Table 5.2 on page 125.

On the Inconel 600 sample, the measured thermal diffusivity is underestimated at all three test temperatures (h, m and l) by approximately 25%.
compared with the literature values of Touloukian [100]. However, the trend of slightly increasing thermal diffusivity with increasing temperature is correctly reproduced.

The thermal diffusivity of ARMCO iron shows stronger variation with temperature than the other two materials. Specifically, the thermal diffusivity of ARMCO iron decreases with increasing temperature for temperatures below its transition point at 1038 K. As can be seen from Figure 5.7, at the high (h) and medium (m) temperatures, the discrepancy compared with the literature values are small, 0 and 12%, respectively. At the low test temperature (l), the error increases to 51%, see Table 5.2 on page 125.

For the large offset distance of 5 mm, the phase lag is only measurable at the two lowest pulsation frequencies. As described in Section 5.2, the amplitude of the periodic signal decreases with increasing distance from the beam center. Furthermore, the heat that is supplied to the surface per laser pulse decreases with increasing excitation frequency and, consequently, the temperature response on the surface. Thus, the periodic signal is not detected at the large offset distance and high excitation frequencies.

The measured thermal diffusivity versus frequency for ARMCO iron at 830 K is shown in Figure 5.8. The literature value [100] is also shown for comparison. The thermal diffusivity is over-predicted at low frequencies. At the higher
5.4 Experimental Results

Figure 5.6: Phase lag versus excitation frequency for the Inconel 600 sample. Different curves obtained at different temperatures, designated by h, m and l for high, medium and low temperature, respectively (Table 5.2).

frequencies, it agrees well with the literature value. An explanation for this behavior may be found in the theoretical phase analysis, Figures 5.1 and 5.2. For the lowest two excitation frequencies, 0.1 and 0.2 Hz, the temperature amplitude on the surface is high. However, as the phase diagram shows, these settings are fairly close to the non-linear phase behavior that is observed close to the excitation beam. This may explain the relatively poor diffusivity measurement at low excitation frequencies. At the highest pulsation frequency of 5 Hz, the periodic signal on the surface is not detected. Therefore, no diffusivity value is shown in Figure 5.8 at a pulsation frequency of 5 Hz.

The differences between the Inconel 600 and NiCr 80/20 samples and their respective literature values seem to be quite high, ranging from -22.7 to -30.8%. However, impurities in the metal or changes in the chemical composition of the alloy can cause considerable changes in thermal diffusivity [100].
Figure 5.7: Phase lag versus excitation frequency for the ARMCO iron sample. Different curves obtained at different temperatures, designated by h, m and l for high, medium and low temperature, respectively (Table 5.2).

Figure 5.8: Thermal diffusivity versus pulsation frequency for ARMCO iron at 830 K. Frequency axis stops at 2 Hz, since periodic signal is not detected at excitation frequency of 5 Hz.
Table 5.2: Summary of the diffusivity measurement results. Radial offset distance is 2.5 mm.

<table>
<thead>
<tr>
<th>Label</th>
<th>Temperature (K)</th>
<th>$\alpha \times 10^6$ (m$^2$/s)</th>
<th>$\alpha_0$ [100]</th>
<th>Diff. (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inconel 600, l</td>
<td>720</td>
<td>3</td>
<td>4.1</td>
<td>-26.8</td>
</tr>
<tr>
<td>Inconel 600, m</td>
<td>870</td>
<td>3.2</td>
<td>4.3</td>
<td>-25.6</td>
</tr>
<tr>
<td>Inconel 600, h</td>
<td>1000</td>
<td>3.3</td>
<td>4.4</td>
<td>-25</td>
</tr>
<tr>
<td>ARMCO iron, l</td>
<td>680</td>
<td>17.4</td>
<td>11.5</td>
<td>51.3</td>
</tr>
<tr>
<td>ARMCO iron, m</td>
<td>830</td>
<td>5.6</td>
<td>5</td>
<td>12</td>
</tr>
<tr>
<td>ARMCO iron, h</td>
<td>950</td>
<td>4</td>
<td>4</td>
<td>0</td>
</tr>
<tr>
<td>NiCr 80/20, l</td>
<td>710</td>
<td>3.4</td>
<td>4.4</td>
<td>-22.7</td>
</tr>
<tr>
<td>NiCr 80/20, m</td>
<td>860</td>
<td>3.4</td>
<td>4.9</td>
<td>-30.6</td>
</tr>
<tr>
<td>NiCr 80/20, h</td>
<td>1000</td>
<td>3.6</td>
<td>5.2</td>
<td>-30.8</td>
</tr>
</tbody>
</table>
5.5 Uncertainty Analysis

This section describes the uncertainty analysis of the thermal diffusivity measurement. The thermal diffusivity for a single excitation frequency according to equation 5.11 is given by

$$\alpha = \frac{\omega r^2}{2\phi^2} = \frac{2\pi f r^2}{2\phi^2}, \quad (5.13)$$

where $f$ is the excitation frequency, $\phi$ the phase angle in radians and $r$ the radial offset from the center of the beam. According to Coleman [26], the uncertainty in the measurement result can be expressed by the sum of the absolute values of the uncertainties of the measured variables. The expression for the measurement uncertainty of the thermal diffusivity then reads

$$\Delta \alpha = \left| \frac{\partial \alpha}{\partial f} \cdot \Delta f \right| + \left| \frac{\partial \alpha}{\partial r} \cdot \Delta r \right| + \left| \frac{\partial \alpha}{\partial \phi} \cdot \Delta \phi \right|. \quad (5.14)$$

The uncertainty in the radial offset is determined by how accurately the holes in the probe head are drilled. The radial offsets used in the experiment are 2.5 and 5 mm. The uncertainty is taken as $\pm 0.1$ mm. The pulsation frequency $f$ is generated by a function generator and is manually set. It is monitored throughout the experiments by a digital multimeter; the uncertainty is less than 2%. The calculation of the phase angle is performed by built-in Matlab routines. A conservative value of 10% is assumed for the phase angle uncertainty. Table 5.3 summarizes the uncertainty values of the variables for the thermal diffusivity measurement.

Figure 5.9 shows a typical error distribution for the ARMCO iron sample for

<table>
<thead>
<tr>
<th>Variable</th>
<th>Symbol</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radial offset</td>
<td>$\Delta r$</td>
<td>$\pm 0.1$ mm</td>
</tr>
<tr>
<td>Pulsation frequency</td>
<td>$\Delta f$</td>
<td>2%</td>
</tr>
<tr>
<td>Phase angle</td>
<td>$\Delta \phi$</td>
<td>10%</td>
</tr>
</tbody>
</table>
5.5 Uncertainty Analysis

Figure 5.9: Diffusivity error vs. pulsation frequency for ARMCO iron sample at 830 K surface temperature. Individual variable uncertainties also shown.

different pulsation frequencies. The sum of the individual errors, i.e. the total error, is shown by the triangular symbols. The principal uncertainty arises from the radial offset distance between the reading fiber and the center of the laser beam. The absolute value of the uncertainty is high at low pulsation frequencies. The percentage error, however, is independent of the pulsation frequency, Figure 5.10. The highest pulsation frequency that is used in the experiment, 5 Hz, is not shown in the error analysis, since a periodic signal is not detected. Thus, the phase angle can not be determined at this high frequency.

It is also observed that the measurement error is independent of the sample surface temperature. At low surface temperatures, the periodic temperature signal is somewhat noisier than at high surface temperatures. However, since a constant value of 10% for the phase angle uncertainty is applied, the resulting diffusivity error is not affected by the variation in surface temperature.

The resulting measurement error is 16.1% for the variable uncertainties that are listed in Table 5.3.
5.6 Conclusions

The present method of measuring thermal diffusivities was developed with the prospect of future in-situ applications. Thus, optical fibers are used for the thermal excitation of the sample surface as well as for the reading of the temperature response. A prototype probe holding the three optical fibers was developed and used for the present experiments.

The measured data shows good agreement with literature data for ARMCO iron. The discrepancy to the literature value at the high and medium surface temperature is 0 and 12%, respectively. For the two other tested materials, Inconel 600 and NiCr 80/20, the agreement is less favorable: The mean difference to the literature value is 26.9%.

The measurement error is independent of both the excitation frequency and surface temperature. It’s absolute value is 16.1%.

The actual measurement technique may be improved in several ways. The optical fibers used in the present setup have a lens diameter of 1.4 mm. As the theoretical considerations show, smaller lens diameters could improve the quality of the data. A small lens diameter extends the linear regime in the phase diagram and thus broadens the range to include the higher values of

Figure 5.10: Percentage error vs. pulsation frequency for ARMCO iron sample at 830 K surface temperature.
applicable pulsation frequencies.

Furthermore, 40% of the available laser power are lost in the optical fibers. The application of single-mode fibers could reduce this value and thus increase signal-to-noise ratios of the measured temperature signal.
Chapter 6

Measurement of Surface Emissivity

6.1 Introduction

Today, radiation thermometry is a widespread technology utilized in both research and the industry for the measurement of surface temperatures. The advantages over traditional measurement methods are the high temporal and spatial resolution and its non-contact way of operation. However, in order to allow for high measurement accuracy, the target surface’s emissivity needs to be determined.

Surface emissivity generally depends on temperature, wavelength, measurement angle and surface condition [87]. Many different methods are described in the open literature to measure surface emissivity. Alaruri [6] measured TBC emissivity at a single wavelength by evaluating both thermocouple and pyrometer temperature data. Ng [76] used a two-color pyrometer to measure the surface temperature and emissivity of specimens with 250 μm thick ZrO₂ TBC. In this two-color pyrometry, the radiation intensity is measured at two different wavelengths, simultaneously yielding both the emissivity and surface temperature. Using a four-color pyrometer, Tago [97] determined surface temperature and emissivity on stainless steel and showed improved accuracy over the traditional two-color methods. Ganz [44] studied the effect of various attributes such as layer thickness, surface roughness and composition of TBC on the surface emissivity. Ganz determined the normal spectral emissivity
using emission spectroscopy. In this technique, the emitted radiation of the specimen was compared with the radiation from a blackbody source at the same temperature.

In radiation thermometry, the temperature measurement may be biased by reflected radiation. Depending on the temperature levels of target and surrounding surfaces and the respective emissivities, a portion of the target’s total radiance consists of reflected radiation from the surroundings. The reflected radiation leads to an overestimation of the measured surface temperature. Three concepts were proposed in the literature to correct for reflected radiation [107, 84]. In the first concept, reflected radiation is ignored. However, this concept only yields acceptable measurement results if the temperature of the target surface is considerably higher than the temperatures surrounding surfaces. In the second concept, the environment and the target are of similar temperature. Here, the system behaves as a blackbody source and the emissivity is assumed to be close to 1. In the third concept, large temperature gradients are present and a correction for reflected radiation is necessary. Freid [43] adopted this third strategy and applied it in a modified form for the measurements of temperatures in solar reactors. De Lucia [29] developed a computer program based on this method to correct measured temperatures of a turbine blade model.

This chapter describes the measurement of emissivity and temperature on TBC coated and uncoated blade samples. The measurement technique adopted was described by Brandt [20]. Spectral emissivities are determined by comparing the measured radiance temperature of a reference area of known emissivity with the radiance temperature of the target area. The measurements are conducted in an environment of reflected radiation. Thus, the third concept mentioned above is used to correct the temperature and emissivity measurements. For the correction method, the surface temperatures of the surroundings also need to be measured. The associated measurement errors are assessed by means of a Monte Carlo analysis and as well as an analytical procedure.
6.2 Data Reduction

The measurement procedure for the determination of TBC emissivity is based on the pyrometric measurement of surface temperatures on TBC coated samples. An area on the sample surface is painted with a high-temperature paint of known emissivity. The temperatures on the painted and unpainted surface areas are used to determine the normal spectral emissivity.

A measured temperature value can be converted to a radiosity value according to the integrated Planck equation given by Equation 6.1.

\[
B_{\lambda i} = \varepsilon_{\lambda} \int_{\lambda_1}^{\lambda_2} \frac{2\pi C_1}{\lambda^5 \left[ \exp \left( \frac{C_2}{\lambda T} \right) - 1 \right]} d\lambda,
\]

where \( C_1 \) and \( C_2 \) are Planck’s constants, \( \lambda \) is the wavelength, \( T \) the temperature and \( B_{\lambda i} \) is the spectral radiosity of surface \( i \). Since a narrow bandwidth pyrometer is used, the radiosity is calculated by means of the monochromatic assumption and Wien’s approximation to Planck’s law:

\[
B_{\lambda i} = \varepsilon_{\lambda} \frac{2\pi C_1}{\lambda^5 \left[ \exp \left( \frac{C_2}{\lambda T} \right) \right]} \Delta \lambda,
\]

where \( \Delta \lambda \) is the bandwidth of the pyrometer. The errors introduced by these simplifications are on order of 0.2% for the temperatures extracted from the experiments (see Section 6.6).

The method to reduce the measurement errors due to reflected radiation is described by different authors [107, 84, 43]. It employs the rigorous application of Equation 6.3 to calculate the target surface radiosity:

\[
B_{\lambda i} = \varepsilon_{\lambda} B_{\lambda ib} + (1 - \varepsilon_{\lambda}) \sum_{j=1}^{n} B_{\lambda i} \cdot F_{Ai-Aj},
\]

where \( \varepsilon_{\lambda} \) is the target surface normal spectral emissivity, \( B_{\lambda ib} \) is the target surface spectral blackbody radiosity, \( B_{\lambda j} \) is the spectral radiosity of the surrounding surface \( j \) and \( F_{Ai-Aj} \) is the configuration factor from surface \( i \) to surface \( j \) (see Section 6.3).
Equation 6.3 is rearranged to yield an expression for the normal spectral emissivity of the surface:

\[ \varepsilon_{XTBC} = \frac{\sum_{j=1}^{n} B_{\lambda j} \cdot F_{Ai-Aj} - B_{XTBC}}{\sum_{j=1}^{n} B_{\lambda j} \cdot F_{Ai-Aj} - B_{\lambda b}}, \tag{6.4} \]

where \( B_{XTBC} \) is the radiosity derived from the measured apparent temperature of the unpainted area and \( B_{\lambda b} \) is the radiosity of the painted area of the sample. \( B_{\lambda b} \) is determined from the apparent surface temperature of the pyrometer and the known emissivity of the black paint. The radiosities of the surrounding ring-shaped surfaces \( B_{\lambda j} \) are determined from the temperature measurement of these surfaces. In the absence of the knowledge of their emissivities, an emissivity value of 1 is used. The error that is introduced by this assumption is considered to be small, due to the dark coloration of these surfaces in the visible range. Ignoring the effect of reflected radiation, Equation 6.4 simplifies to

\[ \varepsilon_{XTBC} = \frac{B_{XTBC}}{B_{\lambda b}}. \tag{6.5} \]

The determination of the configuration factors that are used in Equation 6.4 is described in the following section.

### 6.3 Configuration Factors

Geometric relations are needed in the calculation of radiative heat transfer between surfaces. These geometric relations reflect how the surfaces see each other and are defined as configuration factors. The configuration factors allow the calculation of radiative heat transfer according to Equation 6.3.

The configuration factor from a finite area \( A_2 \) to an area \( A_1 \) is defined by \[87\]

\[ F_{1-2} = \frac{1}{A_1} \int_{A_1} \int_{A_2} \frac{\cos \theta_1 \cos \theta_2}{\pi S^2} dA_2 dA_1, \tag{6.6} \]

where \( S \) is the line of length between \( dA_2 \) and \( dA_1 \) and \( \theta \) is the angle between the surface normal and \( S \).
The configuration factors for the present setup are calculated both analytically and by means of a Monte Carlo analysis. The analytic solution for the present geometry is defined by Equation 6.7 [87]. The geometrical configuration is shown in Figure 6.1.

\[
F_{1-2} = \frac{(X_1 - X_2) - (X_1^2 - 4R^2)^{\frac{1}{2}} + (X_2^2 - 4R^2)^{\frac{1}{2}}}{4R(H_2 - H_1)},
\]

where \( R = R_2/R_1 \), \( H_i = h_i/R_1 \) and \( X_i = H_i^2 + R_i^2 + 1 \) (\( i \) represents surfaces 1 and 2). The analytic solution neglects the radial offset of the sample to the cylinder axis.

In the Monte Carlo analysis, the radiative heat exchange between two surfaces is modeled by following the progress of discrete amounts of energy defined as bundles. The energy flux is then calculated by the number of bundles that arrive at a surface per unit area and time [50]. In the case of the configuration factors, the bundles can be considered as rays going from a surface to the target surface. The direction and starting point on the surface are randomly
determined. Using a high number of rays, the ratio of the total number of rays to the number arriving at the target surface then determines the configuration factor. Thus, the configuration factor for the following geometrical configuration is computed: $h_1=10$ mm, $h_2=20$ mm, $R_1=3$ mm and $R_2=17.7$ mm. The Monte Carlo solution is compared with the analytical solution in Figure 6.2. For a high number of bundles, the Monte Carlo solution converges towards the analytical solution.

The Monte Carlo technique can be applied to complex situations and geometries, where the analytical approach may be too expensive. Thus, the radial offset of the sample from the center of the cylinder is included in the Monte Carlo model and the configuration factor is computed. As expected, the variation in configuration factor is small, on the order of 1%.

In the determination of the surface emissivity, the configuration factors for all the segments of the furnace cavity are calculated using the analytical solution and the respective geometrical data.

Figure 6.2: Configuration factor by means of Monte Carlo analysis (MC) for different numbers of bundles.
6.4 Experimental Setup

Square samples of nickel alloy coated with a YSZ thermal barrier coating are placed inside the cylindrical furnace cavity. The samples have a side length, thickness and coating thickness of 1 cm, 3 mm and 0.5 mm, respectively. Two different types of specimen are used: TBC coated blade samples with a coating thickness of 0.5 mm and samples of the nickel alloy substrate material (Hastalloy X). The samples are supported by means of a phlogopite insulating material and radiatively heated from the backside using a blackbody source. The insulating material ensures a uniform heating of the samples and minimizes the heat loss caused by thermal conduction. Half of the sample surface is covered with a black thermal paint of known emissivity. The measurement procedure requires the determination of the radiating temperature of several points inside the cavity: The painted and unpainted areas of the sample and the temperatures of the four annular segments of the cavity walls, see Figure 6.3. It is assumed that the temperature variation around the circumference of the annular sections is small.

The pyrometer readings of the annular sections of the cylinder walls are used to correct for the reflected radiation as described in Section 6.2. Two such temperature distributions are shown in Figure 6.4 for the sample location at the bottom of the cavity.

In this study, a single wavelength pyrometer with a bandpass filter of 10 nm bandwidth centered around 1.55 μm is used. The front-end optics consist of an optical quartz fiber, a convex lens of 1.4 mm diameter and an additional
biconvex lens, which enables the adjustment of the size of the measurement area. (For more information on the pyrometer system refer to Table 2.3 on page 24). Unlike in the engine experiments, the tip of the front end is air cooled in these measurements to ensure that the optical parts maintain a constant temperature. The constant, low temperature of the optics prevents any biasing of the measurement from radiation of the optical parts. Care is taken to ensure that the cooling air flow is directed away from the measurement target surface.

The test matrix includes three furnace temperatures: 1000, 1100 and 1200 °C. For each of these furnace temperatures, the surface temperature of the sample inside the cavity is measured as well as the temperature of the cavity’s walls. To study the effect of the reflected radiation on emissivity, the sample is moved closer to the radiation source rather than changing the furnace temperature to increase the surface temperature.

### 6.5 Results and Discussion

Emissivity results for the TBC coated sample and the uncoated nickel alloy substrate are shown in Figures 6.5 and 6.6 respectively. Normal spectral
emissivity is shown versus surface temperature of the respective sample. As described in Section 6.4, the sample is placed at different axial positions inside the cylindrical cavity to vary the surface temperature. A higher surface temperature corresponds to a position further within the heating cylinder of the furnace. Data points obtained at a fixed furnace temperature are indicated by the same symbol shape. The solid symbols represent emissivity values derived by using Equation 6.4. This equation accounts for the effect of reflected radiation. The open symbols represent measured emissivity data that ignored the effect of reflected radiation. These data are calculated based on Equation 6.5.

Referring to Figure 6.5, the solid symbols are slightly shifted towards lower surface temperatures with respect to the open symbols. The shift is caused by the emissivity of the thermal paint, which is used to determine the surface temperature. The emissivity of the paint is lower than the ideal blackbody value of 1. In the data reduction, a value of 0.92 was used. Thus, radiation is also reflected from the painted area on the sample. As a consequence, the surface temperature is over-predicted when the reflected radiation is ignored.

The uncorrected emissivity values of the TBC sample at a furnace temperature of 1000 °C increase proportionally with the increase of surface temperature (Figure 6.5). The same behavior is observed for the high furnace temperatures of 1100 and 1200 °C. This increase in emissivity is due to the increased effect of the reflected radiation from the cylinder walls. As the sample was positioned farther into the furnace cavity, radiation from the cavity walls reflects off the unpainted area of the sample. The increased amount of reflected radiation leads to an overestimation of the measured temperature of the unpainted area, which causes the over-prediction of the emissivity. Figure 6.5 also shows the emissivity data, when the effect of reflected radiation is taken into account. Apart from experimental scattering, the emissivity remains constant despite varying surface temperatures. Temperature independent emissivity behavior for TBC was also reported by Alaruri [6], who measured the emissivity of YSZ thermal barrier coatings by means of an integrating sphere. The mean value of the corrected emissivity data, as seen in Figure 6.5, for this study is determined to be 0.33 ± 0.03. Several authors have reported similar emissivity values of production line YSZ thermal barrier coatings (Table 6.1).
Table 6.1: Comparison of surface emissivity data for white yttria-stabilized zirconia (YSZ) thermal barrier coating (TBC) of the present work with data from the literature.

<table>
<thead>
<tr>
<th>Emissivity</th>
<th>Wavelength (nm)</th>
<th>Thickness (mm)</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.33 ±0.03</td>
<td>1550</td>
<td>0.5</td>
<td>present</td>
</tr>
<tr>
<td>0.4 ±0.02</td>
<td>1600</td>
<td>0.33</td>
<td>[6]</td>
</tr>
<tr>
<td>0.3 ±0.03</td>
<td>2000</td>
<td>0.5</td>
<td>[44]</td>
</tr>
<tr>
<td>0.48 ±0.015</td>
<td>900</td>
<td>unknown</td>
<td>[65]</td>
</tr>
</tbody>
</table>

Figure 6.5: Normal spectral emissivity of TBC coated sample versus sample surface temperature. Solid symbols corrected for reflected radiation, open symbols uncorrected. Different surface temperatures result from different locations in furnace cavity.

Figure 6.6 shows the measured normal spectral emissivity of the nickel alloy substrate sample. The TBC layer from the sample was removed before the measurement. The emissivity was measured at the low furnace temperature (1000 °C) at the beginning of the experiment. A distinct difference in emissivity can be observed between this first experiment and the subsequent measurements at the higher furnace temperatures. The difference in emissivity may be explained by the progressing state of surface oxidation of the substrate sample. According to Kerr [57], a good estimation of the emissivity
of well-oxidized nickel alloys is a value of 0.8. In the experiment at the furnace temperatures of 1100 and 1200 °C, the measured averaged emissivity value is 0.8 ± 0.02. However, for the low temperature experiment at the beginning of the series, the mean value is 0.67 ± 0.03. This low emissivity value suggests that with the progressing oxidation of the surface of the substrate sample, the normal spectral emissivity increases. At the high furnace temperatures, the corrected emissivity data shows a slight decrease with increasing surface temperature. In the literature, both increasing and decreasing emissivities of metals with increasing surface temperature is reported. For temperatures below the so-called X-point, the emissivity decreases with increasing temperature [87]. For temperatures above the X-point, the temperature effect is reversed: An increase in emissivity is observed with increasing temperature. The reported X-point for nickel is at a wavelength of 1500 nm. The spectral emissivity in this study is measured at a wavelength of 1550 nm. Thus, the dependence of the emissivity on the surface temperature is expected to be small. A small temperature dependence is indeed observed in the experimental results at the high furnace temperatures.
6.6 Uncertainty Analysis

For both the corrected and the uncorrected data reduction procedure, an uncertainty analysis was conducted.

6.6.1 Uncorrected Emissivity Data

For the data that ignores the reflected radiation, an analytical uncertainty estimation is done according to Coleman [26].

Based on Equation 6.5, the TBC emissivity can be written as

\[
\varepsilon_{\text{TBC}} = \exp\left[ -\ln\left( \frac{C_2/\exp(\lambda T_b)}{\varepsilon_p} \right) - \frac{C_2}{\lambda T_w} \right]
\]  

(6.8)

The total uncertainty of the TBC emissivity is given by the sum of the uncertainty of the variables as described by Equation 6.9.

\[
\Delta \varepsilon_{\text{calib}} = \left| \frac{\partial \varepsilon_{\text{TBC}}}{\partial T_b} \cdot \Delta T_b \right| + \left| \frac{\partial \varepsilon_{\text{TBC}}}{\partial T_w} \cdot \Delta T_w \right| + \left| \frac{\partial \varepsilon_{\text{TBC}}}{\partial \varepsilon_p} \cdot \Delta \varepsilon_p \right|
\]  

(6.9)

The total error in surface emissivity is composed of: The calibration uncertainty, the pyrometer instrument uncertainty and the uncertainty in the known emissivity of the black paint.

**Pyrometer calibration uncertainty.** Before taking the temperature measurements, the pyrometer was calibrated against a blackbody radiation source. The temperature of the source was measured using a thermocouple. Depending on the quality of the calibration, a consistent measurement error is introduced in the experiment on both the painted and the unpainted areas of the sample. A typical apparent temperature difference between the painted and the unpainted areas of the sample is 90 K at a surface temperature of 1000 °C. By substituting \( T_w \) in Equation 6.8 with \( T_b - 90 \), an expression for the emissivity is found that depends only on the variables for the temperature on the painted area and the emissivity of the paint. Thus, the calibration error is given by Equation 6.10
This expression is added onto the right hand side of Equation 6.9 to include the calibration error in the total emissivity uncertainty.

In addition to the inherent calibration error, several other sources of error that lead to a constant measurement error are addressed.

- **Monochromatic assumption.** The monochromatic assumption in the calculation of the radiance leads to an over-prediction of the radiance at high temperatures and an under-prediction of the radiance at low temperatures [12]. The temperature measurement error ranges from -0.02 K at 800 °C to +0.24 K at 1200 °C.

- **Wien’s approximation.** The application of Wien’s approximation to Planck’s law leads to an error in radiance on the order of 0.2%. Thus, the temperature is under-predicted by 0.2 K at 800 °C and 0.4 K at 1200 °C.

- **Degradation of the optical path.** The degradation of the optical path leads to a constant measurement error. Upon a re-calibration of the pyrometer during the experiments, a typical temperature offset of up to 10 K is observed.

All the above sources of error are combined in the pyrometer calibration uncertainty of ± 20 K.

**Instrument uncertainty.** According to the manufacturer of the pyrometer, the instrument uncertainty after calibration is ± 1 K at a temperature of 1000 °C. Thus, this instrument uncertainty is used for the temperature uncertainty in the first two terms of Equation 6.9.

**Uncertainty in black paint emissivity.** The black high-temperature paint has a nominal emissivity of 0.95. However, measurements of the emissivity of the paint yielded a value between 0.91 and 0.95. Thus, a reference value of 0.93 is taken with an uncertainty of ± 0.02.

The total uncertainty in TBC emissivity as well as the contribution of the individual variables is shown in Figure 6.7. The uncertainty values of the variables that are considered are summarized in Table 6.2. The uncertainty
in the wall temperature measurement is also shown in Table 6.2, which is also considered in the error analysis of the corrected emissivity data of Section 6.6.2

### 6.6.2 Corrected Emissivity Data

For the corrected emissivity data, the error analysis is based on a Monte Carlo simulation. The same variables are considered as for the uncorrected emissivity data of Section 6.6.1. In addition, the uncertainty in the measurement of the wall radiance (temperature) is considered. A standard uncertainty is assigned to the variables that affect the resulting emissivity value, see Table 6.2. In the Monte Carlo method a value within the assigned error band is randomly selected. Thus, the emissivity value is calculated based on Equation 6.4 and the random input variables. This process is repeated for a population of $1 \times 10^5$ input quantity groups to determine the total error distribution.

Figure 6.8 shows the emissivity results of the Monte Carlo analysis. In this figure, a relatively small population of 200 groups of input variables is chosen for the sake of clarity. As can be seen, the considered uncertainties yield errors in the normal spectral emissivity as well as in the surface temperature measurement. The latter error is mainly due to the calibration error.
Table 6.2: Associated standard uncertainties of the input variables for the determination of emissivity.

<table>
<thead>
<tr>
<th></th>
<th>± uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pyrometer calibration uncertainty</td>
<td>±20 K</td>
</tr>
<tr>
<td>Pyrometer uncertainty</td>
<td>±1 K</td>
</tr>
<tr>
<td>Wall temperature measurement uncertainty</td>
<td>±10 K</td>
</tr>
<tr>
<td>Black paint emissivity uncertainty</td>
<td>±0.02</td>
</tr>
</tbody>
</table>

Figure 6.8: Normal spectral emissivity of TBC coated sample and results of the Monte Carlo error assessment for furnace temperature of 1000 °C. A population of 200 input quantity groups is used for sake of clarity.

and the instrument uncertainty (both of which cause constant measurement errors).

In Figure 6.9 the emissivity error based on the Monte Carlo simulation is shown versus the sample surface temperature. The analytical solution for the uncorrected emissivity data is also shown for comparison. These errors are based on a calculation using $1 \times 10^5$ input quantity groups. Figure 6.9a shows the emissivity error ignoring the error of the wall radiance. The errors at low surface temperatures, where the influence of the reflected radiation is small, agree well with the analytical solution for the uncorrected emissivity data. The error increases towards high surface temperatures, where the sample is
Figure 6.9: Normal spectral emissivity error of TBC sample versus sample surface temperature. Analytical solution for uncorrected data shown by dashed line.

positioned far into the cavity and the respective effect of the reflected radiation increases.

Figure 6.9b shows the emissivity error including the uncertainty of the wall radianc measurement. A constant uncertainty value of ±10 K is used, Table 6.2. The uncertainty in wall radianc causes a high error in emissivity at the low furnace temperature of 1000 °C, when the sample is set far into the heating cavity. At this furnace temperature, the wall temperatures of the cylinder are low. Thus, the assumption of a constant error has a higher impact than at higher wall temperatures.

The mean measurement error for the TBC samples over all three furnace temperatures and sample positions is 0.06, corresponding to a percentage error in emissivity of 18%. Ignoring the two sample positions at 1000 °C furnace temperature where the error is exceptionally high, the mean error is 0.054 or 16%.

For the nickel alloy substrate, the mean error is smaller than for the TBC samples. The smaller mean error may be due to the higher surface emissivity, which in turn tends to reduce the bias that is caused by the reflected radiation. The mean error for the nickel alloy samples is 0.04 (5%). However, the qualitative trends shown by Figure 6.9 also apply to the substrate samples.
6.7 Summary and Conclusions

The surface emissivity on TBC coated blade samples and the uncoated nickel alloy substrate was measured at different surface temperatures inside the cavity of a calibration furnace. This testing setup, however, is an environment where reflected radiation distorts the measured surface temperature and emissivity of the samples. In the measurement method, an area on the sample surface is covered with a paint of known emissivity. From the measured radiosities of the painted and unpainted areas, the emissivity of the sample is determined. A strategy is applied to reduce this aforementioned negative effect of reflected radiation on the measurement result. This strategy is based on the measurement of the surface temperatures of the furnace walls and the computation of the corresponding configuration factors.

The measured emissivity data agrees well with data from the literature. For the TBC coated samples, a normal spectral emissivity of 0.33 ±0.03 is measured. In the reflective environment, the measurement error is determined by means of a Monte Carlo analysis. The mean error in emissivity is calculated to be 0.06, which corresponds to a percentage error of 18%. The measurement error depends on the position of the sample in the cavity of the furnace and the furnace temperature. At the low furnace temperature (1000 °C), the errors are highest when the portion of the reflected radiation is high.

The TBC coated samples used in this study were produced from production line hardware (never used in service). The sample surface is clean and shows the typical white color of a new TBC layer. However, in an engine environment, blade surfaces experience degradation due to corrosion, erosion and foreign deposits. This surface degradation results in an increase of emissivity values (0.7 to 0.8). Hence, in the data reduction of the measurement of heat transfer coefficients (Section 2.2.5), considerably higher values than those extracted from the lab experiments are used.
Chapter 7

Conclusions and Prospects

7.1 Conclusive Summary

Measurement of Heat Transfer Coefficients

In-situ measurements of the heat transfer coefficients were successfully made in an operating gas turbine engine. This heavy duty gas turbine (288 MW) is equipped with sequential combustors and operated with gas. Measurements were made on the rotor blades of the low-pressure turbine. These are, to date, the first such measurements obtained in the harsh (high temperature, combusted gases) environment of an operating engine.

The heat transfer coefficients were derived from measurements of the blade surface and upstream gas temperatures while the gas turbine underwent a thermal transient. The temperatures on the surfaces of the rotating blades were derived from measurements using an infrared pyrometer, whose optical access was located at the tip endwall of the upstream vanes. The upstream gas temperature was measured using a rake of thermocouples placed at the exit of the second combustor. The measured time-dependent gas temperature was incorporated into the boundary condition of a one-dimensional heat conduction model that was used to determine the distribution of heat transfer coefficients along the blades’ midspan. This was achieved by computing the least-squares fit of the modeled surface temperature change to the measured surface temperature change.
Heat transfer coefficients were measured at various engine operating conditions. Midspan heat transfer data was presented from TBC coated and uncoated blades. The effect of Reynolds number and incidence angle on the Nusselt number distribution was demonstrated. On the uncoated blades, flow separation was identified on the blade pressure surface for intermediate incidence angles. On the TBC coated blades, the separation bubble was not observed. This led to the hypothesis that the formation of the separation bubble was prevented by an early transition. This transition was induced by the wedged shape of the ceramic top-coat close to the leading edge. On the blade suction surface, heat transfer patterns were observed that are typical of transition regions. This behavior was observed on both the coated and uncoated blades. Leading edge heat transfer rates at blade midspan were also measured on coated and uncoated blades. In spite of the highly unsteady flow around the leading edge, the data showed excellent agreement with correlations for stagnation point flow.

The principle uncertainty in the present measurement technique arose from a limited knowledge of the turbine’s turbulence levels. The turbulence level of the second combustor had a profound effect on the time constant of the thermocouples; this time constant was required to compensate for the first order temporal response of the thermocouple during the thermal transient of the engine. Nevertheless, using a Monte Carlo simulation, the measurement technique had an uncertainty of ±15.8% in the heat transfer coefficients. This uncertainty value is comparable to that encountered in the more benign laboratory environment.

The influence of the surface emissivity on the derivation of heat transfer coefficients was studied. The emissivity was required to determine the blades’ surface temperature from the pyrometer reading. According to the uncertainty analysis, a variation in emissivity of 10% resulted in a change in the measured heat transfer of only 2%.

The equipment for the present measurement technique consisted of: Single-point pyrometers, thermocouples, optical sensors (key phaser) and a computer for the data acquisition. Thus, the total cost of the present hardware amounted to less than 30’000 EUR.

The present measurement technique has proved to be very suitable for in-engine heat transfer measurements. The low cost of the equipment and the measurement accuracy make this technique a powerful and highly relevant
procedure for the assessment of in-situ heat transfer.

**Measurement of Recovery Temperature**

Using the results of the heat transfer measurements, the local adiabatic gas recovery temperature distribution along the blades’ midspan was measured. In fact, these temperature distributions were monumental as they resulted from the first ever successful measurements completed on rotating blades in an operating engine. The measurement method was based on the evaluation of both the conductive and the convective heat fluxes at the blade surface using a one-dimensional model.

The analytical error assessment showed the surface emissivity to have the single most profound influence on the measurement accuracy. The total error was also dependent on: The measured heat transfer coefficients, wall thickness and thermal properties of the blade material. The mean total error was estimated to be 5.9%.

As the relatively large measurement error shows, the present technique does not yet constitute a technically mature measurement method. Further work is required to establish confidence in the present method.

**Calibration**

A prototype probe for the in-situ measurement of thermal diffusivity was devised. The measurement method was based on the thermal excitation of the blade surface using a pulsed laser source. The thermal diffusivity was derived from the measurement of the temperature response on the surface using single-point pyrometers. The prototype probe consisted of three optical fibers, which enable a non-destructive, in situ calibration in the future.

The measurement method was tested on various blade materials at different temperatures in a bench experiment. The resulting diffusivity data was then compared with values from the literature. The estimated mean error in the diffusivity measurement was 16.1%.

The emissivities of the TBC coated and uncoated blade samples were measured in an environment of reflected radiation. A method was developed to
Conclusions and Prospects

successfully reduce the distorting influence of the reflected radiation on the measurement result. The corrected emissivity data showed good agreement with emissivity values from the literature.

As shown in the error estimation of the heat transfer measurement, the surface emissivity had a minor influence on the measured heat transfer coefficient (2% on 10% emissivity variation). In the measurement of recovery temperature, the percentage error due to the emissivity uncertainty was slightly lower (1.4% on 10% emissivity variation).

7.2 Prospects for Further Research

In the measurement of heat transfer coefficient, surface temperature transients were created using a key phaser signal. In the present experimental setup, the temperature and key phaser data were recorded at the same acquisition frequency. Therefore, small variations in rotational speed led to errors in the phase locking procedure, generating a sawtooth temperature pattern in blade regions of high thermal gradients. Hence, for future experiments, it is recommended that the acquisition frequency of the key phaser signal be one order of magnitude higher than the acquisition frequency of the temperature data.

The variation of the reference gas temperature in the present method for the heat transfer measurement was measured at the exit of the second combustor. This temperature data was then translated to the relative frame of reference of the low-pressure turbine blade by means of a through-flow solver. This data reduction procedure may be simplified by measuring the gas temperature variation directly at the inlet of the respective rotor row.

The reference gas temperature was measured using thermocouples with a time constant on the order of 2 seconds. In order to improve the measurement accuracy, efforts may be devoted to developing a higher response gas temperature measurement approach. In particular, the simultaneous application of two thermocouples of different time constants could enable an in-situ determination of these time constants. Such an in-situ calibration would allow for the real-engine effects to further improve the gas temperature measurement procedure.
The thermal calibration of blade materials plays an important role in achieving highly accurate, in-situ heat transfer measurements. The development of a prototype calibration probe as presented in this work, has shown that in-engine calibration may be feasible in the future. A probe could be developed for the simultaneous calibration of the blade material and the measurement of heat transfer coefficients. Such a probe would be comprised of optical feeding fibers for the laser excitation and reading fibers to gather the surface temperature data.

The current method of measuring heat transfer coefficients may be applied to monitor blade heat transfer rates online. Sudden engine load changes performed on a regular basis would be sufficient to gather the necessary temperature data. Due to the thermal aging of blade materials, an in-situ calibration method would most probably be required to achieve sufficient quality in such a monitoring approach.
Appendix A

Surface Temperature Measurement Noise

In the assessment of the blade surface temperature uncertainty described in Section 2.5.2, the measurement errors at different engine load settings were analyzed. The results for the high engine load case were discussed in Section 2.5.2. In the following, the results for the intermediate and low power cases (No. 8 and No. 13 respectively, see Table 2.2 on page 23) are shown.
Figure A.1: Top: Blade-to-blade variation in surface temperature; Center: Pyrometer noise level; Bottom: Nusselt number distribution versus wetted distance. Uncoated blades, load case No. 8 (170 MW).
Figure A.2: Top: Blade-to-blade variation in surface temperature; Center: Pyrometer noise level; Bottom: Nusselt number distribution versus wetted distance. Coated blades, load case No. 8 (170 MW).
Figure A.3: Top: Blade-to-blade variation in surface temperature; Center: Pyrrometer noise level; Bottom: Nusselt number distribution versus wetted distance. Uncoated blades, load case No. 13 (100 MW).
Figure A.4: Top: Blade-to-blade variation in surface temperature; Center: Pyrometer noise level; Bottom: Nusselt number distribution versus wetted distance. Coated blades, load case No. 13 (100 MW).
Appendix B

Stage3D Control Input File

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Bibliography


## Nomenclature

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Publications


Curriculum Vitae

Name: Matthias Herbert Brunner
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