

HEAT TRANSFER MEASUREMENTS IN A TURBINE CASCADE: TAKING ADVANTAGE OF THE THERMAL BEHAVIOUR OF A PSP SENSOR

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ABSTRACT

Film cooling experiments are commonly performed with Pressure Sensitive Paint (PSP). The oxygen sensitive probe molecules allow for measurement of the adiabatic film cooling effectiveness by detecting the concentration of a tracer-gas with different oxygen content. This is made possible by the analogy between transport of heat and mass. The technique is particularly advantageous when the high density of a cold coolant in gas turbines is simulated by the injection of CO₂ into air flow (iso-thermal experiments). Thus the coolant has a different oxygen content compared to the main flow. However, the adiabatic effectiveness is only one of the quantities of interest in film cooling. In order to obtain the convective heat transfer coefficients, a temperature sensor is also required. Most PSP technologies possess some degree of temperature sensitivity that is typically taken into account and eliminated during aerodynamic pressure measurements. However, the unwanted temperature sensitivity can be turned into an advantage in film cooling studies where heat transfer through the wall surface is of interest.

In this paper, the possibility to employ the moderate temperature sensitivity of a standard PSP formulation for heat transfer measurements is explored. An electrically heated foil attached on top of an acrylic wall imposes transient heat flux at the wall surface. The principle of the transient technique is to observe the time response of the surface temperature when a heating step is generated. The proposed thermography technique is based on the linear temperature sensitivity of the photoluminescent sensor for small temperature changes according to a calibration.

Promising results with the proposed technique have been achieved on an engine realistic turbine vane platform. Advantages are full surface coverage with high spatial resolution which is important in film cooling studies. Moreover, the time required for measurements and data reduction is relatively short. Finally, use of a single PSP-coating ensures consistent surface quality during the measurement of all searched quantities – the adiabatic film cooling effectiveness and heat transfer coefficients.

INTRODUCTION

One of the fundamental quantities in film cooling investigations, the adiabatic film cooling effectiveness, has been measured on turbine vane platforms in near-ambient temperature environment. A coating of single-probe PSP (Uni-FIB by ISSI) was used to indicate the partial pressure of oxygen when an oxygen free coolant (CO₂) was injected into a main flow of air. The oxygen concentration at the impermeable surface could then be used to derive the adiabatic film cooling effectiveness according to the analogy between heat and mass transfer. This technique provides high-resolution measurement of the film cooling coverage. However, it does not provide information about the thermal boundary layer in the presence of heat transfer between the flow and the wall. Thus, complementing thermal techniques are employed to measure the wall temperature and heat flux, for example narrow-band thermochromic liquid crystals (TLC) in combination with transient heating of the wall with a given heat storage capacity.

The present test case is shown in Fig. 1. An electrically heated foil made of stainless steel is attached on top of an acrylic wall. The heater foil is employed to generate transient heat flux at the wall surface. As demonstrated by Jonsson and Ott (2008), the heat transfer situation makes it difficult to employ narrow-band TLC techniques for transient experiments for this test case – notably due to the following complexities:

- Compressible flow leading to viscous heating effects and varying adiabatic temperature
- Heated foil configuration making the temperature rise dependent on the heat transfer coefficient
- Engine realistic geometries leading to a complex heating pattern
- Complex 3D flow patterns on platforms leading to large variations in heat transfer

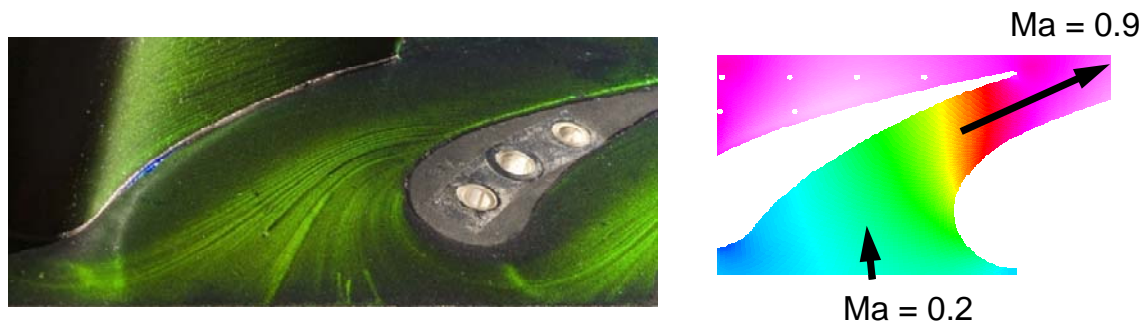


Fig. 1: Flow visualization traces (left) and isentropic Mach numbers on the turbine vane platform (right)

A convenient alternative to TLC-thermography for film cooling experiments would be to exploit the often unwanted temperature dependence of PSP-technologies. Thus, the objective of the present paper is to study the possibility to indicate temperature changes due to heat transfer at the wall with the aforementioned PSP-sensor.

METHODOLOGY

When photoluminescent paint is illuminated with UV-light, the light sensitive molecules are excited to higher energy states. This absorption of energy is followed by a reset to the initial state by the emission of visible, red-shifted light (fluorescence) as well as other types of energy transfer. One such transfer mechanism is oxygen quenching where the presence of oxygen molecules reduces the amount of emitted light. Thus with increased static air pressure the number of oxygen molecules on the PSP sensor increases and the emitted light decreases. The competing actions of oxygen quenching and emission of light allows for optical full-surface measurements of the oxygen content on the surface. A vacuum chamber is employed to calibrate the PSP-sensor through the range of pressures and temperatures encountered in the flow experiments. Pressure sensitivity is measured in terms of the change in luminescent light intensity relative to the light emitted at a known reference condition (e.g. ambient) as depicted in Fig. 2. On the left hand side, it can also be seen that the emitted light goes down when the temperature is increased.

Temperature sensitivity of PSP-systems originates from several combined mechanisms. In this work, the net temperature sensitivity will be considered without going into details of the different mechanisms. This is motivated by the fact that the PSP formulation employed here has properties that approach those of an “ideal paint” according to the concept introduced by Puklin et al. (2000). This means that the pressure and temperature sensitivities can be considered decoupled. In other words, the oxygen pressure sensitivities at different temperatures collapse onto a single curve when the reference intensity is obtained at the same temperature (isothermal) as the other readings. This has been demonstrated by e.g. Narzary et al. (2010) and is illustrated with the present calibration data in Fig. 2 (right).

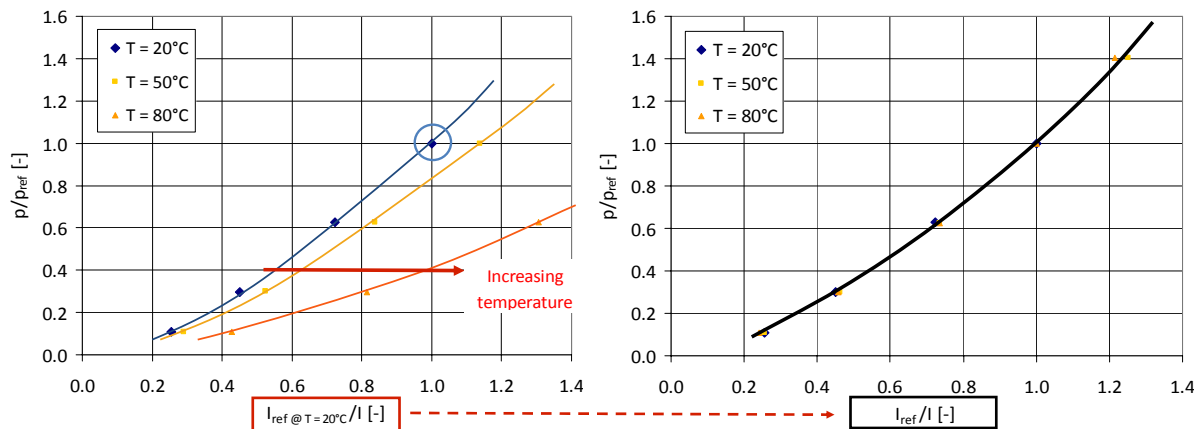


Fig. 2: Pressure sensitivities of PSP reference intensity is measured at $T = 20^{\circ}\text{C}$ (left) and the common pressure calibration resulting from iso-thermal reference cases (right)

Similarly, the temperature sensitivity is almost constant over the encountered pressure range as depicted in Fig. 3. It can be seen that the measured photoluminescence goes down with increased temperature. In the first instance, the temperature sensitivity of the PSP-sensor can be linearized within the restricted temperature range employed in the current study ($T - T_{ref} = \pm 15^{\circ}\text{C}$). This allows for a convenient correlation of the measured intensities by a single (first order) temperature calibration-constant:

$$D = \frac{I_{ref}/I - 1}{T - T_{ref}} \quad (1)$$

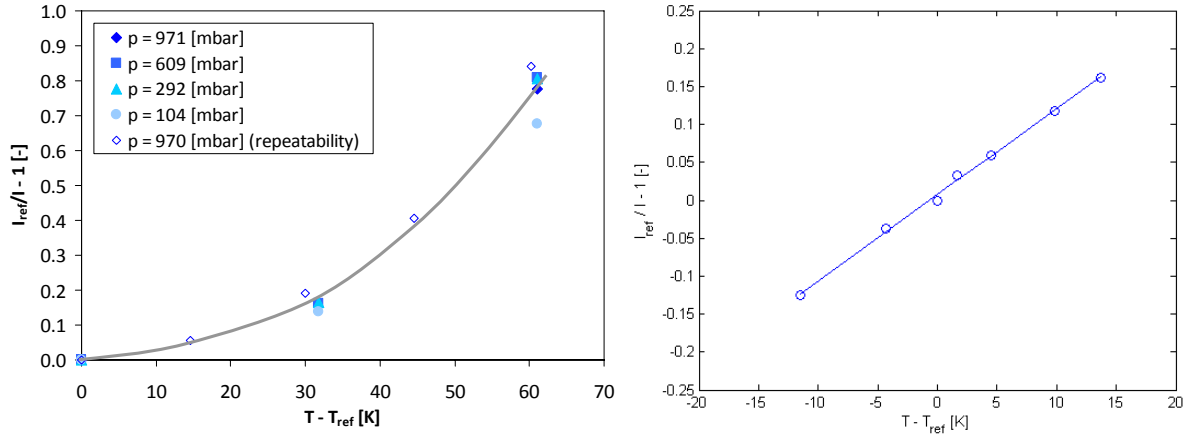


Fig. 3: Temperature sensitivities of PSP at constant pressure (left). Linearized calibration (right).

Transient heat transfer measurements using Temperature Sensitive Paint (TSP) have previously been carried out by e.g. Kingsley-Rowe et al. (2003). In the present work, a transient heat transfer technique using a heated foil is employed. A detailed description of the test set-up is given by Jonsson (2010). The principle of the transient technique is to observe the time response of the surface temperature when a heating step is imposed. The foil, made of stainless steel is glued on the wall surface using a thin double sided scotch tape with high bonding properties (3M VHB®). Before performing a transient test, the cascade is exposed to airflow until thermal equilibrium is reached. It is then assumed that each position of the wall takes the local recovery temperature neglecting any lateral conduction effects in the Perspex wall. This pre-conditioning sets the local initial (adiabatic) temperature according to the recovery temperature as sketched in Fig. 4.

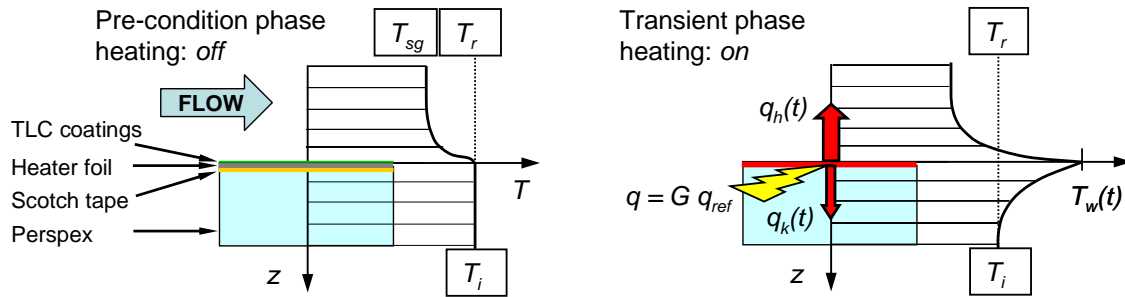


Fig. 4: Adiabatic wall (left) and transient heating situation (right)

Heating the wall surface allows for the determination of a convective heat transfer coefficient. Again, the convective heat flux at the wall is normalized by the driving temperature difference defined as the difference between wall surface temperature at heated and adiabatic states. In contrast to the heat transfer situation in steady-state, heat flux and wall temperature are now time dependent:

$$h = \frac{q_h(t)}{T_w(t) - T_{aw}} \quad (2)$$

Thus, h is assumed to remain constant in time during the transient experiment. In order to determine the heat transfer coefficients from the measured time-temperature pairs, a one-dimensional analytical conduction model for the semi-infinite wall is applied. The governing equation is:

$$\frac{\partial T}{\partial t} = \left(\frac{k}{\rho c} \right)_{wall} \frac{\partial^2 T}{\partial z^2} \quad \text{for } 0 \leq z < \infty \quad (3)$$

With the initial condition:

$$T(z, t = 0) = T_i, \quad (4)$$

the semi-infinite boundary condition:

$$T(z \rightarrow \infty, t) = T_i \quad (5)$$

and the boundary condition for the heat flux at $z = 0$ as determined from an energy balance including the thermal inertia of the heater foil:

$$q = -k \frac{\partial T}{\partial z} \Big|_{z=0} + h(T_w(t) - T_{aw}) + \beta \frac{dT_w}{dt}, \quad (6)$$

the solution for the non-dimensional wall temperature evolution according to von Wolfersdorf et al. (1993) is:

$$\begin{aligned} \theta(t) &= h \frac{T_w(t) - T_{aw}}{G(t) q_{ref}} \\ &= 1 + h \left(\frac{e^{A^2 t} \operatorname{erfc}(A\sqrt{t})}{\beta A(A-B)} - \frac{e^{B^2 t} \operatorname{erfc}(B\sqrt{t})}{\beta B(A-B)} \right) \end{aligned} \quad (7)$$

where

$$A = \frac{\sqrt{(\rho ck)_{Perspex}}}{2\beta} \left(1 + \sqrt{1 - \frac{4h\beta}{(\rho ck)_{Perspex}}} \right) \quad (8)$$

$$B = \frac{\sqrt{(\rho ck)_{Perspex}}}{2\beta} \left(1 - \sqrt{1 - \frac{4h\beta}{(\rho ck)_{Perspex}}} \right)$$

and the heat storage capacity of the heater foil is taken into account in the boundary condition. It is treated as a scalar constant (lumped model) as follows:

$$\beta = (\rho c)_{steel} \delta_{foil} \quad (9)$$

As depicted in Fig. 5, the convective heat flux is a part of the supplied area-specific heat from the electrically heated foil:

$$q_h(t) = q - q_k(t) - q_{foil\ storage}(t) \quad (10)$$

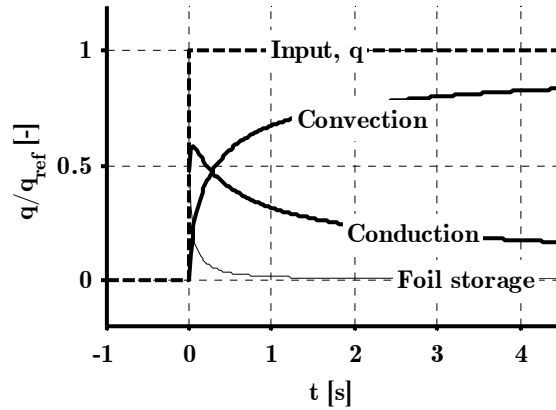


Fig. 5: Theoretical transient evolutions of heat flux terms ($h = 500 \text{ W/m}^2\text{K}$)

The local area-specific heat release $q(x,y)$ depends only on the heater foil and produces the same distribution in each experiment with a given foil set-up. Thus the distribution of heating is independent of the global power setting (G). Here, the technique is based on the non-dimensional temperature evolution (Equation 7). The history of the time derivative (shape of the temperature evolution) allows to identify h which in turn quantifies the heat release. However, the latter requires low noise levels and well calibrated sensors which are not always achievable. By adding a second heating phase over which the power is reduced by a fraction of the initial input (ε), the effect of the heat transfer coefficient emerges more clearly than with a single transient step. This effect can be seen in the relative temperature evolutions of the 1st and 2nd phases as depicted in Fig. 6. The solution for the non-dimensional wall temperature evolution for the 2nd heating phase is:

$$\begin{aligned}
 \theta(t) &= h \frac{T_w(t) - T_{aw}}{G(t)q_{ref}} \\
 &= \varepsilon + h \left(\frac{e^{A^2 t} \operatorname{erfc}(A\sqrt{t})}{\beta A(A-B)} - \frac{e^{B^2 t} \operatorname{erfc}(B\sqrt{t})}{\beta B(A-B)} \right) \\
 &\quad + (\varepsilon - 1) h \left(\frac{e^{A^2(t-t_s)} \operatorname{erfc}(A\sqrt{(t-t_s)})}{\beta A(A-B)} \right. \\
 &\quad \quad \left. - \frac{e^{B^2(t-t_s)} \operatorname{erfc}(B\sqrt{(t-t_s)})}{\beta B(A-B)} \right) \quad t > t_s
 \end{aligned} \tag{11}$$

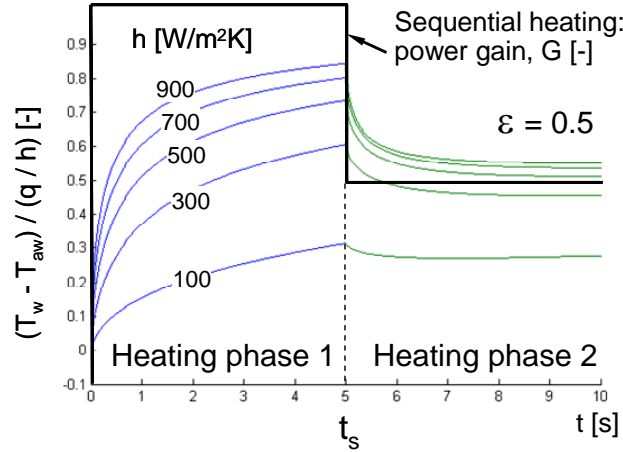


Fig. 6: Transient wall temperature response according to the analytical model

Employing the linearized, first-order temperature sensitivity (D) of the PSP sensor gives a measure of the temperature difference between the initial (adiabatic) and a time instance during the heating phase:

$$(T_w(t) - T_i)_{PSP} = \frac{I_i/I(t) - 1}{D} \tag{12}$$

The corresponding temperature change of the analytical model is given by Equations (7) and (11) for the first and second heating phases respectively. The following expression is valid for any of the heating phases:

$$(T_w(t) - T_i)_{1D\text{-model}} = \frac{q}{h} \theta(h, t) \tag{13}$$

With the two unknowns h and q , and a large number of data points N from the transient experiment, the problem is over-defined and a solution that satisfies Equation (12) and Equation (13) in an optimum way is sought. For this purpose, a regression analysis similar to that of Vogel et al. (2003) is employed. Because the original method also solved for the adiabatic film cooling effectiveness as a 3rd parameter, a simplified version is applied for the present situation. Thus the decoupling of the adiabatic film cooling effectiveness (concentration measurements) from the heat transfer measurements is one of the advantages of PSP.

The two unknowns can be combined by expressing the heat release as a function of the heat transfer coefficient. Thus, the difference between measured and theoretical temperature changes is minimized in each pixel by solving for the heat transfer coefficient that produces the optimal match to all data of the transient experiment. Practically this is done by performing a least-square fit of the analytical 1D model to the measured temperature history (200 points) using vector notation. The remaining error is stated as the standard deviation calculated with vector notation according to:

$$\sigma_{T_w} = \sqrt{\frac{\left\| \left(\frac{I_i/\vec{I}(t) - 1}{D} - \frac{q(h)}{h} \vec{\theta}(h, t) \right) \right\|^2}{N}} \tag{14}$$

Thus the heat transfer coefficient (h) is the primary quantity of interest. In order to derive the absolute heating input (q) from the PSP thermography experiments, an accurate pixel-by-pixel calibration of the temperature sensitivity D is required. This should be done regularly since it is known that the PSP molecules

undergo some curing and ageing depending on the temperature history. Due to the difficulty to obtain accurate pixel-by-pixel calibrations of the temperature sensitivity, a globally averaged value of the temperature sensitivity D of the PSP is employed. As a consequence, the resulting heat release distribution is based on the averaged constant temperature sensitivity. Thus with the heat release,

$$q(h) = \frac{(T_w(t) - T_i)_{PSP} \cdot \left(\frac{\theta(h,t)}{h}\right)}{\left\|\frac{\theta(h,t)}{h}\right\|^2}, \quad (15)$$

Fig. 7 shows examples of measured wall temperature evolutions and the agreement with the theoretical solution (blue dashed curve).

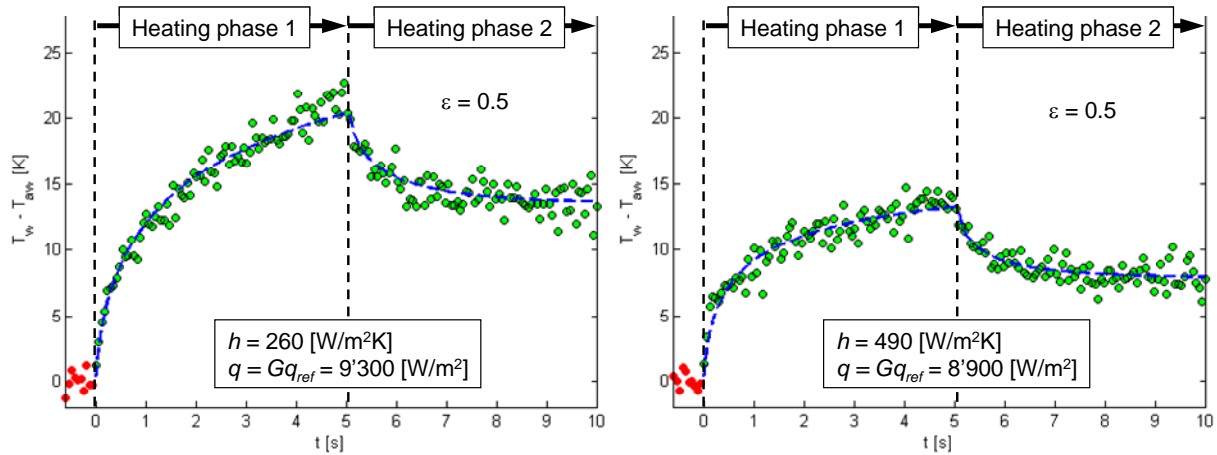


Fig. 7: Measured wall temperature response using PSP (single camera pixel).

RESULTS AND DISCUSSION

Fig. 8 shows the two resulting quantities involved in the present wall heat transfer problem. As intended during the set-up of the experiment, the heating distribution (q) has been successfully focused towards the downstream region by adjusting the electrical connectors. This was done since the highest values of h were expected in this region and, consequently larger amounts of heat flux were required to obtain optimal temperature differences for the measurements.

There is no noticeable cross influence of the heat transfer coefficient visible in the q -result. However, the particular thermal boundary conditions obtained with the heated foil can lead to local extremes in heat transfer coefficients as expected. This effect is visible in areas exposed to under- and over-heated incoming boundary layers leading to low and high heat transfer coefficients respectively.

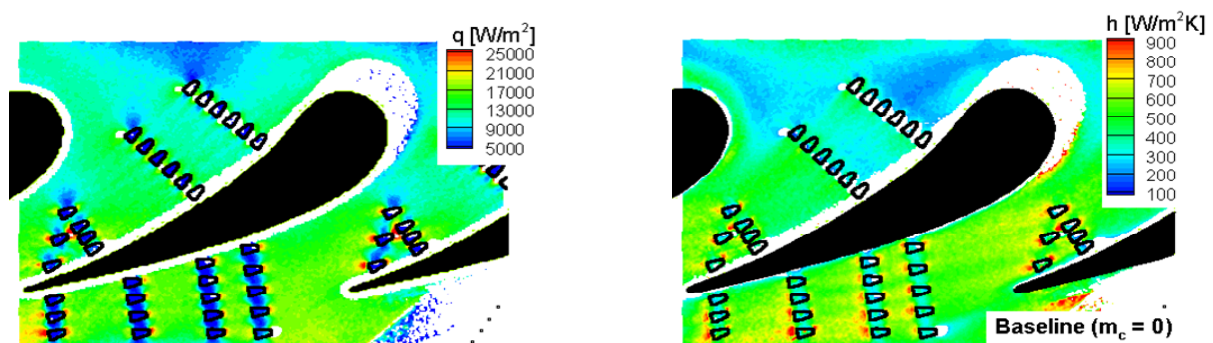


Fig. 8: Distribution of electric heating (left) and convective heat transfer coefficient (right)

Results with excessively high variation are filtered out according to the following criterion on the standard deviation of the wall temperature:

$$\sigma_{\theta_w} = \frac{\sigma_{T_w}}{q/h} > 0.2 [-] \quad (16)$$

Locally high transient temperature rise close to some of the cooling holes makes the linear temperature dependence of PSP a poor approximation. A quadratic calibration function would increase the accuracy for such cases. Those pixels as well as regions shadowed by the airfoil of the turbine blade are blanked out from the surface plots. This is done with the constraint given in Equation (16). For the test case presented here, the underlying assumption of ideal paint properties lead to bias errors that remain within the average *random* uncertainty of $\Delta h/h = \pm 7.5\%$ evaluated by Jonsson (2010). The approximation of a first order relation between temperature and signal intensity is also within this range as long as the heating power is limited like in the present experiments. Finally it is recommended to control repeatability and the time between calibration and flow experiments in order to minimize influence of paint degradation due to UV-light exposure, thermal cycling, frictional wear of the gas flow, humidity, etc. Those factors can have varying influence on the PSP-paint and are not treated in this work. It is generally recommended to minimize the exposure to degradation sources.

CONCLUSIONS

Given the advantages of PSP in adiabatic film cooling measurements, a practical method to obtain convective heat transfer coefficients is sought. It turns out that the temperature sensitivity of the tested PSP-formulation is sufficiently high to allow also for heat transfer measurements. Promising results with the proposed technique have been achieved on an engine realistic turbine vane platform. Advantages are full surface coverage with high spatial resolution important in film cooling studies. Measuring relative temperature changes caused by heat transfer eliminates the need to measure the absolute initial temperature (T_{av}) using complementary techniques or correlations as with narrow-band TLC. Moreover, the time required for measurements and data reduction is relatively short. Finally, use of the same PSP-coating ensures consistent surface quality during the measurement of all searched quantities – notably the adiabatic film cooling effectiveness and heat transfer coefficients. The limitations of the proposed measurement technique consist in near-ambient temperature environments, relatively small allowable changes in wall temperature which in turn lead to relatively high noise levels. Due to possible sensor degradation, calibrations and the repeatability of the measurements should be controlled.

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NOMENCLATURE

Roman

c	specific heat of solid	[J/(kgK)]
D	temperature calibration constant	[-]
G	electric heating power gain	[-]
h	local heat transfer coefficient	[W/(m ² K)]
I	intensity (PSP-measurements)	[-]
k	thermal conductivity	[W/(mK)]
Ma	isentropic Mach number	[-]
N	number of data points	[-]
p	pressure	[Pa]
q	heat flux	[W/m ²]
T	temperature	[°C, K]
t	time	[s]
x, y	surface coordinates	[m]
z	wall-normal ordinate	[m]

Greek

β	thermal inertia of heater foil	[J/(m ² K)]
δ	wall thickness	[m]
ε	factor of heating power reduction	[-]
θ	non-dimensional wall temperature	[-]
ρ	density	[kg/m ³]
σ_θ	standard dev. of non-dim. temperature	[-]

Superscripts

\rightarrow indicating a vector of values

Subscripts

1	1st heating phase
2	2nd heating phase
a	adiabatic
g	main stream gas
h	convection
i	initial
k	conduction
r	recovery
ref	reference case
s	static flow condition, switching time
w	wall

Abbreviations

PSP	Pressure Sensitive Paint
TLC	Thermochromic Liquid Crystals
TSP	Temperature Sensitive Paint