Heated Thin Film Gauge Arrangements for Transient Heat Transfer Measurements

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ABSTRACT

This paper describes the development of heated double-sided thin film gauge configurations for transient heat transfer measurements. By heating the substrate it is possible to measure the heat flux over a range of surface temperatures and deduce the adiabatic wall temperature and the external heat transfer coefficient. The accuracy of the measurement depends on the stability of the regression of heat flux against wall temperature and can be improved by extending the range of wall temperature over which the regression is performed. In this paper we compare two methods of local heating: double-sided gauges with an underside thin film heater and self-heating double-sided gauges. Both arrangements have been used in the Oxford Turbine Research Facility to measure the heat transfer on the uncooled turbine shroud of the MT1 high-pressure turbine stage at engine-representative conditions. These measurements yield improved regressions compared to conventional techniques to determine the adiabatic wall temperature and the heat transfer coefficient.

NOMENCLATURE

A_g	Gauge area
α	Temperature coefficient of resistance
α	Thermal diffusivity
c	Specific heat capacity
d	Gauge track width
h	Heat transfer coefficient
I	Current
k	Thermal conductivity
l	Gauge track length
$\dot{q}$	Heat flux
R	Resistance
r_H	Heating source radius
ρ	Density
S	Stability criterion
T	Temperature
t	Time
V	Voltage
x	Depth / Substrate thickness

INTRODUCTION

Platinum thin film resistive gauges are used to measure surface temperature on account of their high temperature coefficient of resistance. Where substrate thermal properties are well-known, surface heat flux can be deduced from the time response of the surface temperature trace. Thin film gauges are also widely used for transient heat transfer measurements where high frequency response is required, due to their low thermal mass.

Thin film gauges have been developed for many years at the Osney Laboratory and are extensively used in rotating turbine test rigs, where instrumentation is difficult due to the limited access [1]. In recent years there has been an increasing interest in the study of cooled engine parts, including the measurement of film effectiveness and cooling flow redistribution. In many scaled experiments, regions of high film effectiveness are particularly challenging for heat transfer measurements due to the low driving temperature difference between gas and wall, which makes regressions of heat flux as a function of wall temperature (performed to determine the adiabatic wall temperature and the heat transfer coefficient) less stable. In these regions, methods for artificially varying the wall temperature (by heating or cooling) can be used to improve the stability of the regression, i.e. increase the temperature range, enhancing the accuracy with which the adiabatic wall temperature and heat transfer coefficient and can be measured.

Improving the accuracy of this regression is essential for experiments in which the film effectiveness is to be determined. There are three key challenges for the measurement of the heat transfer on cooled engine parts using thin film gauges: to increase the spatial resolution of the measurements by improving the manufacturing technology, to reduce the uncertainties arising from the assumption of a simplified geometry to model the thermal conduction in the gauge substrate (e.g. a semi-infinite layer), and to develop methods to accurately determine the Nusselt number and film cooling effectiveness in regions with low heat transfer driving temperature difference.

Double-sided thin film gauges consist of two thin film temperature gauges mounted on either side of an insulating layer of known thickness and thermal properties. Single-sided gauges, on the other hand, consist of just one gauge on an insulating layer. Although single-sided gauges are easier to manufacture, double-sided thin film gauges have distinct advantages over single-sided gauges. Firstly, double-sided gauges do not require a 1D assumption of the heat conduction into the wall, and therefore any knowledge of the wall thermal properties. On a thin-walled component, such as a cooled aerofoil the assumption of a semi-infinite substrate is usually not valid, making the processing of data from single-sided gauges highly complex. Another strong advantage of double-sided thin film gauges is that the temperature difference can be determined directly at a point of interest without the need for additional thermocouples.

Double-sided thin film gauges were originally introduced by Epstein et al. [2] and have become widely used with improved gauge manufacturing technology. More recently, the Osney Laboratory has been active in miniaturizing thin film gauge technology allowing greater spatial resolutions to be achieved [3].

In this paper two new gauge concepts are presented, which combine double-sided thin film gauges with the possibility to heat the substrate layer and thus improve the stability of the regression. The advantage of locally heating the gauge substrate instead of changing the temperature of the entire turbine part is that the heating system is not limited by spatial constrains in the part. Moreover, it is difficult to achieve the uniform initial surface temperature required to carry out heat transfer measurements, given the thermal losses to the ambient and neighbouring components due to insulation difficulties. With heated gauges an improved regression can also be obtained without the need for complex heating systems.

DEVELOPMENT OF DOUBLE-SIDED TFGS FOR HEAT TRANSFER MEASUREMENT

The theory of thin film surface resistance gauges for heat transfer measurements is well documented [1,4]. The resistance of a thin film gauge increases linearly with temperature:

$$R(T) = R_0(1 + \alpha\Delta T),$$

where R_0 is the resistance at a reference temperature T_0 and α is the temperature coefficient of resistance. Defining the change in the resistance of the thin film gauge $\Delta R = R(T) - R_0$ and the change in surface temperature $\Delta T = T - T_0$ it follows that

$$\Delta T = \frac{\Delta R}{\alpha R_0}.$$

For a constant gauge current I, the temperature history of the gauge can be obtained by recording the potential difference ΔV across the thin film gauge.

In a short-duration test the penetration depth of the thermal pulse into the insulating layer is small compared to other dimensions and hence the insulating layer can be taken to be homogenous and isotropic in the lateral directions. In particular, the heat flux into the wall is much larger than any lateral heat flux. Assuming furthermore that the effect of the gauge is negligible, i.e. that it has negligible thermal resistance, the equation governing the temperature distribution in the insulating layer is the 1D Fourier equation:

$$\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha}\frac{\partial T}{\partial t}$$

with the thermal diffusivity $\alpha = k/\rho\, c$.

As long as the penetration depth of the thermal pulse is sufficiently small that a semi-infinite analysis can be performed, this equation can be used to infer the surface heat flux from the surface temperature [1,4]. While it is difficult to calculate the time-dependent heat flux analytically, the surface temperature signals can be processed into heat transfer rates by the use of analogue electronic circuits [5]. More recently, impulse response processing of temperature signals has been introduced [6] (see appendix).

The design of the thin film gauges employed in this study is based on the work by Collins et al. [3]. The aim of the gauge design is to optimise the gauge sensitivity by maximising the l/d ratio of the thin film, where l is the gauge length and d the width of the gauge. The ratio can be increased through the use of long thin films in serpentine patterns in order to increase spatial resolution. The final gauge design and the manufactured result are depicted in Figure 1. Details on the manufacturing process are given by Collins et al. [3]. Double-sided thin film gauges can be fabricated from a pair of single-sided thin film gauges glued together. The Kapton and glue layers were found to have very similar thermal properties and will be regarded as one layer in the following. The optimum thickness of the insulating layer between the top and bottom gauges was determined as approximately 100 μm based on the modelling of the thermal pulse through the double-sided gauge [7].

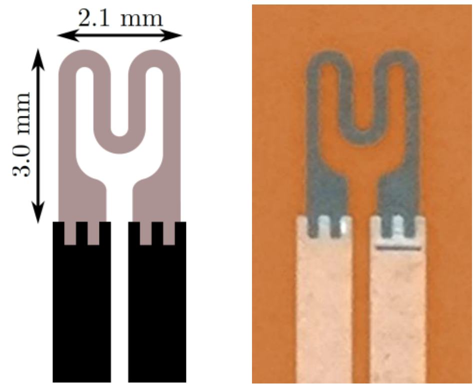

Figure 1: Gauge dimensions and photograph.

The thin film gauges were calibrated in a water bath, in which the resistance of the gauges at different temperatures was recorded to obtain the temperature coefficient of resistance α. The effect of changes in lead resistance on the temperature coefficient prior to the test was accounted for.

DOUBLE-SIDED TFGS WITH AN UNDERSIDE THIN FILM HEATER

The reason of using heated thin film gauges is to heat the gauge substrate in order to increase the temperature range over which the extrapolation to obtain the adiabatic wall temperature and heat transfer coefficient is performed. The first gauge concept presented here is a double-sided thin film gauge with a separate local heating system.

I. Principle of operation

For an unheated gauge test the initial voltage V_i across each gauge is recorded prior to the run (a few minutes beforehand) to acquire an offset signal at a known component temperature. The initial component wall temperature T_{wi} is measured at the same time by nearby thermocouples. Given the 45 min time passing between tests, isothermal conditions can be assumed. The voltage across a gauge is then recorded during the test time. The change in temperature follows from the change in resistance according to

$$\Delta T = T - T_{\mathrm{wi}} = \frac{(R - R_{\mathrm{i}})\,[1 + \alpha(T_{\mathrm{wi}} - T_{\mathrm{cal}})]}{\alpha R_{\mathrm{i}}},$$

where T_{cal} corresponds to the temperature at which the temperature coefficient of resistance α was determined. The gauge current I stays constant $I \approx I_i$, thus the temperature difference can be expressed as a change in voltage

$$\Delta T = \frac{(V - V_{\mathrm{i}})\,[1 + \alpha(T_{\mathrm{wi}} - T_{\mathrm{cal}})]}{\alpha V_{\mathrm{i}}}$$

and no knowledge of the exact current is required. This calculation is valid for single-sided as well as double-sided thin film gauges.

When taking heat transfer measurements with double-sided gauges with a separate heating source, the temperature of the part is recorded before the heating starts. The heating of the gauge substrate is captured by the increase in voltage from the initial voltage to the voltage recorded just prior to the run. The post-processing is the same as described above. The principle of operation for measuring the surface temperature is illustrated in Figure 2.

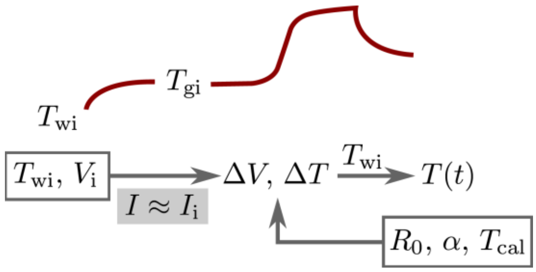

Figure 2: Principle of operation to measure the surface temperature history for a heated thin film gauge arrangement with a separate heating system.

II. Gauge construction

The first system investigated to heat the gauge substrate consists of a resistive heater element mounted on a Kapton sheet and placed underneath the thin film gauges on the surface of the part to be measured. The bespoke heater element has the advantages that it provides uniform heating, does not require space in the wall (as a cartridge heater would), and thus does not affect the structural strength of the component, making it a good choice for a heavily cooled vane. However, the disadvantage of this concept is that it requires an additional insulating layer. Figure 3 shows the gauge configuration with an underside thin film heater.

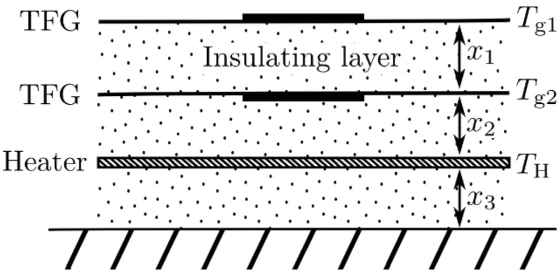

Figure 3: Heated thin film gauge arrangement with an underside thin film heater.

A heater design was developed with a spiral-shaped copper track. The thickness and spacing of the tracks was limited by the accuracy of the etching process to 0.2 mm. An example of a heater design is shown in Figure 4. This heater has a surface area of 525 mm^2 and a resistance of 8.1 Ω. Depending on the length of the heater tracks and the thickness of the copper, a wide range of resistances can be achieved. The heating power is limited by the outgassing temperature of the glue, which resulted in a maximum heating heat flux of 100 kWm^{-2} for a typical gauge configuration and a maximum preheating temperature difference of 100 K. For a certain heater power, the preheating temperature difference increases as the thickness of the insulating layer between the heater and the wall increases and as the thickness of the insulating layer between the heater and the bottom gauge decreases. In the study presented here all three substrate layers have the same thickness of 100 µm.

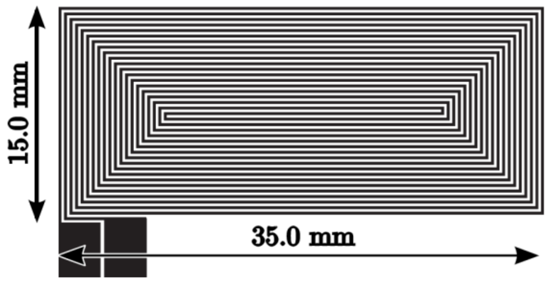

Figure 4: Example of a heater design.

III. Modelling and bench test validation

As the lateral extension of the thin film gauges is small compared to that of the heater element, the heat transfer in the gauge system depicted in Figure

3 is considered one-dimensional and is described by the following equations:

$$\dot{q}_{\mathrm{conv}} = h(T_{\mathrm{g1}} - T_{\infty}) = k\frac{T_{\mathrm{g2}} - T_{\mathrm{g1}}}{x_1}$$
$$k\frac{T_{\mathrm{g2}} - T_{\mathrm{g1}}}{x_1} = k\frac{T_{\mathrm{H}} - T_{\mathrm{g2}}}{x_2}$$
$$k\frac{T_{\mathrm{H}} - T_{\mathrm{g2}}}{x_2} = \dot{q}_{\mathrm{H}} - k\frac{T_{\mathrm{H}} - T_{\mathrm{w}}}{x_3},$$

where h and k are respectively the surface heat transfer coefficient and the thermal conductivity of the Kapton-glue laminate.

The heated thin film gauge arrangement with an underside thin film heater element was bench tested over a range of heater current settings. The preheating surface temperatures were measured by the double-sided thin film gauges and compared to the modelling results (see Figure 5). The current through the heater was kept constant for the duration of each test and increased stepwise from 0 A to 1.6 A. Two consecutive tests were carried out to investigate repeatability. Figure 5 shows the temperature dif-ference relative to the backwall temperature as measured by two thin film gauges. The agreement between two consecutive tests is excellent. The difference between the two thin film gauges is proportional to the total preheating temperature and is at most 2 K. While this temperature difference lies within the expected temperature uncertainty (see uncertainty analysis) it could also point to slightly uneven heating.

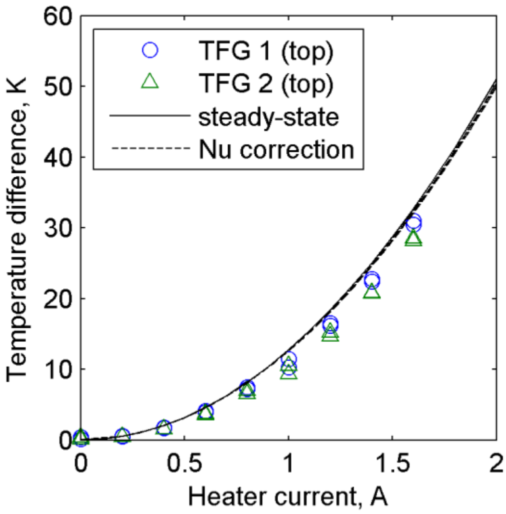

Figure 5: Preheating top gauge temperature differences over increasing heater current for 18 consecutive tests and comparison to analytical results.

The steady-state model results were obtained for an isothermal top layer, i.e. $T_{\mathrm{g1}} = T_{\mathrm{g2}} = T_{\mathrm{H}}$. To account for the heat dissipation into the environment a correlation for the Nusselt number for a surface in natural convection was used. The Nusselt number Nu can be expressed as a function of Grashof number Gr and Prandtl number Pr [8]

$$\mathrm{Nu} = C\,(\mathrm{Gr\,Pr})^n,$$

where C and n are two empirical factors, which depend on the geometry and the flow conditions. For a heated horizontal plate the Nusselt number can be calculated for a given range of surface temperatures to obtain the convective heat transfer coefficient h. It was found that the natural convection, and hence Nusselt number, were very low. Therefore, the gauge substrate is nearly isothermal and the Nusselt number correction is very small. This prediction is in agreement with the observation that the temperatures of the bottom gauges were very similar to those of the top gauges. Overall, the measured temperatures agree well with the model predictions given the experimental uncertainties in backwall temperature and thickness of the insulating layers.

IV. Rig testing results

Transient rig tests were carried out in the Oxford Turbine Research Facility (OTRF). The OTRF is a short duration piston tunnel capable of testing engine-sized high-pressure (HP) turbine stages at engine-representative conditions for aerodynamic and heat transfer measurements. The fundamental operation of this type of facility was first described by Jones et al. [9]. The aim of these tests was to demonstrate the accuracy improvement in the regression of heat flux as a function of wall temperature – performed to determine the adiabatic wall temperature and the heat transfer coefficient – obtained by locally heating the substrate.

During this test campaign the OTRF was run without the turbine stage, i.e. with the HP NGV and rotor removed and replaced with a simple annular nozzle configuration. The gauges were placed on the casing endwall, just after what corresponds to the blade over-tip region. These tests were performed under uniform inlet conditions with an inlet total temperature of 412 K and Mach number M = 0.9 at the throat (just upstream of the gauge location).

The temperature traces for one double-sided gauge with an underside thin film heater are shown in Figure 6. The dashed lines indicate the stable run period used for the data regression. The temperature achieved by the gauges before the run depends on the amount of preheating. The heater is switched on a few minutes before the run after taking the measurement of the reference voltage at isothermal conditions. This voltage is used to compute the preheating temperature difference. Note also that the heater was not switched off until the test run was finished. A maximum preheating surface temperature difference of 80 K was achieved. Although not shown here, at identical heater settings the results were repeatable. The surface temperature drops just before the run when ambient air accumulated in the rig is blown out. The temperatures of top and bottom gauges agree with each other within expected experimental uncertainties and agree well with model predictions.

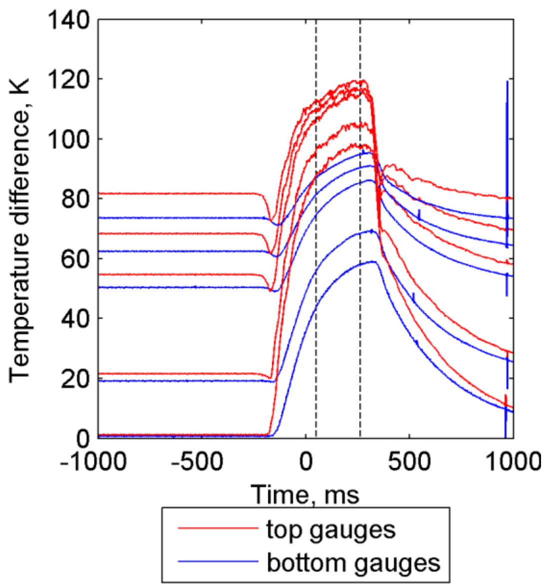

Figure 6: Temperature traces for one double-side gauge with an underside thin film heater (one unheated run and four heated runs).

The convective heat flux is obtained from the temperature history using the impulse response method [6]. The preheating temperature difference is set to zero at the start of the run, because the heat flux resulting from this temperature difference reflects the effect of heating and should therefore be subtracted from all subsequent measurements. This procedure ensures the heat flux is zero at the start of the run. Crucially, uncertainties in the initial temperature difference between top gauge and bottom gauge do not affect the measured convective heat flux.

The "reduced" heat flux traces for three test runs conducted at different heater current settings are shown in Figure 7. At higher preheating temperatures the heat flux during the run decreases and thus the range for the regression is increased in comparison to the unheated double-sided gauges. The regression is performed on data from multiple runs over the stable run period taking into account any changes in the inlet total temperature T_{01} [10]. The regression data for a heated and unheated test case are shown in Figure 8. The dashed vertical lines mark the regression range. The adiabatic wall temperature was determined as 425 K and the heat transfer coefficient as 1397 Wm^{-2}K^{-1}.

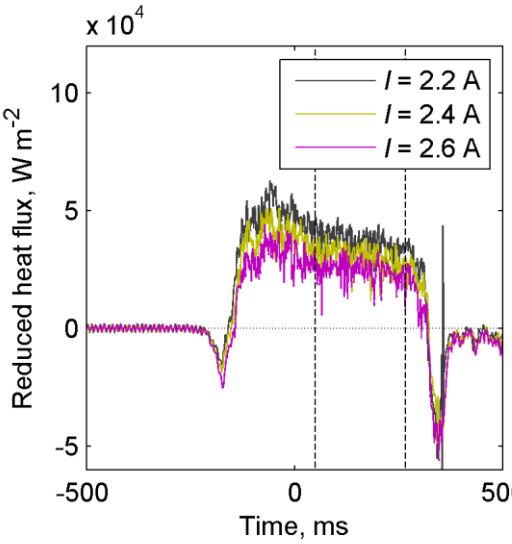

Figure 7: Heat flux traces for one double-sided gauge with an underside heater.

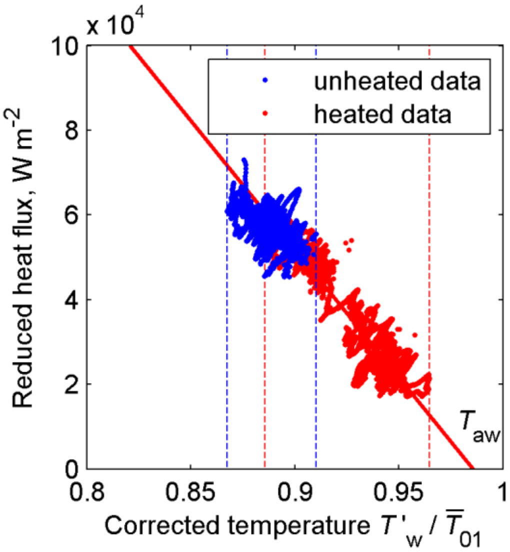

Figure 8: Regression data for one double-sided gauge with an underside heater compared to unheated results.

The accuracy with which the adiabatic wall temperature can be determined depends on the ratio of the outer wall temperature difference during the stable part of the run $\Delta T_{\mathrm{w}} = T_{\mathrm{w2}} - T_{\mathrm{w1}}$ and the difference between the adiabatic wall temperature T_{aw} and the first (furthest) regression point T_{w1}. The stability criterion S is introduced, where

$$S = \frac{\Delta T_{\mathrm{w}}}{T_{\mathrm{aw}} - T_{\mathrm{w1}}}.$$

For the unheated case S is as low as 0.36. For the heated case the value for the stability criterion rises to 0.79. The best results are obtained with a combination of unheated and heated data ($S = 0.82$).

SELF-HEATING DOUBLE-SIDED TFGS

The concept for self-heating gauges is to use an array of heat transfer gauges as heaters in order to achieve a uniform heating of the test surface prior to the run. To exploit the ohmic heating of the gauges, they are supplied with a higher current than usually used for the heat transfer measurements. Although in such a set-up the surface temperature peaks at the position of the heaters (i.e. the thin film gauges), the variation of the temperature on the surface is small compared to the average temperature difference between gas and surface.

At the beginning of the test the surface is in thermal equilibrium, i.e. the electrical power converted into heat by the thin film gauges equals the heat losses via conduction and convection. Since the conduction field after the beginning of the run is expected to remain nearly constant, it is possible to infer the change in convective surface heat flux during the run by measuring the surface temperature with the thin film gauges. The local heat flux into the surface can be calculated using a conventional 1D conduction analysis, provided the convective heat flux does not vary significantly between different points on the preheated surface. The measured heat transfer rate is equal to the surface heat flux minus the known pre-run heat flux.

I. Principle of operation

For thin film gauges with variable heating current through the gauge, the equation for heated gauges with an underside thin film heater cannot be used because both the current and the gauge temperature change during preheating. Therefore, heat transfer measurements require additional information on the initial gauge temperature T_{gi}. This temperature cannot be measured directly in the rig due to access limitations, but it can be infer-red from external information such as bench test data or thermal models. Once T_{gi} is known and the corresponding voltage V_{gi} has been measured, any subsequent change in temperature can be calculated:

$$\Delta T = T - T_{gi} = \frac{(V - V_{gi})\,[1 + \alpha(T_{gi} - T_{cal})]}{\alpha V_{gi}}$$

In this study, the current through the bottom gauge was kept constant and at a level that did not cause any ohmic heating. The temperature increase of the bottom gauges can therefore be calculated as for the heated gauges with an underside heater, where the initial temperature of the part is recorded under isothermal conditions before the heating starts. Using an accurate model of the heat conduction between the two gauges, it is then possible to derive the temperature of the top gauge from the bottom gauge temperature, provided the two gauges are very well aligned.

In summary, for the self-heating gauges the top and bottom temperatures are calculated in two different ways. The bottom gauge temperature trace follows from the initial temperature and the corresponding change in voltage, as for a gauge with an underside heater. However, the current through the top gauge is increased after the reference measurement is taken and knowledge of the initial gauge temperature is required to obtain the temperature trace (see Figure 9). The problem of determining the initial gauge temperature will be further discussed below.

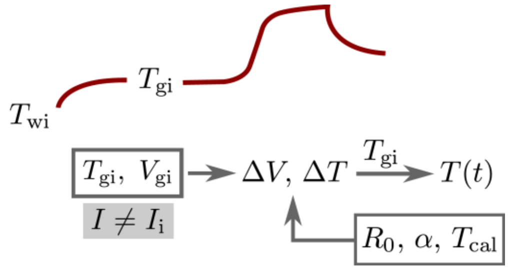

Figure 9: Principle of operation to measure the surface temperature history for self-heating thin film gauges.

II. Gauge construction

In this case the gauges self-heat by ohmic heating, hence the gauge configuration is identical to that of a conventional double-sided thin film gauge (see Figure 10). In comparison to the double-sided thin film gauges with an underside thin film heater, the total thickness of the package is reduced by the thickness of one additional insulating layer, which constitutes about one third of the package.

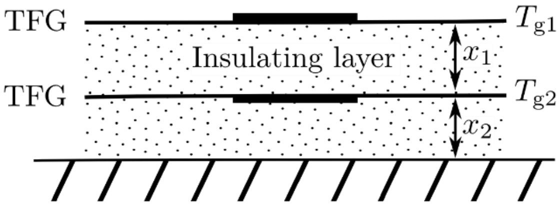

Figure 10: Self-heating gauges.

The heating of the gauges depends on the current through the gauge and the resistance and area of the gauge. The change in gauge resistance with temperature can be calculated from the temperature coefficient of resistance. As with the heater element, the heating power was limited to 100 kWm^{-2} to avoid glue outgassing. The variable current supply for the heated gauges is described in the appendix.

III. Modelling and validation

For the self-heating gauges it is necessary to determine the initial top gauge temperature, which is difficult to measure directly. If this temperature is to be inferred from another measured temperature such as the bottom gauge temperature or the wall temperature, it is necessary to describe the conduction in the gauge substrate. The heating heat flux can be calculated from the known voltage and resistance of the top gauge $\dot{q}_H = V^2/(RA_g)$, where A_g is the platinum surface area. If the preheating has reached

steady-state before the run, the temperature of the top gauge then follows from thermal conduction.
Given the gauge track width of 0.3 mm which is about three times the thickness of around 100 µm of each insulating layer, there are two possible thermal models for this multi-layer system. For either model, the platinum thickness of less than 1 µm can be neglected, as it has negligible thermal resistance. As the length of the platinum tracks is large in comparison to their width and to the thickness of the substrate layer, a 2D thermal model can be considered. This case is represented in Figure 11. The radius of the heating source is half the track width, i.e. r_H = 0.15 mm. An alternative description is obtained by neglecting the space between the tracks and taking the total gauge area as a "point" heating source with 3D thermal conduction in the gauge substrate. This representation will not be further discussed, as the 2D model achieved a better agreement with measured bottom gauge temperatures.

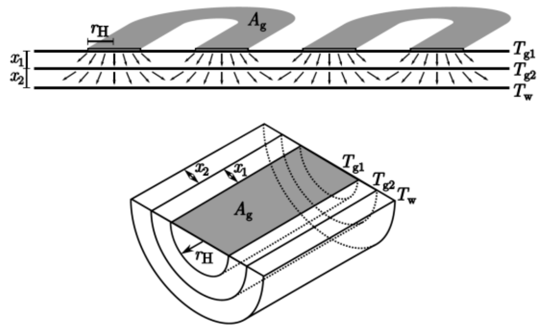

Figure 11: Representation of the gauge geometry (see Figure 1 and 10) with a 2D conduction model.

Figure 12 shows the preheating temperature difference between the bottom gauge and wall as measured in the OTRF for a range of top gauge voltages. This data serves to validate the thermal model that is used to calculate the initial top gauge temperature for a known heating heat flux (given that there are no heat losses into the rig at vacuum prior to the run). The agreement between the four self-heating gauges is very good. The experimental results are compared to the predictions from the 2D and 3D models. In contrast to the 3D model, the prediction of the 2D model is in line with the observed bottom gauge temperature. Up to 2.5 V the agreement between the 2D model prediction and the experimental bottom gauge temperature differences is good (within experimental uncertainties). At higher voltages the bottom gauge temperature does not increase proportionally to V^2 and a flattening of the temperature curve is observed. This effect can be partially explained by the increase in gauge resistance with increasing gauge temperature, which leads to a decrease in the heating heat flux, and was taken into account for the post-processing. The uncertainty analysis gives an estimate of the temperature difference error introduced by the need to make model assumptions to relate the temperatures of top and bottom gauges.

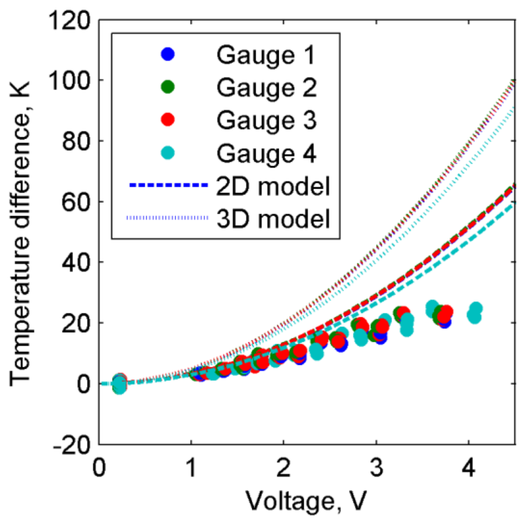

Figure 12: Bottom gauge temperature difference for different gauge settings plotted over voltage across the top gauge and superimposed to the 2D and 3D thermal conduction model results.

IV. Rig testing results

Identical transient rig tests were carried out in the OTRF with self-heating double-sided thin film gauges as described previously for the double-sided gauges with an underside heater. Both types of gauges were mounted at the same axial location in the annular nozzle, and as no circumferential variation in the flow was expected, this allowed a direct comparison between the two configurations.

The temperature traces for the same double-sided gauge with a self-heated top gauge tested over

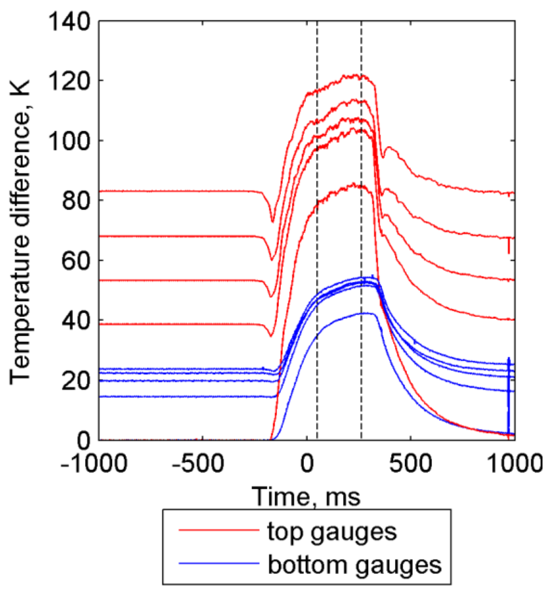

Figure 13: Temperature traces for one self-heating gauge (one unheated run and four heated runs).

a range of heating power settings are shown in Figure 13. Preheating surface temperature differences of up to 80 K were achieved. As for the double-sided thin film gauges with an underside heater, the heat flux is calculated using the impulse response method with the temperature difference set to zero at the start of the run. The resulting reduced heat flux is equal to the convective heat flux alone, i.e. the surface heat flux minus the pre-run heat flux. With increasing gauge current, the surface temperature increases and the heat flux decreases.

The regression data for a heated and unheated test case from one self-heating double-sided thin film gauge are presented in Figure 14. The heated results agree well with those of the unheated double-sided thin film gauge. A wide temperature range is covered with the heated runs and a very good regression is achieved, which enables the adiabatic wall temperature and the heat transfer coefficient to be determined separately.

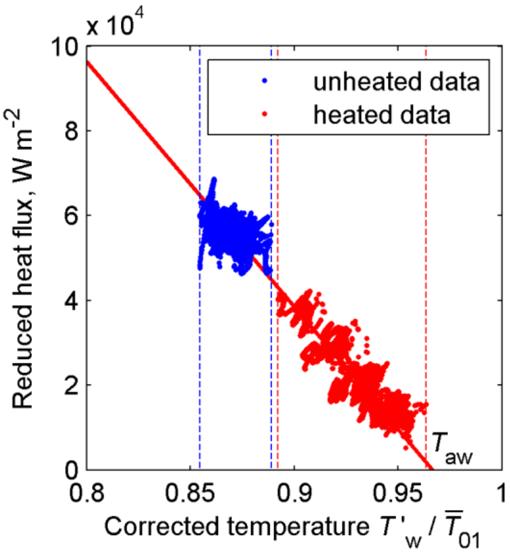

Figure 14: Regression data for one self-heating double-sided gauge compared to results from unheated double-sided thin film gauges.

The heat transfer results between the two heated double-sided gauge configurations were very similar. The adiabatic wall temperature and heat transfer coefficient measured with the self-heating double-sided gauges were 417 K and 1333 Wm^{-2}K^{-1} respectively. These compared well to the values of 425 K and 1397 Wm^{-2}K^{-1} measured using the gauges with an underside heater.

The improvement in the stability of the regression can again be quantified with the stability criterion introduced above for the double-sided thin film gauges with an underside heater. For the unheated results the stability criterion S is 0.31, whereas for the heated results it is 0.96. A combination of unheated and heated runs yields the highest value ($S = 0.97$).

UNCERTAINTY ANALYSIS

To estimate the uncertainty of the heat transfer coefficient, it is instructive to consider two discrete measurements of the wall temperature, T_1 and T_2, and the corresponding heat fluxes $\dot{q}_1$ and $\dot{q}_2$. The heat transfer coefficient is then given by

$$h = \frac{\dot{q}_2 - \dot{q}_1}{T_2 - T_1}.$$

The achievable temperature differences during a single run are rather small and statistical uncertainties are typically very large. It is therefore desirable to combine heated and unheated runs. While this approach leads to larger systematic uncertainties due to the need to compare runs with different initial wall temperatures, it significantly increases the range of T_w and hence improves the reliability of the linear regression. The resulting uncertainties in h can be calculated for a specific example, which is presented in Table 1. For this example he measurements of T_1 and $\dot{q}_1$ correspond to the unheated case, whereas T_2 and $\dot{q}_2$ result from one of the heated cases. The adiabatic wall temperature is determined from a linear extrapolation of the heat flux results for a set of different values of T_w. The given temperatures and heat fluxes are representative for the experimental data. With the current thermal models for the self-heating gauges, the uncertainties affecting the determination of the initial gauge temperature are high, and the gauges with an underside heater achieve more accurate results.

Quantity	Unheated	Underside heater	Self-heating
T	(360 ± 1.8) K	(420 ± 4.5) K	(420 ± 15) K
T_i	(290 ± 1) K	(370 ± 3.7) K	(370 ± 15) K
ΔT	(70 ± 1.5) K	(50 ± 2.6) K	(50 ± 2.6) K
$\dot{q}$	$(6 \pm 0.3) \cdot 10^4$ W m^{-2}	$(1.5 \pm 0.1) \cdot 10^4$ W m^{-2}	$(1.5 \pm 0.1) \cdot 10^4$ W m^{-2}
h	–	(750 ± 80) W m^{-2} K^{-1}	(750 ± 196) W m^{-2} K^{-1}
T_{aw}	–	(440 ± 7) K	(440 ± 20) K

Table 1: A specific example to illustrate the effect of systematical uncertainties on the linear regression.

CONCLUSIONS

Improving the cooling systems in HP turbine stages presents a key challenge for increasing the thermal efficiency of gas turbines. For this purpose it is crucial to understand the interaction of cooling flows with engine-realistic inlet flows (with temperature distortion and high swirl). Transient turbine facilities with engine-size turbine stages offer a unique oppor-tunity for performing this kind of measurements. Nevertheless, making the best use of these facilities and accurately studying such a complex environ-ment requires the development of improved instru-mentation. The work presented in this paper contri-butes to advancing heat transfer measurements for cooled turbine stages.

Novel gauge configurations based on double-sided thin film gauges have been designed to measure the heat transfer at low temperature differences on non-isothermal surfaces. The heating of the gauge substrate allows the adiabatic wall temperature and the heat transfer coefficient to be accurately determined. A stability criterion has been introduced to assess the improvement in the regression.

The combination of double-sided gauges with local surface heating arrangements is unique in that it allows measurements to be taken on intricate geometries without the need for complex heating systems. This technology has since been used to perform heat transfer measurements on fully film-cooled HP NGV aerofoils.

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APPENDICES

1. Impulse response processing of transient heat transfer signals

Historically, the experimental heat transfer rate was obtained from the surface temperature signal using numerical approximations to the solutions of the linear differential equations relating the two. The impulse response method, in contrast, only uses known pairs of exact solutions, such as the temperature response to a step in heat flux, to derive a sampled approximation of the impulse response of the gauge system [6]. This impulse response is then used as a finite impulse response digital filter to convert the surface temperature signal into a heat transfer rate. It is enough to calculate the impulse response once for each set of gauge parameters, making this processing of surface temperature signals computationally very efficient. This method and the corresponding filter routines have been extended to two-layer substrate gauges and double-sided gauges [6].

2. Use of HTA3 amplifier and modification for higher currents

The HTA3 thin film signal conditioning amplifier is used as a current source for the thin film gauges and to record the heat transfer signal. In order to achieve the higher currents necessary for using the HTA3 amplifier with self-heating gauges, the original maximum current of 20 mA had to be increased. This change required a modification of the high constant current supply to allow for thin film gauge currents up to 150 mA [11].

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