Research Article

Heat transfer and film cooling measurements on aerodynamic geometries relevant for turbomachinery

Patrick Jagerhofer¹ · Jakob Woisetschläger¹ · Gerhard Erlacher¹ · Emil Göttlich¹

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Abstract

A measurement technique for recording convective heat transfer coefficient and adiabatic film cooling effectiveness in demanding environments with highly curved surfaces and limited optical access, such as turbomachinery, is presented. Thermography and tailor-made flexible heating foils are used in conjunction with a novel multistep calibration and data reduction method. This method compensates for sensor drift, angle dependence of surface emissivity and window transmissivity, heat flux inhomogeneity, and conductive losses. The 2D infrared images are mapped onto the 3D curved surfaces and overlapped, creating surface maps of heat transfer coefficient and film cooling effectiveness covering areas significantly larger than the window size. The measurement technique's capability is demonstrated in a sector-cascade test rig of a turbine center frame (TCF), an inherent component of modern two-spool turbofan engines. The horseshoe vortices were found to play a major role for the thermal integrity of turbine center frames, as they lead to a local increase in heat transfer, and at the same instance, to a reduction of film cooling effectiveness. It was also found that the horseshoe vortices lift off from the curved surface at 50% hub length, resulting in a pair of counter-rotating vortices. The measurement technique was validated by comparing the data against flat plate correlations and also by the linear relation between temperature difference and heat flux. This study is complemented with an extensive error and uncertainty analysis.

Article highlights

- This paper presents an accurate measurement technique for heat transfer and film cooling
- on 3D curved surfaces with limited optical access
- using flexible tailor-made heating foils, infrared thermography and a high-fidelity multistep calibration process.

✉ Patrick Jagerhofer, patrick.jagerhofer@tugraz.at; ttm@tugraz.at | ¹Institute of Thermal Turbomachinery and Machine Dynamics, Graz University of Technology, Graz, Austria.

Graphical abstract

Keywords Heat transfer · Film cooling · Turbomachinery · Infrared thermography · Heating foil · Calibration

List of symbols

C	Chord length (m)
D	Distance, diameter (m)
DR	Density ratio ($= \rho_p / \rho_M$)
h	Convective heat transfer coefficient ($\text{W}/\text{m}^2\text{K}$), coordinate along TCF height (m)
H	Channel height (m)
I	Current (A) momentum flux ratio ($= \rho_p V_p^2 / \rho_M V_M^2$)
M	Blowing ratio ($= \rho_p V_p / \rho_M V_M$)
Nu_s	Local Nusselt number ($= h_s s / \lambda$)
Pr	Prandtl number (–)
$\dot{q}$	Heat flux (W/m^2)
Re_C	Reynolds number based on chord length ($= V_\infty C / \nu$)
Re_s	Reynolds number based on local chord coordinate ($= V_\infty s / \nu$)
s	Streamwise coordinate (m)
t	Thickness (m)
T	Temperature (K)
T_s	Diabatic surface temperature (K)
$T_{a,s}$	Adiabatic surface temperature (K)
Tu	Turbulence intensity (%)
U	Voltage (V)
U_F	Uncertainty in parameter F
V	Velocity (m/s)
w	Weighting function for blending (–)
α	Surface angle (degrees), temperature coefficient of resistance ($1/\text{K}$)
ε_s	Emissivity of surface (–)
η	Adiabatic film cooling effectiveness (–)

λ	Thermal conductivity ($\text{W}/\text{m}\cdot\text{K}$)
ν	Kinematic viscosity (m^2/s)
ρ	Density (kg/m^3)
τ_W	Transmissivity of window (–)

Subscripts

NP	No purge
P	Purge or paint
S	Surface
W	Window
∞	Freestream

1 Introduction

Driven by environmental protection legislation, the technological development in the field of modern turbomachinery must address process and efficiency optimization [1]. However, possible improvements are often hindered since machinery is designed with excessive safety margins strictly required because sufficiently accurate flow numbers are not at the engineer's disposal [2]. In thermal turbomachinery, the convective heat transfer coefficient and the film cooling effectiveness are such numbers, subject to greater uncertainty. Due to the coupling of heat transfer with secondary flow structures and the presence of cooling and purge flows, the situation quickly becomes complex [3]. With this work, the authors contribute to the development of experimental measurement techniques for heat transfer coefficients and film cooling effectiveness in the challenging environment of thermal turbomachinery.

1.1 Geometry investigated

This work discusses a steady measurement technique for heat transfer and film cooling demonstrated in a so-called turbine center frame (TCF), an inherent component of modern turbofan engines. The TCF is an S-shaped duct connecting the high-pressure (HPT) to the low-pressure turbine (LPT) in a turbofan aero-engine and is designed as a diffuser characterized by large concave and convex bends on the end-walls. Accordingly, the aerodynamics in these annular ducts is complex with a variety of secondary flows present [4]. In modern aero-engines, the turbine inlet temperature is increased to further improve the overall joule-efficiency of the engine, a requirement causing higher thermal loads in the TCF. To master these thermal loads, purge and cooling flows are needed, which lead to

complex aerodynamics and non-uniform distributions of temperature and heat transfer.

1.2 State of the art

When heat transfer coefficients are required, the surface temperature and the heat flux must be measured. Thermocouples or resistance thermometers positioned at specific positions along the surface are commonly used for temperature measurements [5, 6]. However, the ready accessibility of easy-to-use *thermography* caused a paradigm shift for surface temperature measurements. Cameras sensitive to infrared (IR) radiation for *thermography* can record areal surface temperatures non-intrusively with high sensitivity and low response time [7]. Therefore, the majority of more recent experiments use such *IR cameras* for temperature measurements [8–14]. In contrast to thermocouples, however, IR cameras have to undergo a more extensive calibration to deliver accurate temperature readings. Martiny et al. [15] present an in-situ calibration for infrared thermography with thermocouples embedded in the surface of interest; a setup later improved for highly curved surfaces by Aberle et al. [16]. Considering non-uniform background radiation reflected by the surface, Elfner et al. [17] developed a spatially resolved calibration based on ray-tracing algorithms.

For heat flux measurements different steady and transient techniques exist. Steady heat flux measurements either calculate the heat flux through a material of known thermal conductivity by measuring the temperatures on both sides [6, 11, 12], or record the local heat flux with the help of heat flux sensors, e.g., thin-film heaters [9, 10, 13]. Carlomagno [18] reviewed different types of heat flux sensors and concluded that thin-film heaters are precise and effective devices. Carlomagno [18] stated that an inherent problem with all foil-type heat flux sensors is tangential or lateral conduction. A correction for this type of conduction is discussed by Astarita et al. [10], using a generalized form of Fourier law adapted for anisotropic conductivity in a thermally thin plate. For transient techniques, the rapid heating or cooling of a specimen is recorded, and the heat transfer coefficient is calculated according to analytical or numerical models of the specimen. Metzger et al. [5] recorded the transient heat up of an aluminum block, and von Hoesslin et al. [14, 19, 20] recorded the temperature decline of a low conductivity coating after exposure to a high-energy laser pulse.

Only one experimental study about heat transfer in TCFs exists to the best of the authors' knowledge. Arroyo Osso et al. [11] investigated an aerodynamically "aggressive" TCF with non-turning structural struts. In this study,

the endwalls and the struts were heated by internal water flows to measure the heat flux. The endwalls were made out of a thin polycarbonate sheet with a water channel on the backside. The struts were made of aluminum with bores for the hot water and a thin cover layer of epoxy resin. The heat flux was calculated for the endwalls with a one-dimensional heat transfer model and for the struts with a finite element analysis (FEA) simulation. Multiple spring-loaded hatches enabled optical access to the opposite side of the channel to cover the whole TCF surface while ensuring favourable viewing angles [11].

1.3 Concept of current research

Different to Arroyo Osso et al. [11], a single pair of IR windows and heating foils or thin-film heaters on top of an adiabatic surface are used for this work to enable the heat transfer measurements in the TCF. The tailor-made and flexible heating foils can be slightly stretched in order to fit the three-dimensional curvature of the investigated surfaces and are designed for constant heat flux. Since the foils were mounted on an adiabatic carrier material, no FEA simulation considering heat conduction through the material is needed. On the other side, temperature-dependent electrical resistance variations within the heating foil and non-uniform heat release are corrected.

To cover the whole region of interest, the IR camera had to be positioned at various angles to the surfaces investigated and the single pair of IR transparent windows used. To ensure high precision and accuracy, a novel four-step temperature calibration of the recorded IR images is presented in this work, including:

- the calibration against an isothermal surface in a vacuum chamber,
- the correction of emissivity as a function of observation angle,
- the correction of window transmissivity at sharp viewing angles and
- an in-situ calibration with embedded thermocouples.

Since heat transfer and cooling are closely related, the film cooling effectiveness of the purge flow ejected from the cavity of the upstream turbine rotor was investigated. Firstly, the heat transfer coefficient and the film cooling effectiveness of an *undisturbed inflow* condition are presented and then compared to a setup where the TCF inflow is disturbed by *inlet pegs*. This study is completed by an error analysis covering all of the aforementioned challenges in using infrared thermography and heating foils in engine-relevant geometries.

Fig. 1 Exploded view of the sector-cascade test rig (adapted from [21])

2 Materials and methods

This section briefly introduces the sector-cascade test rig and its instrumentation. A more thorough explanation of the design and commissioning of the test rig and the aerodynamic measurement equipment can be found in Jagerhofer et al. [21]. The main focus of this paper lies on the data reduction scheme starting with raw measurement data, such as uncalibrated IR images and electrical power to the heating foils, until the final surface maps of convective heat transfer coefficient h and film cooling effectiveness η .

2.1 Test rig and operating conditions

Figure 1 shows an exploded view of the sector-cascade test rig. The incoming main flow of approx. 0.54 kg/s was delivered by a variable speed centrifugal compressor. After the inlet duct, the main flow entered the test section of one full TCF passage situated between two quarter passages. The hub (inner endwall) and struts of the TCF were machined out of Rohacell IG-F 71, a quasi-adiabatic material with a very low thermal conductivity of approx. $\lambda_{\text{Rohacell}} = 0.03 \text{ W/m}\cdot\text{K}$. On this insulating surface, the heating foils were applied and spray-painted with high emissivity paint (Nextel Velvet 811–21) with an emissivity of 0.967 and a thermal conductivity of $0.197 \text{ W/m}\cdot\text{K}$ [22]. After spray-painting, the surface was sanded to obtain a hydraulically smooth surface finish [23]. The heating foils were powered by standard laboratory power supplies. A row

of cylinders or pegs was installed at the outlet of the TCF to simulate the blockage effect of the downstream low-pressure turbine vanes. The purge flow emanating from the aft hub cavity of the upstream HPT bears a significant cooling potential for the TCF [21]. For this reason, the aft hub and shroud cavities of the HPT were realized as purge plenums with engine-relevant seal geometries. The optical access for the IR camera was implemented by fitting two barium fluoride windows on the shroud surface of the TCF.

Table 1 summarizes the operating conditions of the two test cases, a case with *undisturbed inflow* conditions and a case where cylinders or *inlet pegs* were installed at the TCF inlet. For both cases, the free-stream Mach number at the TCF inlet equaled 0.14 and the Reynolds number based on the strut chord length was 4.25×10^5 . The blowing ratio of the hub purge flow was set to 0.21, a relatively high value for aero-turbines. The shroud purge flow was not investigated in this paper and was switched off for the experiments presented in this work. The *undisturbed inflow* case corresponds to the "chilled high purge" case in Jagerhofer et al. [23] and is explained more in detail there. The purpose of the *inlet pegs*, installed in the second test case, was to create disturbed inlet flow conditions with wakes and increased turbulence; a highly simplified abstraction of an upstream turbine stage. As illustrated in Fig. 1, four inlet pegs with a pitch of 7.5° were positioned in front of one TCF passage and were aligned in a way that no wake of the pegs impinged on the struts' leading edges. The pegs diameter, d , equals 35% of the struts' maximum thickness with the pegs axially situated $6.5d$ upstream of the struts' leading edges.

2.2 Instrumentation

Figure 2 shows a cross-section of the sector-cascade rig with its instrumentation. A total temperature probe, a pitot-static tube, and an orifice plate (not seen in Fig. 2) were positioned far upstream in the main supply pipe

Table 1 Operating conditions

Parameter	Test Case	
	Undisturbed Inflow	Inlet Pegs
Free-stream Mach number at inlet, Ma	0.14	
Reynolds number, Re_c	4.25×10^5	
Turbulence intensity, Tu	3.5%	–
Hub purge blowing ratio, M	0.21	
Hub purge density ratio, DR	1.103	1.096
Hub purge momentum flux ratio, I	0.0414	0.0417
Inlet pegs pitch	–	7.5°

Fig. 2 Sector-cascade rig instrumentation (adapted from [21])

to set the operating point. To characterize the inflow, a thermocouple-equipped five-hole probe was radially traversed at the TCF inlet. The five-hole-probe measurement-results of the *undisturbed inflow* condition (without inlet pegs) can be found in Jagerhofer et al. [21]. The hub purge flow temperature was measured using a single calibrated thermocouple in the axial clearance of the rim seal. The circumferential purge flow uniformity was monitored with nine equally spaced pressure taps in the same location.

The hub and strut surfaces were covered with six tailor-made heating foils designed with and produced by the Austrian industrial collaborator ATT GmbH. The heating foils of this manufacturer consist of several different layers. Starting from bottom, the first layer is heat resistant and flexible glue with a thickness of approx. 200 μm , thick enough to compensate for the different thermal expansions of the heating foil and the Rohacell substrate. The next layer is a 50 μm thick Kapton insulation layer, followed by the 35 μm thick active layer of etched copper conductor tracks. The width and the spacing of the meandering copper tracks dictate the local electric heat production and are designed in an iterative process to deliver constant heat flux. After another insulating 50 μm layer of Kapton, a full sheet of 35 μm copper is added to laterally distribute the heat between the hot copper tracks and the cold interstitial gaps between them. This distribution layer is just thick enough so that the single copper tracks are not visible in the IR image but thin enough to prevent significant lateral conduction on a macroscopic scale. The last layer on top is again a Kapton layer of 50 μm . The overall thickness of the heating foil without glue equals 220 μm . A relatively thick (100 μm) layer of the high emissivity paint and the uppermost Kapton layer acted as thermal insulation that

reduces the impact of the underneath copper layer on the lateral conduction along the flow-wetted surface.

The surfaces of interest were observed using a FLIR T650-sc IR camera with an uncooled microbolometer and a relatively high thermal sensitivity of 20 mK. To further improve the IR measurement accuracy, nine single-calibrated thermocouples braced on 0.3 mm thick and 6 mm diameter copper discs were placed along the surfaces of interest for an in-situ calibration. These in-situ thermocouples were located on the gaps between the heating foils and were painted together with the heating foils after installation.

At the end of the measurement campaign, an oil dot flow visualization was performed to visualize the trajectories of the wall shear stress. Instead of regular industrial oil, a mixture of glycerol, talcum powder and food coloring was used in order not to harm the high emissivity paint. Talcum powder was used to adjust the desired viscosity of the mixture. Equally sized dots of the mixture were applied on the hub and struts of the TCF. Then the test rig was operated at the nominal operating point until a desired running length of the oil dots was achieved. The running direction of the oil dots then visualized the local wall shear stress trajectories.

Light sheet flow visualization based on a fundamental particle image velocimetry setup [24] was used as an additional qualitative visualization technique of the flow field. The main flow was seeded with small oil droplets coming from a seeding generator far upstream in the supply pipe of the test facility. The light sheet was produced by a 100 mW diode laser with a cylindrical lens and guided through the BaF₂ window into the TCF. Images of the light sheet were taken with a standard DSLR camera (Canon EOS 250D).

3 Data reduction

Figure 3 shows the main data reduction scheme as a data flowchart based on ISO 5807. Blue parallelograms denote input data acquired during the measurement, white parallelograms denote interim data, and white rectangles denote data processing. The whole scheme in Fig. 3 can be divided into the postprocessing of the temperatures measured with the IR camera, the postprocessing of the heat flux produced by the heating foils, and the final combination of these data into the convective heat transfer coefficient h and the film cooling effectiveness η . The following section thoroughly illustrates how data is processed during each step.

Fig. 3 Main scheme of data reduction, calibration and correction

3.1 Definition of heat transfer coefficient and film cooling effectiveness

To measure the heat transfer coefficient h of a given operating point, two sets of IR images and the electric power used to heat the foils had to be acquired at constant operating conditions. The first set of images was acquired with heated foils and the second without heating. The condition without heating is considered quasi-adiabatic due to the very low conductivity of the substrate material. The heat transfer coefficient h is thus defined by:

$$h = \frac{\dot{q}}{T_s - T_{a,s}} \quad (1)$$

where $\dot{q}$ is the final heat flux field, T_s the final temperature field with heating and $T_{a,s}$ the quasi-adiabatic final temperature field of the investigated surfaces.

To measure the film cooling effectiveness η , the heating was off and two sets of IR images were acquired, one with the hub purge flow switched off and one with the purge flow set to the desired blowing ratio M . The film cooling effectiveness η is defined by:

$$\eta = \frac{T_{a,sNP} - T_{a,sP}}{T_{a,sNP} - T_p} \quad (2)$$

where $T_{a,sNP}$ is the quasi-adiabatic final temperature field without purge, $T_{a,sP}$ the quasi-adiabatic final temperature field with purge and T_p the temperature of the purge flow.

3.2 Temperature data reduction

Starting with a single raw IR image, the first step is to map the 2D image onto the 3D surface of the TCF using projective geometry [25], as shown in Fig. 3. This direct linear transformation (DLT) is used to project the pixels of the image onto a block-structured surface mesh of the TCF. Please note that the surface mesh resolution must be at least as fine as the finest spatial resolution of all IR images (pixels/mm) to avoid downsampling and mapping errors. Point correspondences between the 2D image plane and the 3D surface mesh of the TCF are necessary for transformation. These correspondences were realized by painting 114 reference points with 1.5 mm diameter on the TCF surface using a high reflectivity paint. This made the reference points visible in the raw IR image, marked with circles in Fig. 4a. The 3D coordinates of the points were measured with a laser scanning measurement arm (Quantum FaroArm). The points were distributed so that at least ten

Fig. 4 Temperature postprocessing: from raw IR image to one calibrated temperature 3D patch

reference points are visible in every image. Please note that the DLT algorithm only needs at least $5^{1/2}$ reference points, but over-determination leads to a more stable and accurate mapping. After rejecting undesired pixel areas (i.e., window frame or unheated areas), the 2D-3D mapping results in the *raw IR 3D patch* shown in Fig. 4c.

An inherent problem of a microbolometer sensor is its long-term drift and the relative drift from pixel to pixel. Therefore, a calibration procedure tailored to the experimental setup was conducted before every test campaign, where the IR camera was calibrated against an isothermally heated copper block placed in a vacuum chamber. The copper block was painted with the same high emissivity paint as the TCF surface, and the optical access of the vacuum chamber was realized with the same IR window as used in the test rig. The 180 mm × 150 mm × 20 mm copper block used for calibration was heated on its backside with a heating foil, and its back and side faces were insulated with Rohacell. The block's temperature was measured with four single calibrated thermocouples immersed into the sidewalls of the block. There was no influence of natural convection due to the vacuum in the chamber. The temperature drop over the paint thickness was corrected by a 1D heat flux balance, where the heat flux crossing the layer of paint was assumed to be equal to the radiative heat flux from the paint surface to the inner walls of the vacuum chamber. To convert the raw counts of the IR footage into temperatures, the Atlas software development kit (SDK, FLIR) was used, with the paint's emissivity, the window's transmissivity and the temperatures driving the background radiation set. These driving temperatures are the window temperature and the temperature of the vacuum chamber, which were measured by single calibrated thermocouples. The *copper block calibration* results in a calibration curve for every pixel of the IR camera, needed to convert the *raw IR 3D patch* into the *temperature field* along the surface. Figure 4d shows the impact of the copper block calibration as temperature difference.

Figure 2 illustrates that the camera has to be used at shallow viewing angles, α_S , on the observed surface and, α_W , on the window, to cover all surfaces of interest. These

Fig. 5 Paint emissivity and window transmissivity as a function of observation angle

variations in angle lead to variations in IR emission from the surface, here discussed in terms of surface emissivity variations of the paint ϵ_S , and transmissivity variations of the window τ_W , always compared to the perpendicular viewing angle of the copper block calibration. To account for the emissivity drop, an isothermally heated copper cylinder, again painted with the same high emissivity paint, was placed in the vacuum chamber. The emissivity was recorded as a function of the surface observation angle α_S , using the freshly calibrated IR camera. The resulting emissivity curve is shown in Fig. 5, and the values were found to be similar to the measurements of Lohrengel et al. [22]. The increasing deviation from the Fresnel correlation for viewing angles $< 40^\circ$ can be explained with the paint's solid pigments causing a dull and rough surface. Please note that the paint of the copper block and cylinder was also sanded to obtain the same surface finish as in the test rig. The transmissivity drop of the 10 mm thick BaF₂ IR window was recorded by exposing the IR camera to the

Fig. 6 Temperature postprocessing: from the set of calibrated temperature 3D patches to the final temperature field

isothermal copper block and by step-wise inclining the IR window in the optical path. The transmissivity curve in Fig. 5 shows a significant drop for shallow viewing angles below 40° , underlining the importance of this calibration step.

The DLT algorithm can also estimate the *camera position*. This eases image acquisition procedure since no precise traversing mechanism is needed, and the camera could also be used handheld and moved freely in space while acquiring the images. The estimate of the camera position is then used to calculate the viewing vector (Fig. 4b) and subsequently the surface observation angle α_s , and the window angle α_w , for each pixel. With the curves of Fig. 5, the corresponding surface emissivity and window transmissivity are found and must be updated in Atlas SDK, delivering the viewing angle corrected temperatures for every pixel. Note that the window, ambient and reflected temperature had to be acquired for every test run and set for the correct conversion of digital intensity counts to temperature. The result is the final *calibrated temperature 3D patch* shown in Fig. 4g. The impact of the surface angle and window angle calibration is shown as temperature difference in Fig. 4e and f. This procedure has to be repeated for every image of the set.

After having mapped and calibrated one complete set of IR images, the patches had to be combined to produce a temperature field covering all sections of interest, the TCF's hub and struts. Due to spatially varying background radiation and other imperfections in the measurement

technique, the *calibrated temperature 3D patches* may show a temperature offset in overlapping areas. This is most pronounced at very shallow observation angles, mainly when the outer window surface reflects the radiation from the hot window frame. In the first step, the offset is removed by subtracting it from the affected patch. The starting point for this procedure always is a patch observed at a nearly perpendicular observation angle with respect to surface and window. This first patch reflects the fully calibrated conditions, and the other patches are corrected for varying background radiation at shallow angles.

In a second step, the following blending function is used to merge the temperature patches while providing a smooth transition in overlapping regions:

$$T = \frac{\sum_i^n w_i T_i}{\sum_i^n w_i}; w_i = \frac{d_{\max,i} - d_i}{d_{\max,i}} \quad (3)$$

where w_i is the weight of the temperature T_i of the i th patch. The weight w_i is based on the distance d_i of the affected point from the center of the patch. The weight of the i th patch equals zero at the corner farthest from the center ($d_{\max,i}$) and increases towards the center of the patch. By using the offset correction and the blending function in Eq. 3, the set of *calibrated temperature 3D patches* shown in Fig. 6a are overlapped and result in the *3D temperature field* shown in Fig. 6b.

In the next step, the *in-situ calibration* of the *3D temperature field* has to be performed. The copper discs of the

nine in-situ thermocouples are marked with circles in the temperature field in Fig. 6b. A single calibrated thermocouple was brazed to the bottom side of each disc. The size of the copper discs ensured that a sufficient number of the camera's pixels read the same temperature as the thermocouple. In the case of heating the foils, an unknown heat flux passed from the borders of the heating foils to the copper discs and subsequently through the paint into the main flow. This heat flux led to a temperature drop over the paint thickness $t_p \approx 100 \mu\text{m}$, which needed to be compensated for the in-situ calibration. By assuming one-dimensional conduction through the layer of paint, this temperature drop is calculated by:

$$\frac{\lambda_p}{t_p} (T_{TC,heated} - T_{corr.}) = h(T_{corr.} - T_{TC,unheated}) \quad (4)$$

Here, $T_{TC,...}$ is the thermocouple reading of the heated and unheated condition, $T_{corr.}$ is the temperature on top of the painted copper disc when heated, and λ_p the thermal conductivity of the paint taken from Lohrengel et al. [22]. Note that for unheated conditions, no temperature drop exists over the paint thickness, and the temperatures above and below the paint are identical. The left-hand side of Eq. 4 is the heat flux through the paint, which is equal to the right-hand side, representing the convective heat flux from the paint's surface into the main flow. Since the convective heat transfer coefficient, h , is needed, Eq. 4 has to be solved for $T_{corr.}$ iteratively.

In the next step, the temperature difference between the (paint thickness corrected) thermocouple reading and the 3D temperature field in Fig. 6b is calculated for the nine discrete positions of the thermocouples. To obtain a field of temperature offsets from the nine discrete thermocouples, the natural neighbor interpolation [26] is used. Outside the area spanned by the nine thermocouples marked as crosses in Fig. 6c, "ghost positions" marked with G are introduced. The ghost positions have the same ΔT value as the closest thermocouple and have the purpose to extend the interpolation area over the whole TCF surface. The result of the nearest neighbor interpolation is the temperature difference field in Fig. 6c, representing the in-situ calibration offset which is added to the 3D temperature field in Fig. 6b.

The reference points and the gap between the heating foils at the centerline are removed from the in-situ calibrated 3D temperature field in Fig. 6d using an inverse distance interpolation. The gaps between the struts and the hub are still visible in the results because the gaps are too wide for a reliable estimation of the temperature by the neighboring heated areas. In a final step, the temperature field is smoothed by shifting the temperature value at a specific point towards the average of its neighboring data

points. After applying all these steps, the final temperature field in Fig. 6e results and the temperature postprocessing is finished.

3.3 Heat flux data reduction

Since heat production in the heating foils is based on the resistance of the copper conductor tracks, the local heat release is linked to the local temperature through the copper's temperature coefficient of resistance α . The average heat flux $\dot{q}_{el\ avg}$ produced by the heating foil is the supplied voltage times current divided by the heated area of the heating foil. $\dot{q}_{el\ avg}$ equals the local supplied electric power $\dot{q}_{el}$ only when the temperature on the heating foil's surface T_s equals the average foil temperature, $T_{s,avg\ HF}$. Otherwise, the heat flux must be corrected by the following equation [21]:

$$\dot{q}_{el} = \dot{q}_{el\ avg} (1 + \alpha(T_s - T_{s,avg\ HF})) \quad (5)$$

Figure 7a shows the heat flux after correction with Eq. 5 for the temperature dependence of the copper track's resistance. The temperature field used for the correction is the final temperature field seen in Fig. 6e, with this data flow indicated by an arrow in Fig. 3.

Albeit the heating foils were designed to deliver constant heat flux, a residual inhomogeneity of $\pm 10\%$ per heating foil is possible. Therefore, a step-wise transient correction similar to Lazzi Gazzini et al. [13] is implemented, where the local heat release of the heating foils is assumed to be proportional to the local temperature increase in a transient experiment without flow. Starting at ambient temperature and without flow, the heating foils were switched on with the temperature increase recorded. By choosing two frames where the impact from free convection and lateral conduction is still negligible, a ΔT map covering all heating foils on the TCF is created. The relative heat flux inhomogeneity per heating foil $\dot{q}_{inhom.rel.}$ is then calculated with

$$\dot{q}_{inhom.rel.} = \frac{\Delta T}{\Delta T_{avg\ HF}} - 1 \quad (6)$$

where $\Delta T_{avg\ HF}$ is the average temperature increase per heating foil. This simple approach is possible because the thermal properties of the heating foil and the thermal effusivity of the substrate are constant, and the time step between the frames is the same for every position along the surface. The relative heat flux inhomogeneity map is shown in Fig. 7b.

The hub and the struts are machined out of Rohacell foam. Although nearly adiabatic, the conductive heat loss through the relatively thin struts cannot be

Fig. 7 Heat flux postprocessing

completely neglected. The struts were only heated on the sides facing the flow passage investigated. Figure 8 shows a schematic of the strut and the underlying one-dimensional heat resistance network. Since the strut is symmetric and the inflow is without swirl, the same heat transfer coefficient h existed on both sides of the strut.

Fig. 8 Simplified heat resistance network of the struts

The heating foil had the temperature T_{HF} , with the thicknesses of glue and heating foil's layers being neglected. Due to the very low conductivity of Rohacell and the excellent bond between the glue and the Rohacell substrate, their contact resistance was neglected. These assumptions lead to an error of less than one percent in the following calculations. In Fig. 8, the Rohacell and the high emissivity paint acted as competing thermal resistances. Their ratio drove the splitting of the supplied power $\dot{q}_{el}$ into the desired heating of the investigated surface $\dot{q}$, and the undesired lost heat flux into the not investigated quarter passage $\dot{q}_{loss}$.

Since the driving temperature difference $T_{HF} - T_\infty$ was the same for the desired and the lost heat flux, the relation of $\dot{q}$ and $\dot{q}_{loss}$ is a function of the heat resistances:

$$\frac{\dot{q}_{loss}}{\dot{q}} = \frac{1/h + t_p/\lambda_p}{1/h + t_{Rohacell}/\lambda_{Rohacell}} = \dot{q}_{loss,rel.} \quad (7)$$

with $t_{Rohacell}$ the local thickness of the Rohacell substrate. The relative conduction loss is shown in Fig. 7c. Since this 1D correction is used on the struts only, a radial blending function is defined with the fillet radius between hub and struts.

The relative heat flux inhomogeneity in Fig. 7b is used together with the relative conduction loss in Fig. 7c to correct the heat flux, $\dot{q}_{el}$:

$$\dot{q} = \dot{q}_{el} \frac{1 + \dot{q}_{inhom.rel.}}{1 + \dot{q}_{loss,rel.}} \quad (8)$$

Firstly, the heat transfer coefficient h is calculated without using the one-dimensional conduction loss correction in Eq. 7 and secondly, h is iteratively corrected by computing values for $\dot{q}$ using Eqs. 7 and 8. Convergence was achieved after less than ten repetitions. The final $\dot{q}$ field with all aforementioned corrections is shown in Fig. 7d.

3.4 Computational cost

The computational cost for the entire data reduction scheme shown in Fig. 3 is approximately 4 h using 24 Intel Xeon

E5-2680 v3 cores for a surface mesh with approximately 500,000 cells. The most time-consuming step is the calculation of the surface and window angles for each pixel of the raw IR images, which takes 94% of the computation time. The rest of the data reduction scheme is processed using a single core.

4 Results and discussion

This section presents the final results after conducting all of the aforementioned data reduction schemes, calibrations and corrections, summarized in Fig. 3. Figure 9 shows a qualitative oil dot flow visualization of the *undisturbed inflow* condition and surface maps of h and η on the hub for the *undisturbed inflow* and the *inlet pegs* condition. A hub purge flow with a blowing ratio of $M=0.21$ and a density ratio of $DR \approx 1.1$ was injected from the hub cavity exit in both conditions.

The mapped photography of the oil dot visualization is shown in Fig. 9a overlaid with wall shear stress directions in red and a qualitative illustration of one leg of the horseshoe vortex (HSV) in blue. As indicated by the curvature of the wall shear stress trajectories, the horseshoe vortices strongly direct the flow from the hub towards the center of the channel in the vicinity of the strut leading edge. After 50% hub length, the influence of the horseshoe vortices

on the wall shear stress orientation diminishes as the vortices lift off from the hub surface and migrate towards midspan. This lift-off was confirmed with the light sheet flow visualization shown in Fig. 10. Figure 10a shows the position of the light sheet, and Fig. 10b shows the photograph of the light sheet fitted on the 3D model of the TCF. The core of one leg of the now expanded horseshoe vortex can be identified as the dark spot circled red in Fig. 10b because the seeding oil droplets are driven out from the core of the vortex due to centrifugal forces. Although not shown here, the same is true for the other leg of the horseshoe vortex since the flow in the TCF is fully symmetrical and momentum must be conserved. After the lift-off, the horseshoe vortices transition into a vortex pair, obviously driven by the convex curvature of the hub close to the TCF exit [27]. On the strut in Fig. 9a, an s-shaped deviation of the shear stress orientation from the aerodynamic contour was found. As described by Steiner [28], this alternating (first up, then down) radial flow migration is caused by the radial pressure gradient in the passage, which is initially oriented from the hub to the shroud and reverses its orientation after about 50% hub length. Compared to Steiner [28], this radial flow migration starts later in the measurement discussed here due to a lower inlet Mach number.

In Fig. 9b and c, the heat transfer coefficient h is normalized with the maximum value of the non-purged *undisturbed inflow* condition $h_{NP\ max}$. Both conditions, the

Fig. 9 Oil dot flow visualization, heat transfer coefficient and film cooling effectiveness of the investigated test cases (Horseshoe vortex HSV)

Fig. 10 Light sheet flow visualization of the horseshoe vortex lift-off

undisturbed inflow and the *inlet pegs*, have the highest heat transfer coefficients on the hub at the cavity exit, followed by an asymptotical decrease in the streamwise direction typical for an unheated starting length setup. Around the leading edge at the hub, where the horseshoe vortices cause local deflections of the flow, the heat transfer is intensified. For both inflow conditions, h is increased by up to $\sim +10\%$ in the vicinity of the horseshoe vortices' onset. This intensification slowly fades out at 50% hub length, where the horseshoe vortices lift off from the surface.

Adding the inlet pegs leads to an $\approx +10\%$ increase of heat transfer on the first third of the hub compared to the *undisturbed inflow* condition. This increase reduces to $\approx +6\%$ at half of the hub length and remains until the TCF exit. The signature of the inlet pegs' wakes can be seen in the heat transfer distribution of the hub marked with the dotted line. Two spots of decreased heat transfer are found for both operating conditions at the end of the hub at the TCF outlet. It is speculated that this decrease of heat transfer is caused by the combination of multiple factors: 1. the growth of the boundary layer, driven by the convex curvature of the hub at this position, 2. the influence of the migrated arms of the horseshoe vortices and 3. the increased diffusion and deceleration of the flow after the trailing edges of the struts. However, it must also be stated that the hub surface downstream the heating foils is made from aluminum, and therefore a systematic error due to lateral heat loss cannot be ruled out.

At the bottom of Fig. 9, the η distributions are compared. The *undisturbed inflow* condition has superior film cooling performance with a maximum film cooling effectiveness of 0.47 close to the hub cavity exit and a cooling film coverage extending until half of the hub length. Please note that a cooling film coverage is herein defined when $\eta > 0.1$. The film cooling performance of the inlet pegs condition is inferior, with a maximum of 0.4 close to the hub cavity exit and a cooling film coverage to only 30% of the hub length. The magnitude and the longitudinal spread of the cooling film deteriorate in the presence of

the inlet pegs due to enhanced mixing of the main and the purge flow in the wake of the pegs. As shown in Jagerhofer et al. [21] for the *undisturbed inflow* condition, η is virtually zero on the fillet radii of the struts and the struts itself due to the horseshoe vortices, which dilute and sweep the purge flow away from the struts.

At the top of Fig. 11, the heat transfer coefficient distributions on the struts of the same conditions as in Fig. 9 are compared. As above, the inlet pegs produce a disturbed inflow that intensifies the heat transfer. This intensification is higher towards the leading edge with $\approx +15\%$ and

Fig. 11 Heat transfer coefficient on the struts, chordwise variation of h at strut midspan for all investigated conditions

decreases towards the trailing edge to zero. The overall heat transfer behavior of both conditions is again comparable since the same zones of high and low h can be identified on both struts.

At the bottom of Fig. 11, the chordwise variation of h along the strut's midspan is plotted for the two conditions above ($M=0.21$, solid lines) and their corresponding no purge conditions ($M=0$, dotted lines). Additionally, the laminar and turbulent flat plate correlations for constant heat flux are shown as dashed lines, and their formula is given in the diagram [29]. The first data points of both *undisturbed inflow* conditions $M=0.21$ and $M=0$, agree with the laminar correlation and the laminar-turbulent transition instantaneously onsets, as the black solid and dotted line start to diverge from the laminar correlation. The transition is indicated by the fading grey bar, and the center of the transition is indicated at the first inflection point of the curves with the vertical dotted line.

As shown in Jagerhofer [23], the heat transfer is enhanced for the purged *undisturbed inflow* condition ($M=0.21$) due to a flow acceleration of the main flow caused by a mild blockage effect of the injected purge flow. After the transition, especially the no purge ($M=0$) *undisturbed inflow* condition excellently agrees with the turbulent flat plate correlation, validating the measurement technique. As already noted above, the inlet pegs lead to an intensification of heat transfer. No indication for a laminar-turbulent transition exists in the observable area for the *inlet pegs* conditions. The boundary layer seems to immediately start turbulent due to the high freestream turbulence caused by the pegs. The red solid and dotted lines agree well with the turbulent correlation from the leading edge up to 8% chord length and then start to deviate from the correlation to even higher heat transfer coefficients. The difference in h between the no purge ($M=0$) and the purged ($M=0.21$) condition is less pronounced than for the *undisturbed inflow* condition. The influence of the inlet pegs seems to dominate the heat transfer behavior over the blockage effect of the purge flow.

For further validation of the measurement technique, the no purge *inlet pegs* condition was repeated with 50% and 130% of the nominal heating power. These conditions are shown as grey dotted lines in the diagram. According to the linearity of the energy equation [30, 31], the heat transfer coefficient h as it is defined in this study has to be independent of the heating power. In other words, different power settings must always give the same value of h . As the two grey dotted lines and the red dotted line collapse within $\pm 4\%$, the herein shown measurement results are in accordance with this linearity; a further validation of the measurement technique.

5 Uncertainty

The uncertainties presented in this section were calculated based on the *guide to the expression of uncertainty in measurement* [32]. The impact of the input variables' uncertainties (blue parallelograms in Fig. 3) on the final measurands h and η are computed using a sensitivity analysis. By offsetting one input variable at a time by its standard deviation (SD) and rerunning the data reduction in Fig. 3, the SD of the input variable is converted to the SD of the final measurands h and η . The root-sum-square (RSS) of these "converted" SDs gives the final SD of h and η . The presented full-width uncertainties in Figs. 12 and 13 are twice the SD.

For input parameters where no probabilistic uncertainty information but guaranteed maximum errors are given, a rectangular or triangular probability distribution between the maximum errors is assumed and its SD is calculated [32]. This is the case for the measured voltage and current, and for error sources where only the maximum possible error is known from a worst-case estimation.

Table 2 lists a breakdown of the most influential error and uncertainty sources of the temperature and heat flux measurements. The main contributors to the temperature's uncertainty are: 1. the 2D-3D mapping accuracy converted into a temperature uncertainty using the highest temperature gradient of the field, 2. The overall uncertainty of the copper block calibration, 3. the uncertainty of the window and surface angle calibration, 4. the uncertainty of the single calibrated in-situ thermocouple readings and 5. the uncertainty of the temperature drop correction for the paint thickness on the in-situ thermocouple's copper disc. The SD of the *final temperature field* equals 0.3 K for the heated and 0.17 K for the unheated case and is the RSS of all considered uncertainty sources. The uncertainties of the angle calibrations for the emissivity and transmissivity are low, because the in-situ calibration compensates for an absolute error in the ϵ or τ measurement shown in Fig. 5. Therefore, only errors in the inclination of the ϵ or τ curves impact the accuracy but not the absolute values. Note that for the sake of brevity, only the most influential uncertainty sources are mentioned here, and the remainders are summarized under "other" in Table 2.

The major contributors to the heat flux's uncertainty are not the uncertainties of measuring electric voltage, current and the heating foil area (altogether SD equals 0.4 to 1.4%), but the uncorrected heat loss through radiation and the uncorrected lateral conduction within the heating foil. The maximum error of the lateral conduction only occurred in areas of very high temperature gradients, such as the laminar-turbulent transition on the struts and at the

Fig. 12 Uncertainty of h along the hub centerline and midspan of the strut

Fig. 13 Uncertainty of η as a function of its value

Table 2 Breakdown of measurement uncertainty in the final temperature and heat flux fields

	Error Limit	Standard Deviation heated/unheated	Unit
Temperature			
2D-3D mapping	–	0.12/0.05	K
Copper block calibration	–	0.12/0.12	K
ε and τ angle calibration	–	0.15/0.08	K
In-situ: thermocouple	–	0.06/0.06	K
In-situ: T-drop paint	–	0.18/0.00	K
Other (T_{wp} , T_{amb} , T_{refl} , ...)	–	0.05/0.04	K
Final temperature field	–	0.30/0.17	K
Heat flux			
Voltage	2.67	1.09	mV
Current	42.5	17.4	mA
Heating foil area	–	25	mm ²
Radiation	2.3	2.3	% $\dot{q}$
Lateral conduction	10	4.1	% $\dot{q}$
Residual $\dot{q}$ -inhomogeneity	3	1.2	% $\dot{q}$
Final $\dot{q}$ field	–	5	% $\dot{q}$

hub cavity exit and was calculated using an inverse FEA analysis with the IR temperature field as boundary condition. Since the maximum error of lateral conduction only exists in a small area, the maximum error was converted into a standard deviation by assuming a triangular probability distribution. The standard deviation of the final heat flux measurement is slightly different for every heating foil and equals approx. 5%.

Figure 12 shows the 95% confidence interval of h at midspan of the strut and at the hub centerline along the strut chord length. The uncertainty is highest towards the beginning of the hub and towards the strut leading edge because the temperature difference is the lowest there and towards the strut trailing edge because the heat transfer coefficient is lowest there. Figure 13 shows the 95% confidence interval of η . With increasing film cooling, the measured temperature differences increase and the relative uncertainty decreases. In comparison, Arroyo Osso et al. [11] investigated the heat transfer coefficient in a TCF using thermography and water-heated walls. For h and with a 95% confidence interval they recorded error levels between -6% and $+16\%$ on the hub and -34% and $+54\%$ on the struts. Lazzi Gazzini et al. [13] measured h and η on a rotor endwall with an error ranging from $\pm 7\%$ to $\pm 12\%$ for h and ± 0.05 to ± 0.2 for η , using the same confidence interval.

6 Summary and conclusions

This study presented a measurement technique for the convective heat transfer coefficient, h , and the adiabatic film cooling effectiveness, η , using thermography and flexible heating foils. The application of the measurement technique was demonstrated in a sector-cascade test rig of a turbine center frame (TCF), an inherent component of modern two-spool turbofan engines. The geometry of interest was milled from low conductivity Rohacell foam, producing a quasi-adiabatic surface for the heating foils, which were coated with a high emissivity paint. The optical access was enabled through two trapezoidal BaF₂ windows. The backbone of the measurement method is an elaborate data reduction, calibration and correction scheme:

1. Any sensor drift of the IR cameras microbolometer, the paint's emissivity and the windows transmissivity, as well as their angular dependence, are recorded and corrected in the multistep pre-calibration conducted in a vacuum chamber. This allows for very shallow viewing angles with respect to the investigated surface and the window without compromising the accuracy of the measurement.

2. By mapping the 2D infrared images on the 3D surface and overlapping them, it is possible to obtain temperature, h and η surface maps of highly curved surfaces covering areas significantly larger than the window's size.
3. The heat flux is corrected for the temperature dependence and inhomogeneity of the heating foils, and a 1D heat loss correction for thin parts is implemented.
4. The measurement technique was validated by comparing the data against flat plate correlations and also by confirming the linearity of ΔT and $\dot{q}$.

The horseshoe vortices were found to play a major role for the heat transfer and film cooling in turbine center frames: In the vicinity of the horseshoe vortices' onset, the heat transfer was intensified by up to 10%, and at the same instance, the cooling film was diluted and swept away. It was also found that the horseshoe vortices lift off from the surface at 50% hub length, resulting in a pair of counter-rotating vortices. From there, the detrimental influence on heat transfer and film cooling effectiveness vanished. Close to the TCF exit on the hub, the now lift-off pair of counter-rotating vortices even contributed to a local heat transfer reduction. The inlet pegs led to an additional increase in heat transfer (approx. + 10%) in the first third of the TCF and an additional reduction of film cooling effectiveness and coverage. This dangerous combination of intensified heat transfer and reduced cooling in the first half of the TCF must be addressed in the design process.

Two adaptations, a quasi-adiabatic surface and optical access, are the essential prerequisites for the applicability of the herein proposed measurement technique. Therefore, this measurement technique can be transferred with low effort to other applications in and outside the field of turbomachinery. Possible practical applications can be found in all disciplines that have to deal with heat transfer and or film cooling problems. In turbomachinery, this technique could be used to precisely study the thermal behavior of combustors (as steady-state cold flow models), turbine stages and turbine exit casings. Outside of the field of turbomachinery, this technique could contribute to a better understanding of complex heat exchanger geometries, film-cooled rocket nozzles (as steady-state cold flow models), supersonic vehicles and earth re-entry vehicles with complex shapes.

Authors' contribution Patrick Jagerhofer: development of measurement technique, conduction of measurements and post-processing, wrote the paper and created the images. Jakob Woisetschlager: pre-calibration concept development, wrote the paper. Gerhard Erlacher: design, assembly and commissioning of sector-cascade test rig. Emil

Göttlich: heating foil and test rig concept development. All authors read and commented on previous versions of the manuscript and read and approved the final manuscript.

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Code availability The following commercial software was used in the following order: Flir—ResearchIR, Matlab with Flir Atlas SDK, Tecplot.

Declarations

Conflict of interest The authors have no relevant financial or non-financial interests to disclose.

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