The Influence of Combustor Swirl on Turbine Stator Endwall Heat Transfer and Film Cooling Effectiveness in a 1.5-Stage Axial Turbine

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Abstract
Modern low-emission combustor concepts, such as lean combustion pose challenges to turbine design in terms of a flatter temperature profile with increased thermal load towards the endwalls. Moreover, a highly 3-dimensional inflow condition of both total pressure and inflow angles is present. This means, that a careful understanding of heat transfer and sophisticated cooling techniques are required with respect to the turbine’s nozzle guide vane. A large scale 1.5-stage axial turbine rig has been set up to examine varying inflow conditions to the turbine. This work presents commissioning results of simultaneous heat transfer and film cooling effectiveness measurements by the means of infrared thermography with the auxiliary wall method, yielding high spatial resolution. Gas concentration measurements are used as a reference. Results are shown for swirling inflow, typical for a state-of-the-art aero-engine lean combustor and compared to a baseline case with axial inflow for different coolant injection rates. Nusselt numbers show the typical characteristics for a turbine vane endwall. An increase is observed with increased coolant injection, except for the trailing edge region, where a decrease is achieved for the design configuration. Film cooling effectiveness values determined by the auxiliary wall method show a good agreement with gas concentration measurements for the upstream part of the endwall. Swirling inflow increases Nusselt numbers within the passage, while film cooling effectiveness values are decreased to 75% resp. 50% of their level compared to axial inflow.

Keywords: heat transfer, combustor turbine interaction, infrared thermography, gas concentration

Nomenclature

c Concentration
D Cooling hole diameter
dim. Dimension
ETFE Ethylene tetrafluoroethylene
HTC Heat transfer coefficient,
\( HTC = \frac{q}{(T_w - T_{Ref})} \)
I Momentum ratio, \( I = \frac{\rho_c u_c^2}{\rho_\infty u_\infty^2} \)
IR Injection rate, \( IR = \dot{m}_c / \dot{m}_\infty \)
k Pressure coefficient
\( \lambda \) Thermal conductivity
L Cooling hole length
LSTR Large Scale Turbine Rig
M Blowing Ratio, \( M = \frac{\dot{m}_c}{\dot{m}_\infty} \)
ME Measurement plane
NGV Nozzle guide vane
NO\(_x\) Nitrogen oxides
Nu Nusselt number, \( Nu = \frac{HTC \cdot c_{real}}{\lambda_{air}} \)
NUC Nonuniformity correction
\( \eta \) Film cooling effectiveness,

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\[ \eta = \frac{\bar{c}_\infty - \bar{c}_w}{\bar{c}_\infty - \bar{c}_c} \approx \frac{\bar{c}_0 - \bar{c}_w}{\bar{c}_0 - \bar{c}_c} \]

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tr>
<td>p</td>
<td>Pressure</td>
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<tr>
<td>( \dot{q} )</td>
<td>Local heat flux, ( \dot{q} = -R_{th} \cdot (T_W - T_R) )</td>
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<tr>
<td>( R_{th} )</td>
<td>Thermal resistance, ( \frac{1}{R_{th}} = \frac{s_{ETFE}}{\lambda_{ETFE}} + \frac{s_{coat}}{\lambda_{coat}} )</td>
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<tr>
<td>RIDN</td>
<td>Rear Inner Discharge Nozzle</td>
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<tr>
<td>s</td>
<td>Layer thickness</td>
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<tr>
<td>( \tau )</td>
<td>Transmissivity</td>
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<tr>
<td>T</td>
<td>Temperature</td>
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<tr>
<td>( \theta )</td>
<td>Non-dimensional temperature ratio, ( \theta = \frac{(T_\infty - T_c)}{(T_\infty - T_0)} )</td>
</tr>
<tr>
<td>( x/c_{ax} )</td>
<td>Axial position relative to axial vane chord length</td>
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**Subscripts**
- ad: Adiabatic
- air: Air property
- ax: Axial chord
- B: Base material
- c: Coolant flow property
- coat: Nextel velvet (paint) coating
- dyn: Dynamic
- ETFE: Ethylene tetrafluoroethylene
- norm: Normalized to ISA conditions
- p: Pressure
- Ref: Reference value
- Real: Real chord
- t: Stagnation value
- th: Thermal
- w: Wall
- \( \infty \): Main flow property

**Introduction**

In modern gas turbine engines, turbine inlet temperatures are well above the melting temperature of any useful material. Hence, sophisticated cooling technologies and a thorough understanding of heat transfer are necessary for turbine design, especially for the high pressure turbine. Both airfoils and endwalls of the first stator stage, the so called NGV are extensively film-cooled to achieve a protective layer and to reduce air temperatures in the vicinity of the surface. This coolant is supplied by compressor bleed.

Driven also by legislative requirements, a combustor concept known as lean combustion is developed to lower emissions with the focus on NOx according to Lazik et. al. [1]. A swirl stabilized combustion with a lean fuel-to-air-mixture is used to achieve this goal. More air enters the combustor already at its inlet and less dilution air is injected at the combustor walls.

This results in increased residual swirl at turbine inlet and a performance impact is observed as Schmid et. al. [2] show. In addition, there is an influence on the distribution of cooling films on both endwall and airfoil of the first stator vane row. A more homogeneous temperature distribution with lower peak temperatures, but increased thermal load near the endwall is generated.

A review on endwall heat transfer studies can be found in Friedrichs [3]. He states three main factors with influence on endwall heat transfer: The inlet temperature distribution, the thickness of the boundary layer and the three-dimensional flow field near the endwall. The latter is predominantly influenced by the secondary vortex system in a vane passage, which is also the crucial factor determining endwall heat transfer, according to Han et. al. [4]. The flow situation within a vane passage is explained in detail by the model from Goldstein and Spores [5].

Endwall film cooling in axial turbines has been studied experimentally by Friedrichs. Accordingly, the flow field within the passage is dominating cooling as well. The effect of surface curvature on film cooling effectiveness has been studied by Schwartz et al. [6]. They state that for low blowing ratios, an improvement is observed for convex shaped surfaces downstream of the injection. The setup examined in the presented analysis is similar to flow conditions and the geometrical setup found in their work.

An important aspect in endwall cooling is the region downstream of the separation line induced by the horseshoe vortex. Benton et. al. [7] illustrate ways to achieve cooling of this region by injecting coolant upstream of the leading edge at a specific pitch wise location. They mention that this is a useful option, since injecting at this position is less detrimental in respect to losses: The coolant is injected into a lower Mach number flow, compared to cooling holes on the endwall within the passage.

Film Cooling is also studied in detail by Dückershoff [8]. For an injection from a single hole, a typical vortex pattern is illustrated, containing a horseshoe vortex that creates two downstream maxima in heat flux adjacent to the injection from the centerline rather than directly downstream of the hole. He lists main factors of influence to film cooling such as, among others, main flow turbulence, blowing ratio \( M \), momentum ratio \( I \) and geometrical parameters of the cooling injection. Thole et. al. [9] show that the penetration depth of film cooling jets into the freestream is determined by \( I \). Accordingly, Baldauf et. al [10] show that film
cooling effectiveness scales with $I$ in the near hole region scales, while it varies with $M$ in the downstream region.

Gritsch et. al. [11] list the non-dimensional temperature ratio $\theta$ as parameter of influence. By using a superposition approach they have shown, that a linear relationship exists between the heat transfer coefficient $HTC$ and $\theta$, when the governing flow conditions are kept constant. At least two measurements at different $\theta$ with corresponding $HTC$ are required to determine the adiabatic heat transfer coefficient $HTC_{ad}$ and the non-dimensional temperature ratio $\theta_{ad}$ with adiabatic wall conditions, as it was defined by Goldstein [12] to describe heat transfer in film cooling applications. The inverse of $\theta_{ad}$ corresponds to the adiabatic film cooling effectiveness $\eta_{ad}$. This context is also mentioned by Astarita and Carlomagno [13] for high Mach number flows.

**Test rig description and aerodynamic setup**

The experiments are conducted at the Large Scale Turbine Rig (LSTR) of the Institute of Gas Turbines and Aerospace Propulsion at Technische Universität Darmstadt, Germany (Figure 2). The test rig is set up in a closed loop. A centrifugal compressor is used to drive the main flow and a secondary blower provides cooling air. Mass flow is controlled using venturi pipes and orifices. Reynolds-similarity to a real engine high pressure turbine is achieved, whereas the rig is operated at low Mach numbers. The measurement section contains a reconfigurable combustor module to vary the inflow condition to the turbine (Figure 1).
The flow in the combustor module is non-reacting and hence isothermal with near ambient temperature. For the presented measurement campaign, both a clean annulus configuration with axial, low-turbulent inflow and a lean combustion aerodynamic setup with swirling, turbulent inflow are examined. An engine-realistic whirl angle and pressure distribution is modeled at turbine inlet with a homogenous radial temperature distribution. The setup has been numerically examined by Schmid et. al. [2] with averaged whirl angles at turbine inlet of 14° near the casing, 11° near the hub endwall and local peak angles of up to 25°.

The turbine stage contains 24 film-cooled NGV and a hub side endwall RIDN-coolant injection. There are 12 swirler modules in the lean combustion configuration, which can be completely disassembled for the baseline reference configuration. Both swirlers and NGVs can be traversed in circumferential direction to allow for clocking examinations. For the present investigation, the swirler core is aligned to the center of a vane passage. Therefore, two adjacent vane passages face different inflow conditions, denoted as condition A (aligned with the swirl center) and B in the discussion of results. Two adjacent NGV including a section of the hub and casing side endwall can be replaced, housing a variable measurement module (schematic in Figure 3). This offers the possibility to implement measurement techniques to the NGV endwall and also contains the thermo-optical access to the hub side surface.

The endwall coolant injection is located directly upstream of the measurement module. The gas path and a sectional view is shown in Figure 4. More details of the design and all capabilities of this test rig can be found in Krichbaum et. al. [14].

Figure 4: Test rig sectional view with measurement planes (ME), RIDN coolant and main flow gas paths

Experimental setup – infrared thermography

The test rig is controlled to fulfill constant reduced mass flow and spool speed parameters during different test days. This reduced mass flow is calculated at turbine inlet (ME01). Total pressure and temperature profiles are determined using rakes at the turbine inlet and then mass flow averaged. The temperature of main and secondary airflow are individually controlled by two water-driven air coolers.

The operating point with swirler modules installed is set similarly. The total inlet pressure contour with implemented swirler features an area of low pressure caused by a recirculation zone of the swirler module. The inlet total pressure profile to the turbine is shown in Figure 5. The low pressure zone of two adjacent swirlers can be seen towards the sides, reaching a difference to the mean total pressure of up to 1.5, compared to the dynamic pressure at ME01.

\[ k_p = \frac{p_t - \bar{p}_t}{\bar{p}_{dyn}} \] (1)
The operating point has been set such that the same level of area averaged pressure is achieved compared to axial inflow in ME01. Experimental data has been acquired for four different coolant injection rates, with the parameters shown in Table 1.

| $IR \text{ [\% } \dot{m}_{\infty}]$ | 0.4 | 0.9 | 2.0 | 2.9 |
| $M$ [-] | 0.4 | 0.9 | 2.0 | 2.9 |
| $I$ [-] | 0.2 | 0.9 | 5.0 | 10.2 |
| $\Delta(T_{\infty} - T_r)$ [K] | 10 |
| $DR$ [-] | 1.04 |

The coolant injection module consists of two rows of 20 (downstream) resp. 19 (upstream) cylindrical holes per vane passage, a transversal and axial spacing of 3 D and a L/D ratio of 6.5. The second row of holes is positioned 1 D upstream of the experimental endwall measurement modules. The holes are inclined by 60° towards the main flow direction which corresponds to 20° to the wall surface. The coolant module has been designed for the injection rate highlighted in bold.

A FLIR x6540sc infrared camera is used to detect surface temperatures on the stator endwall. Its InSb detector has a usable wavelength range of 1.5-5.1 μm and a resolution of 640x512 pixels. Data is recorded for 30 seconds with a frame rate of 10 Hz leading to 300 thermograms per camera position, operating temperature and $IR$. In total, three camera positions are used to cover the whole endwall area, resulting in a minimum spatial resolution of 0.4 mm. Temperatures are recorded through a coated CaF$_2$ window with a transmissivity $\tau$ of an average of 0.98 in the range of 1.5-4 μm.

The endwall aluminum body (1, Figure 6) is heated by kapton foil heaters (3) to six quasi-steady-state conditions with constant wall temperature with steps of 5 K. Nine heater foils with a maximum power output of 1 kW with a 12 VDC power supply have been applied and controlled by pulse width modulation to the desired heat flux setting. The surface is covered by an auxiliary wall (2) of ETFE of an average thickness of 0.82 mm and a Nextel Velvet Coating paint of 0.12 mm with an emissivity of 0.97 (Lohrengel et. al. [15]). The base material temperature is monitored by an array of 59 type K thermocouples (4). The corresponding measurement module is shown in Figure 7. 20 type K reference thermocouples (5) are placed underneath small cylindrical metal pins with a diameter of 1.5 mm, which are only coated by the paint layer.
image. The measurement method has been developed and tested at a linear cascade wind tunnel. Experimental results and a more detailed description of the experimental technique are found in Werschnik et. al. [16].

To calculate local heat flux rates, several geometrical and thermal parameters had to be determined. The layer thickness $s$ of both ETFE and paint have been measured for the for the whole endwall area using a laser-triangulation device with an accuracy of about 20 $\mu$m. The grid spacing is 0.25 mm.

By applying a 3-dim.-2-dim. projective transformation described in Laveau et. al. [17], a correspondence between the camera image and the positions in the experimental setup is established. Since the reference thermocouples are clearly identifiable in the thermographs they are used as geometric reference. The geometrical and thermal parameters based in the 3-dim. coordinate system, namely the local surface thickness and the thermocouple readings underneath the auxiliary wall, are transformed and interpolated onto the grid of the camera image.

The thermograms are acquired, averaged and calibrated at each constant heater setting. Before the data is averaged, a nonuniformity correction (NUC) is performed. The object signals detected by the camera are calibrated in-situ using the reference thermocouples. Since they are also covered by the low-conductive paint layer, a temperature correction needs to be conducted for the thermocouple reading. The calibration procedure used is fully described in Ochs et. al. [18]. Additional information regarding infrared image calibration can be found in Schulz [19] and Martiny et. al. [20].

Together with the thermocouples underneath the auxiliary wall as well as the layer thickness of the auxiliary wall, local heat flux values $\dot{q}$ can be determined for each of the six heater settings with equation (2).

$$\dot{q} = -R_{ch} \cdot (T_W - T_B)$$ (2)

With the temperature difference between main flow and coolant, adiabatic heat transfer coefficients and film cooling effectiveness are acquired with the superposition approach of Gritsch et. al. [11] using a linear regression.

![Figure 7: Coated instrumentation carrier before painting, thermal/geometrical reference positions visible](image)

The thermal values calculated in the grid of each camera image are then transformed into the 3-dim. coordinate system using the reference thermocouple positions as geometric reference. Data is averaged for areas, where information from more than one camera position is available.

Supportive 3-dim. steady state thermal FEM-analysis is used to examine the measured data for undesired heat flux within the auxiliary wall. The thermal loads measured on both sides of the auxiliary wall are used as boundary conditions. A comparison between the FEM calculation and the 1-dim. analytical solution is conducted. Except for the upstream and downstream boundary to the non-heated casing rings, no significant heat flux in transversal direction within the auxiliary wall exist: The values do not exceed 3% of the normal heat flux and are much lower for most parts of the endwall. Hence, the assumption of 1-dim., normal heat conduction through the auxiliary wall is justified. The data analysis procedure for the infrared thermography measurements has been developed and defined by Steinhausen [21] during commissioning of the test rig.

**Experimental setup – CO₂ gas concentration measurements**

CO₂ gas sampling with an array of small holes on the endwall is used to determine adiabatic film cooling effectiveness values by the heat and mass transfer analogy with a separate experiment and instrumentation carrier. A total of 120 measurement positions (see Figure 8) is used for gas sampling, covering the hub side NGV endwall area between -5.5 % and 100 % axial chord length. There are eleven sets of holes with constant $x/C_{ax}$...
position. The holes have a diameter of 0.5 mm and are drilled perpendicular to the endwall surface. They are connected metal tubes through the measurement line chamber inside the NGV to the gas analyzers. The velocity of gas sampling is selected in such a way, that no influence on the boundary layer occurs in accordance with the measurements of Schrewe [22]. The level of CO₂ is increased to a steady state concentration of about 14,000 ppm in the secondary air, which is measured through a wall tapping of similar diameter as the sampling holes in the coolant plenum. Main flow CO₂ concentration is kept at a constant level of about 2,000 ppm by a partial open loop configuration of the test rig and measured using a pitot probe in the main flow. Adiabatic film cooling effectiveness values are calculated with equation (3)

\[ \eta_{ad} = \frac{c_{\infty} - c_w}{c_{\infty} - c_t} \]  

(3)

Figure 8: CO₂ gas concentration measurements – endwall sampling taps

Experimental results – axial inflow

Figure 9 show the pitch wise average of Nusselt numbers along the axial vane chord length for all cooling configurations studied. Without coolant injection, a heat transfer peak is present upstream of the leading edge. Nusselt numbers then decrease to lower values of about 1000 and then increase again towards the trailing edge up to 2700. This increase is caused mainly by the increasing flow velocity in the region. Nusselt numbers are increased with higher injection rates. This is most prominent upstream of the leading edge, close to the injection location.

For IR 2.0 and 2.9, Nusselt numbers are significantly increased also within the vane passage by one third, resp. two thirds. In the trailing edge region, heat transfer is decreased compared to the case without coolant injection. According to Friedrichs [3], the peak in this region is caused by the downwash created by the trailing edge vortex. It needs to be studied with further experiments during the course of the measurement campaign, which flow phenomena are responsible for the attenuation of heat transfer. A comparison to numerical simulations will be conducted as well.

Figure 9: Pitch wise averaged Nusselt numbers along axial vane chord length

The general characteristics of a peak upstream of the leading edge, with then decreasing values and a subsequent increase towards the trailing edge can be found for all injection rates, as it is also observed by Lorenz [23] for a vane endwall passage.

Figure 10: Heat transfer, IR=2.0, axial inflow

Film cooling effectiveness values for IR 0.4 (Figure 11) agree with data found in Knost et. al [24], showing significant film cooling to the endwall area upstream of the separation line for coolant injection into the incoming boundary layer only. The injected coolant is driven from the pressure side endwall area towards the suction side due to the low momentum ratio and the trajectory of the pressure side horseshoe vortex is visible on the endwall. At about midpassage, discrete steps in the cooling effectiveness contour are visible. Difficulties with
the third camera position, directed towards the throat of the vane endwall area, are responsible. This aspect will be addressed with the comparison to the gas concentration measurements.

**Figure 11:** Film cooling effectiveness, IR=0.4, axial inflow

For higher injection rates and momentum ratios, the cooling is improved, such that the whole endwall area receives cooling (Figure 12). A comparison of film cooling effectiveness values determined with infrared thermography and gas concentration measurements is shown for three lines, highlighted A, B and C.

**Figure 12:** Film cooling effectiveness contour for IR 2.0, axial inflow. Three lines (A, B and C) are shown for comparison to gas concentration measurements

For higher IR towards the coolant injection design point, film cooling effectiveness values of both measurement techniques show good agreement in the upstream half of the vane passage. Both close to the leading edge, where discrete film cooling streaks can be identified (Figure 13) and at 50% axial chord length (Figure 14), where the cooling air is already mixed out with the near-endwall flow, the results are comparable, while slightly overpredicted for the infrared measurements.

**Figure 13:** Comparison of film cooling effectiveness value for infrared thermography and gas concentration measurements, x/cax=0 (line A)

**Figure 14:** Comparison of film cooling effectiveness value for infrared thermography and gas concentration measurements, x/cax=0.5 (line B)

At the trailing edge (Figure 15), film cooling effectiveness is overpredicted by the infrared measurements, especially for the camera position viewing towards the trailing edge. The thermal calibration of the infrared image in this area is complicated due to the high heat transfer, which compromises the reference temperature reading. Moreover, due to the high heat transfer in this region, the difference in temperature between main flow and wall is small. This leads to a poor robustness of the linear regression used to extrapolate the adiabatic wall temperatures and film cooling effectiveness values.
Experimental results - swirling inflow

Figure 16 and Figure 17 show the Nusselt number contours for swirling inflow for both flow conditions A and B. It is observed, that heat transfer in general is increased throughout the vane passage, compared to axial inflow, which is in accordance with the numerical simulation by Schmid et. al. [2]. The increase is even greater for flow condition B.

Figure 18 and Figure 19 show the film cooling effectiveness values for swirling inflow and both flow conditions. The cooling effectiveness level is decreased compared to axial inflow. The decrease is slightly more pronounced in vane passage B. In the midpassage area, a constant level of about $\eta=0.2$ is achieved where the coolant streaks are mixed out, compared to an average of $\eta=0.35$ for the axial inflow condition.
The calculation of the film cooling effectiveness still needs to be improved towards the trailing edge region, as the comparison to the gas concentration measurements already shows. When considering adiabatic wall temperature for the swirling inflow case, it is observed that the values are gradually underestimated towards the trailing edge in comparison to the incoming flow temperature by up to 2 K (Figure 20).

**Conclusion and Outlook**

Experimental heat transfer and film cooling effectiveness contours have been determined on the hub side endwall of a 1.5-stage axial turbine. Infrared thermography was used with the auxiliary wall method to determine local values with high spatial resolution. For the baseline case with axial inflow, heat transfer is shown to increase with coolant injection throughout the vane passage. Towards the trailing edge region, a decrease in heat transfer is observed for the design cooling configuration. Film cooling effectiveness values are determined with the superposition approach in the same experiment by a linear regression. They are compared to adiabatic film cooling effectiveness values determined by gas sampling measurements. The comparison shows good agreement in the upstream part of the endwall. Towards the trailing edge, film cooling effectiveness is underestimated using infrared thermography.

A lean combustion inflow condition is examined as well, with a total pressure contour at the inlet to the turbine stage, generated by a swirlier configuration. The swirler-vane-count is 1:2 and a clocking position of swirler core relative to the center of a vane passage was examined. Nusselt numbers increase throughout the passage for both flow conditions, whereas this increase is more significant for passage B. In the trailing edge region, Nusselt numbers are decreased. Film cooling effectiveness values are significantly lowered, reaching only about 60 % of the baseline level with axial inflow in the center of the vane passage.

During the upcoming measurement campaign, further clocking positions will be examined. Moreover, it is planned to conduct turbulence level measurements using hot-wire-anemometry. The swirling inflow configuration will be compared to an axial inflow condition with a higher turbulence level, comparable to the one generated by the swirling turbulent configuration.

**Acknowledgments**

The work reported was partly funded within the framework of the “AG Turbo” by the Federal Republic of Germany, Ministry for Economic Affairs and Energy, according to a decision of the German Bundestag (FKZ: 03ET2013K) as well as by Rolls-Royce Deutschland GmbH and ALSTOM Power. The contribution of Christoph Steinhausen to the development of the data analysis procedure, Marcel Adam and David Neubauer to the instrumentation of the experiments and Bergmaier GmbH for the manufacturing expertise of the instrumentation carriers is gratefully acknowledged.

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