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HEAT TRANSFER ANALYSIS IN AN ENGINE-REALISTIC TURBINE REAR STRUCTURE

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ABSTRACT

Rising turbine inlet temperatures and pressure ratios in modern engines increase the challenges of thermal management in the turbine rear structure (TRS). To meet these challenges, the TRS must withstand thermal loads across the full operating range. However, low-power conditions, such as idle and sea-level take-off can introduce large deviations from the on-design conditions, leading to significant changes in heat transfer and cooling effectiveness. Therefore, although industrial design practices typically consider these operating conditions crucial for TRS thermal performance, the experimental validation of numerical tools is very limited.

In this work, experiments were conducted in Chalmers 1.5-stage LPT–OGV facility at engine-representative Reynolds numbers and swirl angles. The investigated TRS was equipped with twelve engine-realistic outlet guide vanes (OGVs). Surface temperatures were measured using an infrared camera used to evaluate OGV heat transfer and hub-region film cooling effectiveness. Complementary steady-state RANS CFD simulations were used to evaluate the predictive capability of current industrial models.

This study shows that OGV heat transfer is strongly influenced by laminar–turbulent transition and the spanwise variation in the aerodynamic loading. It is found that the heat transfer distributions are clearly dependent on the inlet swirl angle, where the off-design cases representative of sea-level take-off and idle swirl lead to a fundamentally different flow and substantially modified heat transfer on the OGV. For the first time, endwall film cooling effectiveness is measured in an engine-realistic TRS at on-design and idle off-design conditions. While relatively effective at on-design condition, the purge film shows significantly lower effectiveness at idle, resulting in reduced cooling capacity in the immediate vicinity of the OGV. The CFD captures the overall qualitative trends but remains limited in predicting heat transfer and film cooling effectiveness in all areas, highlighting the need for comprehensive experimental validation.

Keywords: Turbine rear structure, turbine rear frame, engine exit structure, turbine exhaust casing, tail bearing housing, outlet guide vane, low-pressure turbine, engine-mount recess, bump, experimental, measurement techniques, fluid dynamics in turbine components of gas turbine engines, heat transfer, film effectiveness

NOMENCLATURE

α	Inlet swirl angle [deg]
d_{leads}	Diameter of thermocouple leads [m]
d_{probe}	Diameter of thermocouple probe [m]
γ	Intermittency [-]
h	Local heat transfer coefficient [W/(m ² ·K)]
H	Channel height [m]
k	Thermal conductivity [W/(m·K)]
L	Chord length [m]
ν	Kinematic viscosity [m ² /s]
Nu	Nusselt number [-]
q''_{cond}	Conductive heat flux [W/m ²]
q''_{conv}	Convective heat flux [W/m ²]
q''_{rad}	Radiative heat flux [W/m ²]
Re	Reynolds number [-]
Re_{θ_t}	Momentum-thickness Reynolds number [-]
T_{aw}	Adiabatic wall temperature [K]
T_p	Purge flow temperature [K]
T_w	Wall temperature [K]
T_∞	Temperature of the main flow in the TRS [K]
x	Streamwise coordinate [m]
η	Film cooling effectiveness [-]
ϕ	Flow coefficient [-]

ABBREVIATIONS

ADP	Aerodynamic design point
BPR	By-pass ratio
CFD	Computational Fluid Dynamics
CFX	ANSYS CFX solver
FOV	Field of View

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GTF	Geared turbofan engine
HWA	Hot-Wire Anemometry
IR	Infrared
LPT	Low-pressure turbine
NGV	Nozzle guide vane
OGV	Outlet guide vane
PIV	Particle Image Velocimetry
PS	Pressure side
PT100	Platinum temperature sensor
RMA	Reflective Marker Array
RTD	Resistance temperature detector
SS	Suction side
SST	Shear-Stress Transport turbulence model
TRS	Turbine rear structure

1. INTRODUCTION

The growing demand for sustainable air transport continues to push the aerospace industry toward more efficient and low-emission propulsion systems. With current trends towards lighter and compact aircraft engine designs, reducing the size of the engine core and increasing the bypass ratio (BPR), the optimization of each component becomes crucial. One such component is the turbine rear structure (TRS), located at the aft end of the engine. The TRS provides mechanical support and alignment for the low-pressure turbine (LPT) stages. Moreover, it accommodates the rear bearing of the low-pressure spool and serves as the structural interface for the rear engine mount, transferring mechanical loads from the engine to the airframe (Figure 1).

The design of TRS can vary between engines from different manufacturers. However, generally TRS consists of struts connecting the inner and outer endwalls, commonly known as the hub and shroud, respectively. The flow exiting the final LPT stage is strongly non-uniform in the radial direction, with swirl angles ranging from -40 degrees near the hub to nearly axial at the shroud. To maximize the axial momentum and minimize the mixing losses between the core and by-pass flows, the TRS must redirect this non-uniform swirl toward an axial direction. For this reason, the structural struts are also called outlet guide vanes (OGVs) due to their aerodynamic function of guiding the LPT exit flow and removing residual swirl.

The exhaust gases passing through the OGVs remain hot, as they retain significant thermal energy after passing through the turbine stages. As a result, the OGVs are exposed to high temperatures, while other regions of the TRS, particularly those around the bearing supports, experience significantly lower temperatures. Combined with upstream flow non-uniformities and residual swirl, this temperature contrast leads to significant thermal gradients within the TRS.

Additional complexity arises from the interaction between the main flow and the purge cooling flow, which is extracted from the compressor stages. The purge cooling flow enters the TRS with its own swirl orientation and interacts with the highly swirled TRS flow in the hub region, leading to a strongly three-dimensional and complex local flow field. The degree of mixing between the purge and core flows strongly depends on the engine operating conditions and the purge mass flow rate. Beyond the typical operating conditions considered in aerodynamic design, such as

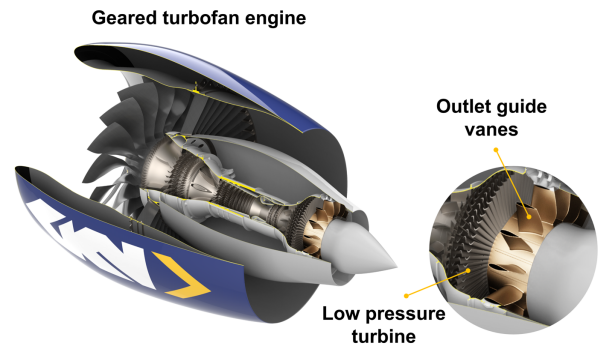


FIGURE 1: Geared turbofan engine and outlet guide vanes. Image © GKN Aerospace.

cruise, take-off and climb, the engine also experiences a range of low-power regimes. During these conditions, the inlet swirl to the TRS can deviate substantially from its design value. In particular, idle operation represents a severe off-design condition where the very large swirl angle leads to a heavy flow separation from the OGV and, consequently, may lead to a significant reduction in cooling effectiveness. As a result, this severe off-design condition becomes highly relevant and should be taken into account when considering the thermal loads within the TRS.

All of these factors lead to a highly complex and non-uniform heat transfer environment within the TRS, promoting thermal fatigue and reducing component durability. Addressing these issues highlights the importance of investigating TRS flow and heat transfer mechanisms.

The flow within the TRS is governed by physics similar to other stators of axial turbines. Sieverding [1] described the fundamental secondary flow mechanisms in turbine blade passages, showing that the roll-up of the endwall boundary layer in the leading edge region forms a horseshoe vortex, which subsequently develops downstream into passage and suction-side corner vortices. Langston [2] later synthesized experimental and analytical studies and demonstrated that these secondary flow structures play a dominant role in flow turning and aerodynamic loss generation in axial turbine stators. Further complexity within the TRS arises from the interactions of OGV boundary layers, tip clearance flow, purge flow, steady NGV wakes, and unsteady LPT wakes.

The initial steps toward a comprehensive understanding of the TRS flow were taken by Hjärne et al. [3], who designed a linear cascade test rig dedicated to LPT/OGV flow investigations. Using this facility, Hjärne et al. [4] analyzed both on- and off-design performance, focusing on pressure distribution, turning, and loss generation. The experimental results were complemented by numerical simulations employing different turbulence models to compare and validate the findings. Based on these comparisons, the use of the realizable $k-\varepsilon$ and $k-\omega$ SST turbulence models with resolved boundary layers was suggested, as they provide conservative predictions of losses and separation. Further insights were gained through experimental [5] and numerical [6] studies of secondary flows and loss development. The experimental data indicated that secondary flow development and loss formation are influenced by blade loading, inlet turbulence intensity, and inlet boundary layer characteristics. The numerical

results showed that the downstream evolution of the secondary flow field was captured, with discrepancies in vorticity intensity increasing at higher loading.

Following the aerodynamic characterization of the cascade, the research focus shifted toward aero-thermal investigations. The same linear cascade facility was employed to perform detailed heat transfer measurements on the OGV surface and endwall under various design conditions [7]. Using a thermochromic liquid crystal technique and complementary numerical simulations, the study analyzed the influence of inlet flow angle, boundary layer transition, and secondary flow structures on local heat transfer. The results showed that boundary layer transition and horseshoe vortex formation strongly affect surface heat transfer, while all turbulence models tended to overpredict heat transfer near the leading edge, particularly under design conditions.

While previous studies have provided valuable insights, limited attention has been given to three-dimensional features associated with the annular configuration of TRS. To address this gap, a 1.5-stage annular test rig equipped with engine-realistic LPT was built at Chalmers University of Technology by Rojo et al. [8]. Using this facility, Rojo [9] obtained initial aerothermal data under engine-representative inlet boundary conditions and established a baseline test case for subsequent experimental and numerical investigations. The same configuration was later utilized by Vikhorev et al. [10] to perform detailed unsteady flow measurements using hot-wire anemometry (HWA) and Particle Image Velocimetry (PIV). The authors examined the development of LPT wake structures and their interaction, leading to a staggered pattern in rotor wakes. Their findings further indicate that downstream velocity and pressure distributions are only marginally affected by rotor–stator wake interactions, resulting in minimal circumferential variations further downstream. More recently, Paier et al. [11] investigated the influence of purge flow on turbulence downstream of the TEGV. Their results showed that purge flow mainly affects turbulence generation and dissipation when unsteady wakes are absent, while in the presence of wakes, the flow behaviour is dominated by wake-induced turbulence with minimal sensitivity to purge variations.

The laminar–turbulent transition on OGV is also among the key mechanisms influencing local heat transfer. Jonsson et al. [12, 13] carried out the first set of measurements in a simplified TRS configuration under engine-representative on- and off-design flow conditions. The results revealed distinct transition and separation regions consistent with previous flow visualization studies and demonstrated the sensitivity of heat transfer to local flow structures. Furthermore, fundamental laminar–turbulent transition correlations were compared with experimental and numerical findings. The experimental methodology and uncertainty analysis developed in these studies were adopted and applied in the present experimental campaign.

Beyond its aerodynamic function, the TRS must also satisfy structural, mechanical, acoustic, weight, and manufacturing requirements, which in turn influence its geometry and consequently affect the overall aerodynamic performance. The structural design often incorporates integrated features, such as oil pipelines for bearing lubrication and localized recesses, or “bumps,” near the engine-mount region. As a result, the OGVs

themselves are subject to geometric modifications and must be optimized to balance mechanical requirements with aerodynamic performance.

Vikhorev et al. [14] conducted detailed experimental investigations on different OGV designs employed in engine-realistic TRS setups. The results showed that the OGV with increased maximum thickness demonstrated aerodynamic performance comparable to the baseline configuration, without a substantial increase in total pressure loss and loss of turning performance. In contrast, the presence of a shroud-placed bump had a more pronounced effect on OGV aerodynamics, extending beyond the shroud region to influence the entire span, including the hub. Although no flow separation was observed on the bump itself under either on- or off-design conditions, its presence changed the inlet total pressure and the swirl profiles, leading to a substantial increase in losses near the hub.

In the present work, a state-of-the-art TRS configuration is investigated, representing a design that integrates all key functional requirements. Unlike simplified setups used in earlier studies, this configuration incorporates multiple OGV designs installed simultaneously, reflecting the engine-realistic arrangement, and features an optimized contouring of the hub and polygonal shroud.

Earlier studies have characterized the aerodynamic performance and loss behavior of this configuration, which serves as the foundation for the current investigation [15, 16]. Through a combination of experimental measurements and numerical analyses, the optimized TRS configuration demonstrated good aerodynamic performance across all OGV types. The aerodynamic characteristics of the baseline and thickened vanes were found to be similar, featuring moderate profile losses and no observable flow separation across the investigated operating range. For OGV with bump, hub losses were effectively minimized through design optimization, while the shroud-located bump introduced a localized vortex region near the shroud as a result of the combined effect of geometry and pressure gradients. Oil-film visualizations and CFD studies revealed that this shroud vortex influences the wake width and local loss distribution, emphasizing the need for an integrated aerodynamic and structural design of both OGV and the bump. Despite the overall agreement between the experiments and the CFD, the numerical simulations tended to overpredict secondary flow structures, particularly near the endwall regions.

Building upon this foundation, the present study provides the first comprehensive investigation of aero-thermal behavior of an engine-realistic TRS under engine-representative on-design and idle-operation conditions. By comparing experimental data with numerical predictions using a state-of-the-art industrial CFD tool, the study provides deeper insight into the complex mechanisms governing surface heat transfer and film cooling effectiveness within the TRS and the cooling effect of the purge flow.

2. EXPERIMENTAL SETUP

Experimental investigations were carried out in a low-speed large-scale 1.5 stage LPT-OGV facility at Chalmers University of Technology, Sweden. The facility was designed to investigate the TRS flow under both on-design and off-design conditions. The operational scope of the facility is defined by the Reynolds

number (Re) and the flow coefficient (ϕ), which are calculated based on the axial velocity and the LPT channel height [14]. The LPT stage comprises 60 nozzle guide vanes (NGVs) and 72 rotor blades, providing inlet boundary conditions that replicate actual engine environments with characteristic wakes, secondary flows, swirl, mass flows, and pressure distributions. Further details of the rig are documented in [8].

The TRS design investigated in the present work is a state-of-the-art configuration similar to currently employed in modern turbofan engines (Figure 2). Both the LPT and TRS designs were done by GKN Aerospace Sweden. However, it shall be noted that the aero surfaces have been developed solely for the experimental rig and are not related to any GKN Aerospace product characteristics.

The hub contour is profiled to gradually contract the flow passage, promoting controlled flow acceleration in the hub region and reducing local aerodynamic losses. The shroud design, driven by manufacturing and mechanical requirements, has a polygonal shape. The TRS integrates a total of 12 OGVs, consisting of 3 thick-, 3 bump-, and 6 regular OGV designs, all assembled simultaneously. The thick OGVs were designed with an increased profile thickness compared to the regular OGVs, while maintaining the same chord length distribution. The bump OGVs incorporate an integrated shroud bump.

The experimental investigation employed two measurement approaches, both performed using the same experimental setup (Figure 3). Temperature measurements were performed using a Phoenix 320 MW-IR camera, with an in-house calibration performed beforehand. Further details of the calibration process can be found in a study by Jonsson et al. [12].

The first set of measurements focused on steady state heat transfer over the OGV. The local heat transfer coefficient was determined from the steady-state surface energy balance as:

$$h = \frac{q''_{\text{conv}}}{T_w - T_\infty}, \quad q''_{\text{conv}} = q''_{\text{cond}} - q''_{\text{rad}}. \quad (1)$$

Here, q''_{conv} is the local convective heat flux to the airflow, expressed as the difference between the conductive heat flux q''_{cond} through the OGV material and the radiative heat losses q''_{rad} . The outer wall surface temperature of the OGV T_w was measured using an IR camera, while the free-stream temperature T_∞ was measured by a downstream RTD sensor in the TRS bulk flow.

The conductive heat flux component q''_{cond} was determined from the temperature difference between the inner and outer surfaces of the OGV shell, accounting for the known wall thickness and thermal conductivity of the OGV material. The inner wall temperature was imposed by heated water circulating through the OGV. For this purpose, one of the thick OGVs (middle of the three) was instrumented with internal water channels (Figure 4a). This OGV was mounted vertically within the TRS module and connected to an external water reservoir. The heated water was circulated through the internal channels by a pump, providing the required flow rate to maintain a constant wall temperature. Two PT100 resistance temperature detectors (RTDs) were installed in the inlet and outlet tubes of the OGV to monitor the water temperature difference, which remained minimal throughout the experiments.

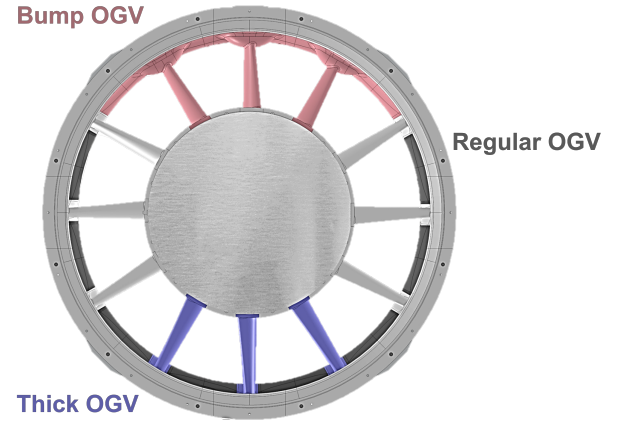


FIGURE 2: Engine-realistic TRS configuration with three OGV types: regular vane (gray), thick vane (blue), and bump vane (red).

The radiative heat flux component q''_{rad} , which is often the most challenging to quantify, was evaluated using a modified version of the Reflective Marker Array (RMA) method, as described by Jonsson et al. [12]. Following this approach, the OGV and endwall surfaces were prepared with two types of coatings of different emissivity: a black high-emissivity Nextel 6081 coating and low-emissivity gold leaf markers (3–4 mm in diameter) applied to the OGV surface (Figure 4b). The combination of these coatings made it possible to eliminate the background radiation component from the heat flux balance equations. Markers were also used as reference points for mapping of infrared images to the OGV surface. In addition, an in-situ calibration was performed by placing an RTD sensor within the camera's field of view (FOV) to compare it with the corresponding infrared camera readings. Further details of the governing equations and data reduction procedure are given in Jonsson et al. [12].

The geometry of the thick OGV introduced a further practical challenge. The pronounced curvature of the suction side limited the infrared camera's line of sight, making full-surface imaging not achievable from a single viewing direction. To overcome this constraint, IR recordings were acquired from multiple viewing angles. This approach allowed both the suction and pressure sides to be captured and increased the overall surface coverage of the OGV.

The leading edge of the OGV was not visible to the IR camera; therefore, three thermocouples were installed at the midspan to obtain the corresponding surface temperatures. The thermocouple probes ($d_{\text{probe}} \approx 0.4$ mm) were embedded in a small adhesive bead to ensure good thermal contact with the OGV surface and to provide a smooth geometric transition between the probe and the wall. The thermocouple leads ($d_{\text{leads}} \approx 0.2$ mm) were routed along the OGV pressure-side surface and further along the trailing edge, where the flow is less sensitive to local disturbances.

The thermocouples were calibrated against RTD sensors in a temperature-controlled water bath, in which the spatial variation of the water temperature was within 0.1 °C. During the calibration, the thermocouple readings were directly compared with those from the RTDs, and an individual temperature offset was applied for each thermocouple.

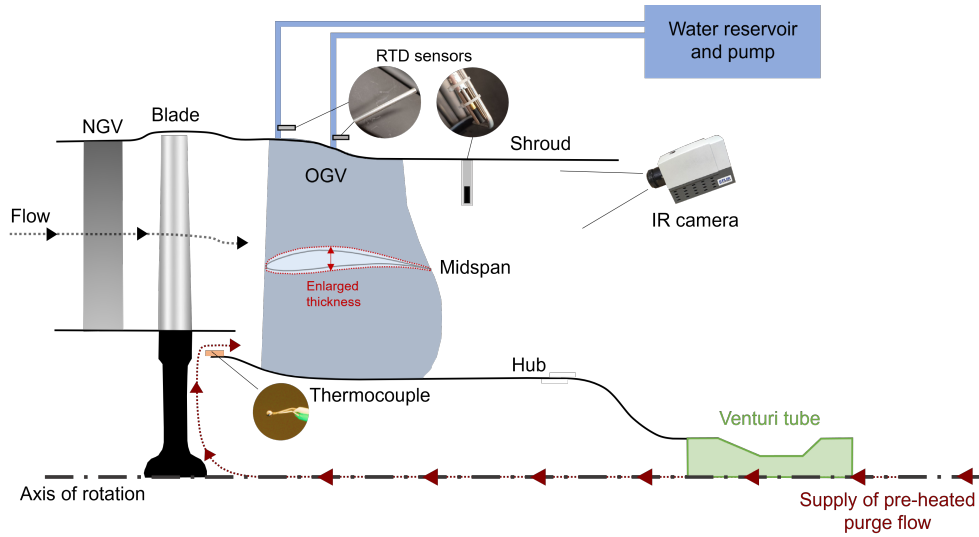


FIGURE 3: Experimental setup

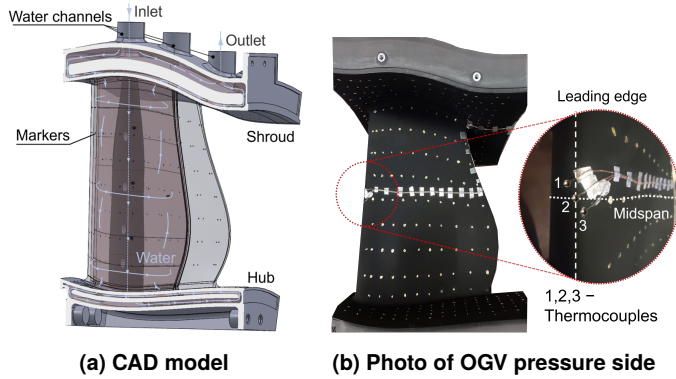


FIGURE 4: Instrumented thick OG with internal channels and leading edge thermocouples.

Thermocouple measurements were performed simultaneously with the IR measurements using the same experimental configuration. Regions in the immediate vicinity of the thermocouples and wiring were excluded from the analysis and subsequently reconstructed using spatial interpolation based on neighboring measurements.

The second set of measurements was used to determine the film cooling effectiveness for the hub endwall under the influence of the purge flow. The purge flow was generated by a fan and pre-heated using an electric heater; thus in the laboratory experiment the purge flow was hotter than the main flow, as opposed to an aeroengine, where the purge flow is colder. The mass flow rate of the purge flow was measured with a Venturi tube and adjusted to 1% of the total TRS mass flow rate for the different Reynolds numbers tested. In contrast to the OG heat transfer measurement configuration, the investigated endwall was adiabatic, with no internal water circulation and, consequently, no heat exchange through the wall material. Therefore, the heat transfer process was governed solely by the interaction between the heated purge flow and the relatively cold TRS flow. Under these conditions, the endwall surface temperature measured by the IR camera is assumed to correspond to the adiabatic wall temperature T_{aw} . The

purge flow temperature T_p was monitored using a thermocouple positioned at the outlet of the purge slot, while the mainstream air temperature T_∞ was measured using an RTD in the same manner as in the previous setup.

The film effectiveness data were acquired in two complementary measurement campaigns. The first measurement campaign covered the region in the vicinity of the OG and within the inter-OGV passage, while the second campaign focused on the upstream region between the OG leading edge and the LPT exit. To obtain this extended coverage, the front part of the OG was also coated with high-emissivity paint and equipped with reflective markers. In addition, the IR camera was positioned so that its FOV covered part of the hub region recorded in the first campaign. Further, the comparison of these values between the two campaigns showed excellent agreement, confirming the high repeatability of the measurements.

In our methodology, the raw infrared images were first processed using a procedure which included a fourth-order non-uniformity correction and a fifth-order calibration polynomial. As a result, surface temperature maps were obtained and the data were complemented with additional thermocouple measurements. The final post-processing stage, performed in MATLAB, was used to determine the local heat transfer coefficient or the film cooling effectiveness, depending on the applied experimental method.

The total uncertainty of the local heat transfer coefficient and the Nusselt number was evaluated using first-order Taylor-series propagation (ASME PTC 19.1), following the methodology described by Jonsson et al. [12]. The resulting combined uncertainty over the entire measured surface is $\pm 8\%$. Uncertainty associated with the thermocouple measurements in the vicinity of the leading edge, including sensor embedding effects, increases the total Nusselt number uncertainty to $\pm 9\%$. For the film cooling effectiveness, the uncertainty was assessed using the same Taylor-series propagation framework and is estimated to be $\pm 7\%$.

Experimental measurements were conducted under both on- and off-design operating conditions of the TRS, defined by the

TABLE 1: Operating conditions for (a) heat transfer and (b) film effectiveness measurements

(a) Heat transfer cases

Case	Condition	Swirl, deg	Flow coefficient	Reynolds number
1	ADP 0 (on-design)	-16.9	0.622	350,000
2	ADP +16.9 (axial, zero-swirl)	0	0.525	350,000
3	ADP +59.4 (idle, mid Re)	+42.5	0.355	350,000
4	ADP 0 (on-design, high Re)	-16.9	0.622	465,000

(b) Film effectiveness cases

Case	Condition	Swirl, deg	Flow coefficient	Reynolds number
5	ADP 0 (on-design)	-16.9	0.622	350,000
6	ADP +59.4 (idle, nominal Re)	+42.5	0.355	100,000

flow coefficient and the Reynolds number. The flow coefficients and their corresponding averaged swirl angles, together with the Reynolds numbers for each case, are presented in Table 1.

For heat transfer measurements (Table 1, *a*), three flow coefficients (Cases 1–3) were selected, representative of current industrial practice to assess on- and off-design operating conditions. The flow coefficient prescribes the sector-averaged inlet swirl angle; however, the swirl distribution varies substantially across the channel. The average swirl is the lowest (largest magnitude) near the LPT hub and increases toward the LPT shroud, where the average swirl angle approaches zero (axial flow) and changes direction to positive in the LPT tip gap.

The typical spanwise swirl angle distribution for the on-design case can be found in [16]. The case names in Table 1 include the value of the average swirl in the TRS relative to the OGV camber line (0, +16.9, and +59.4), where positive incidence is directed toward the suction side. In all three cases (Cases 1–3), the Reynolds number was kept constant. Case 2 (zero-swirl) with the axial flow into the TRS representative to the sea-level take-off condition. In Case 3 (the idle condition), only the swirl angle corresponds to idle operation; the Reynolds number was intentionally increased above its nominal idle value to isolate the effect of swirl on the heat transfer distribution. Case 4 represents the on-design swirl condition repeated at a higher Reynolds number to assess the sensitivity of the OGV heat transfer to the variation of the Reynolds number. The selected Reynolds numbers align with the conditions relevant to larger engines, where increasing mass flow and channel height lead to a higher Reynolds number.

For film effectiveness measurements (Table 1, *b*), the tests were conducted at two engine-representative operating points (Cases

5–6). These cases were selected because they correspond to realistic purge flow cooling scenarios. In contrast to the heat transfer campaign, the idle case here was operated at the engine-relevant Reynolds number of idle operation.

3. NUMERICAL SETUP

The numerical simulations were performed using the commercial CFD solvers ANSYS Fluent 2021R1 and ANSYS CFX 2021R1. Two computational configurations were used in the present work.

The first configuration represents a 30° LPT–TRS sector containing five NGVs, six rotor blades, and one thick OGV. This model has previously been validated by GKN Aerospace for aerodynamic simulations [17, 18] and was re-used in the present work. The configuration was employed for the heat transfer simulations corresponding to experimental Case 1, with the inlet boundary condition taken from the measurements.

The second configuration is identical to the first while includes a purge inlet located in the cavity between the rotor and the TRS hub, as shown in Figure 5. Purge flow simulations were carried out for experimental Cases 5 and 6, with purge inlet conditions prescribed using the measured purge mass flow and the corresponding purge flow temperature.

The mesh generation for the purge flow configuration followed the same procedure as for the first configuration, using ICEM CFD 2021R1 and Fluent Meshing. The near-wall grid is resolved to achieve y^+ values less than unity in all cases.

The steady-state RANS simulations are performed using the four-equation transition SST turbulence model by Langtry and Menter [19], also known as the γ - Re_θ model. The turbulence model relies on the k - ω transport equations with two additional transport equations: one for the intermittency γ , and one for the transition onset criterion expressed in terms of the momentum-thickness Reynolds number Re_θ . All model constants in the turbulence transport equations are kept at their default values in the simulations. Post-processing of the numerical data was conducted in CFX 2021R1 Post and visualized in MathWorks MATLAB.

4. RESULTS

The results are organized into two main subsections according to the applied experimental methodology: the first presents heat transfer measurements on the OGV surface, while the second focuses on the film cooling effectiveness and the hub region.

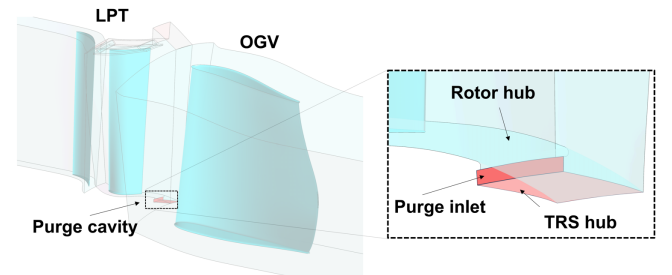


FIGURE 5: Computational domain with hub purge cavity

OGV heat transfer

The results in this section are presented in terms of the Nusselt number (Nu), which represents the non-dimensional local heat transfer coefficient and is defined as:

$$Nu = \frac{hL}{k} \quad (2)$$

where h is the local convective heat transfer coefficient, L is the characteristic chord length of the OGV, and k is the thermal conductivity of the air.

On-design condition. Figure 6 shows the distributions of the Nusselt number on the suction side (SS) and the pressure side (PS) of the thick OGV under on-design conditions. The experimental results are presented on the left, while the corresponding CFD contours are shown on the right for a side-by-side comparison. The OGV geometry is outlined by the grey region in each plot. In the experimental contours, data near the leading edge are missing due to the limited IR camera view, and the trailing edge region is excluded because no uniform heating (no water circulation) was provided there. However, the experimental data for the leading edge region at midspan from thermocouple measurements are presented further.

As observed from the contour plots in Figure 6, a gradual downstream decrease of the Nusselt number from the leading edge takes place on both sides of the vane, a minimum is reached, and is followed by a gradual increase, which are consistent with a growing laminar boundary layer, transition to turbulence, and a growing turbulent boundary layer. CFD data also reveal low-Nusselt number regions near the hub, where a passage vortex affects the heat transfer on the vane surface. The observations

agree very well with the previous aerodynamic studies [15–18]. On the suction side, the low heat transfer region is considerably more pronounced than on the pressure side and is located at approximately 25–50% span. The OGV is more aerodynamically loaded near the hub, which leads to a deeper suction side pressure minimum near the hub compared to the radial spans away from the hub. The corresponding spanwise pressure distributions for the thick OGV were presented in the previous aerodynamic study [15]. The stronger pressure recovery associated with the lower pressure minimum increases the adverse pressure gradient. This, in turn, promotes the formation of a local deceleration zone near the hub, which was reported in a flow visualization study [16]. As a result, the near-wall flow momentum decreases, and the boundary layer rapidly thickens. Consequently, convective heat transfer on the suction side decreases, leading to lower local heat transfer.

In the CFD contours, this low-Nusselt number region on the suction side is observed across larger extent of the span. In areas close to the hub and shroud, the secondary flow influence alters the heat transfer distribution. It is also observed that the entire low heat transfer region resides further upstream toward the leading edge compared to the experiment. This low-Nusselt-number region is consistent with the expected transition region, as derived from intermittency values, with the dashed lines indicating the transition range from $\gamma = 0.1$ to $\gamma = 0.9$. In addition, the CFD contours clearly highlight the passage vortex regions near the hub on both the pressure and suction sides, where the secondary flow leads to reduced heat transfer. The associated aerodynamic losses generated by these structures were analyzed in previous studies [15, 16], where CFD was shown to overpredict the intensity of the secondary flow. Based on these findings, it is reasonable to assume that the low- Nu region predicted in the present work is also overestimated, as the same secondary flow mechanism governs both aerodynamic losses and the local heat transfer reduction in this area. On the pressure side, the spanwise distribution of the Nusselt number generally shows lower values and a more pronounced variation relative to the experiment. The CFD also shows a more pronounced increase in heat transfer near the shroud as a result of the influence of the LPT tip leakage flow.

Figure 7 presents the corresponding Nusselt number profiles at midspan for both pressure and suction surfaces. The data is based on IR-thermography measurements complemented by thermocouple data near the stagnation line (three data points at $x/L \approx 0$). As expected, the largest heat transfer is observed near the stagnation line due to the thin boundary layer and a strong shear induced by the impinging flow. The measured values of the Nusselt number in this region are higher than the CFD predictions, mainly because steady RANS modelling cannot fully capture the intense unsteady velocity and temperature fluctuations that occur near the stagnation region. A similar underprediction was reported by Nikparto and Schobeiri [20], with discrepancy of up to 100% between the experimental and numerical heat transfer coefficients.

Along the pressure side, the Nusselt number shows a gradual downstream decrease followed by an increase starting at $x/L \approx -0.2$. The observed rapid rise in the Nusselt number indicates an onset of the laminar-turbulent transition, which extends to $x/L \approx$

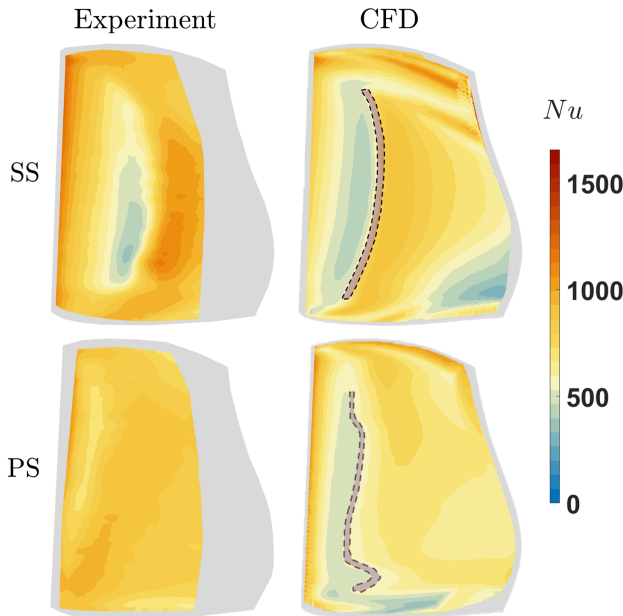


FIGURE 6: Experimental (left) and CFD (right) Nusselt number contours on the OGV suction side (SS) and pressure side (PS) at on-design condition ($\phi = 0.622$, $Re = 350,000$). The dashed lines mark the transition onset and end locations, determined from intermittency thresholds of 0.1 and 0.9.

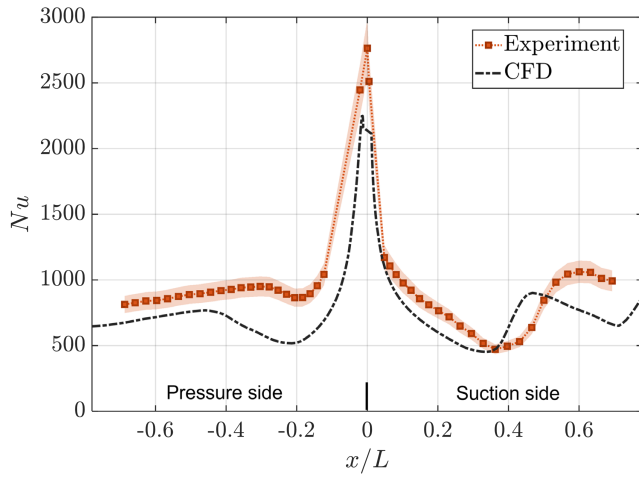


FIGURE 7: Experimental and CFD midspan profiles of the Nusselt number for the OGV at on-design condition ($\phi = 0.622$, $Re = 350,000$). The shaded band indicates experimental uncertainty.

–0.3. This observation relies purely on heat transfer analysis and not supported by flow measurements, since the boundary layers on the pressure side are very thin. However, the very typical shape of the variation and the characteristic upstream shift with the Reynolds number increase are supporting further this interpretation as presented in the following part of this section.

The suction side distribution shows a distinct minimum of the heat transfer coefficient at $x/L \approx 0.36$, which can be explained by the retarded flow region, where the wall shear stress decreases. The laminar-turbulent transition occurs in this region and is induced by the adverse pressure gradient. The current observations based on heat transfer agree very well with previous findings from aerodynamic measurements [16], where the laminar-turbulent transition was found to start at $x/L \approx 0.4$. Furthermore, both observations correlate well with a previous study of a simplified TRS, where IR-thermography was combined with hot-wire measurements of the boundary layer [12, 13].

The CFD results for the suction side show the onset and end of the laminar-turbulent transition slightly upstream of the experiment and predict a lower peak-to-peak variation of the Nusselt number. The minor differences in the location of the laminar-turbulent transition, as well as in the peak-to-peak variation of the Nusselt number can be attributed to the absence of wall-curvature [21] and unsteadiness effects [22] in the transitional SST model. These factors influence the local transition Reynolds number and therefore affect the accuracy of transition prediction on both pressure and suction OGV surfaces.

Influence of operating parameters. Figure 8 presents the experimental Nusselt number contours on the OGV suction (SS) and pressure (PS) sides for three flow coefficient conditions: on-design, zero-swirl, and idle. The case labels follow the notation used in Table 1. The upper row of the figure corresponds to the suction side (SS), while the lower row shows the results for the pressure side (PS).

On the suction side at zero-swirl, the Nusselt number contours show that the low-Nu region shifts downstream and towards the

hub as compared with the on-design case. This can be explained by a severe flow separation on the pressure side imposed by the axial flow, which leads to a blockage of the flow channel and flattening of the streamwise pressure gradient on the suction side. This observation agrees well with the static pressure distributions presented in previous work, where a noticeably flatter pressure profile was observed for the less-loaded OGV compared to the on-design case [16]. At idle flow regime, the very large swirl angle at the inlet causes the flow to impinge on the suction side. The flow stagnates some distance downstream of the leading edge and accelerates towards the leading and trailing edges. The velocity is high along the entire SS, which is the reason for the high heat transfer on this side. The flow detaches from the vane at leading and trailing edges and creates a separation on the vane pressure side.

On the pressure side, shown in the lower row of Figure 8, the zero-swirl case indicates a region of high Nusselt number near the leading edge, followed by a gradual decay downstream, which is caused by a heavily separated flow on the pressure side. At idle condition, the heat transfer on the pressure side is noticeably lower than on the suction side, since the flow is impinging the suction side and the pressure side is dominated by the separated flow. The observed reduction of heat transfer is caused by a low flow momentum on the entire pressure side and the reversal flow which is directed from the trailing to the leading edge near the vane surface. Moreover, the distribution is distinctly three-dimensional, caused by increased magnitude of passage vortex and amplified tip-leakage induced radial flow migration which causes the spanwise variation of the heat transfer.

Figure 9 shows the midspan Nusselt number distributions on the suction (SS) and pressure (PS) surfaces of the OGV for the three investigated flow coefficients: on-design, zero-swirl and idle. On the pressure side, the zero-swirl condition ($\phi = 0.525$) reveals a gradual downstream decay of the Nusselt number. Between $x/L \approx -0.4$ and -0.7 , the Nusselt number values are almost identical to those of the on-design case ($\phi = 0.622$), which may indicate a similar boundary layer development. At idle condition ($\phi = 0.355$), the flow on the pressure side is heavily separated and the flow direction is reversed (from the vane trailing edge to the leading edge). The growth of the heat transfer coefficient between $x/L \approx -0.15$ and the vane trailing edge is attributed to the growth of turbulent boundary from the trailing edge towards the leading edge.

On the suction side, for the zero-swirl flow coefficient ($\phi = 0.525$) the local minimum of the Nusselt number is located at $x/L \approx 0.4$, which is shifted downstream relative to the on-design case ($\phi = 0.622$). In addition, the peak-to-peak variation is noticeably smaller, and the positions of the local minimum and maximum indicate a transition region that extends over a larger portion of the suction surface. At idle flow condition ($\phi = 0.355$), the overall shape of the profile remains similar to that at zero-swirl ($\phi = 0.525$), however, the heat transfer amplifies by a factor of 1.5–2 across most of the suction side. As well, the local maximum of the Nusselt number shifts toward the trailing edge. The very high heat transfer at idle condition causes a very high thermal load of the OGV, which is unfavourable for the durability of the OGV in an aeroengine.

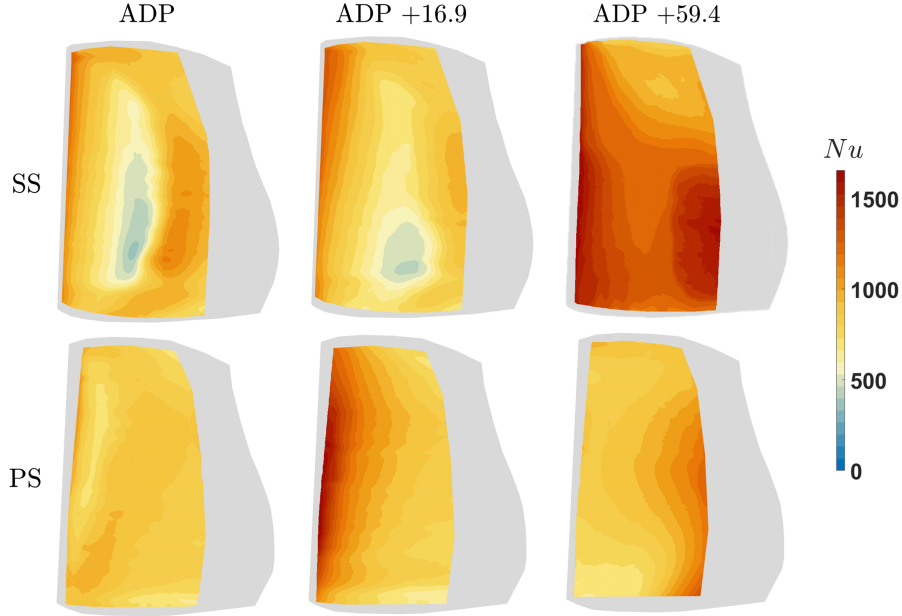


FIGURE 8: Experimental Nusselt number contours on the OGV suction side (SS) and pressure side (PS) at $Re = 350,000$ and different flow coefficients: on-design (ADP, $\phi = 0.622$), zero-swirl (ADP +16.9, $\phi = 0.525$), and idle (ADP +59.4, $\phi = 0.355$).

Figure 10 presents the midspan Nusselt number distributions for the on-design condition and two Reynolds numbers: 350,000 and 465,000. Increasing Reynolds number results in higher heat transfer on both the pressure and suction sides, as expected. The previously identified stages of the laminar-turbulent transition onset and end show a typical upstream shift on both vane sides at increased Reynolds number. These observations confirm the expected Reynolds number dependence of the Nusselt number and indicate that the flow topology at midspan is not greatly affected by the change of Reynolds number between 350,000 and 465,000.

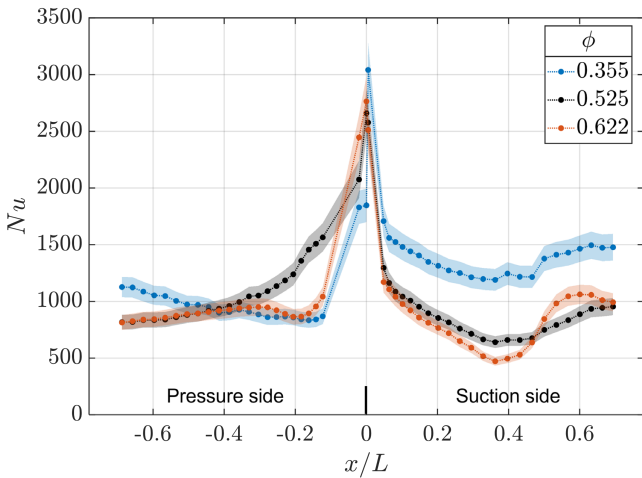


FIGURE 9: Experimental midspan profiles of the Nusselt number for the OGV at $Re = 350,000$ and different flow coefficients: on-design ($\phi = 0.622$), zero-swirl ($\phi = 0.525$), idle ($\phi = 0.355$). The shaded band indicates experimental uncertainty.

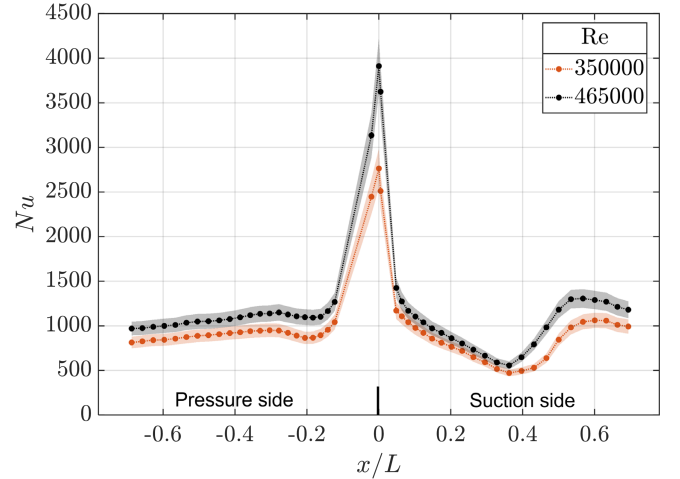


FIGURE 10: Experimental midspan profiles of the Nusselt number for the OGV at two Reynolds numbers: 350,000 and 465,000; $\phi = 0.622$. The shaded band indicates experimental uncertainty.

Hub film cooling effectiveness

The results in this section are presented in terms of the film cooling effectiveness (η), defined as:

$$\eta = \frac{T_{aw} - T_{\infty}}{T_p - T_{\infty}} \quad (3)$$

where T_{aw} is the adiabatic wall temperature, T_{∞} is the temperature of the main TRS airflow, and T_p is the temperature of the purge airflow. The parameter η varies from 0, corresponding to no thermal influence of the purge, to 1, representing a thermal equilibrium between the wall and the purge flow.

On-design condition. Figure 11 presents the experimental and numerical contours of the film cooling effectiveness on the hub endwall at on-design condition. For clarity, a set of three arrows is included in each sub-figure, indicating the flow direction at the TRS inlet for: the purge flow (p), the average main flow (a), and the flow near the LPT hub (h). For the on-design case, the average swirl angle from the LPT is -16.9° , whereas near the LPT hub the swirl angle is -41° . The direction arrows clearly highlight this difference between the bulk flow and the near hub flow.

In the experimental contour plot, a well-defined region of high film effectiveness (blue contours) forms immediately downstream of the purge slot, where the injected flow remains largely unmixed with the TRS flow. The distribution in this region is modulated by a wavy pattern which is imposed by the upstream NGV wakes. As the purge flow film approaches the OGV it interacts with the passage horseshoe vortex developing around the OGV. The passage vortex lifts the purge flow film up which results in reduced cooling effectiveness around the OGV and in the passage between the OGVs. The lowest cooling effectiveness is observed near the vane leading- and trailing edges. The tangential pressure gradient drives the flow toward the suction side, resulting in a higher film effectiveness in the passage towards the vane suction side (yellow-orange contours).

The CFD contour plot indicates that the film stays near the hub to a larger extent than in experiments and mixes less with the mainstream, leading to an optimistically high prediction of the film cooling effectiveness in the passage towards the suction side down to the trailing edge (blue contours). This is caused by under-prediction of the flow mixing in the CFD. However, the CFD is conservative in the prediction of the extent of the low-effectiveness region around the vane leading edge and around the vane suction side (red contours). On the pressure side, the CFD captures the magnitude of the film effectiveness very well, although the contours do not fully reproduce the flow non-uniformity observed in the experiment.

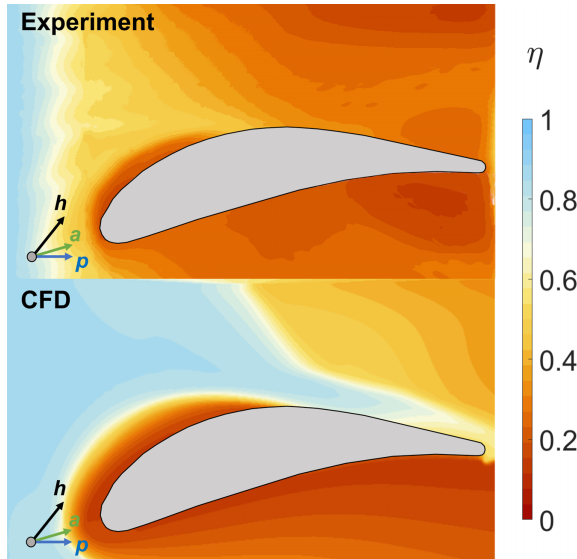


FIGURE 11: Adiabatic film cooling effectiveness on the hub endwall at on-design condition ($\phi = 0.622$, $Re = 350,000$).

Idle condition. Figure 12 presents the experimental and numerical contour plots of the film cooling effectiveness on the hub endwall at off-design idle condition. From the experimental contour plot, the film cooling effectiveness remains nearly uniform in the tangential direction over a large portion of the passage upstream of the OGV (blue and yellow contours). As described above, the flow at idle condition impinges vane suction side. The interaction of the main flow and the purge flow preserves the relatively high cooling effectiveness in the passage (orange contours), while the horseshoe vortex lifts up the film near the vane (red contours near the suction side). Since at idle condition, the flow impinges the suction side of the vane, the pressure on the suction side is higher than on the pressure side. Consequently, the tangential pressure gradient is reversed relative to the on-design condition and is directed from the suction side toward the pressure side. As a result, the mixed flow is deflected toward the pressure side and modifies the cooling effectiveness distribution across the passage up to $x/L \approx 0.5$. The cooling film does not propagate farther downstream since the purge flow becomes fully mixed with the mainstream, and its cooling contribution becomes small.

In the vicinity of the OGV, the measured effectiveness remains low along both sides of the vane (red contours). This suction side region is strongly influenced by the horseshoe vortex which governs the near-wall coolant transport. The vortex is wrapping around the suction side and remains closely attached to the OGV, sweeping the coolant upward and away from the endwall which leads to locally weaker film coverage. In contrast, on the pressure side the coolant is pushed away by the strong flow separation on the pressure side.

The CFD prediction shows a larger extent of the low-effectiveness region on the suction side, which means a stronger amplitude of the horseshoe vortex and a conservative prediction. A higher film effectiveness is observed on the pressure side, which means less severe flow separation and an optimistic cooling efficiency prediction.

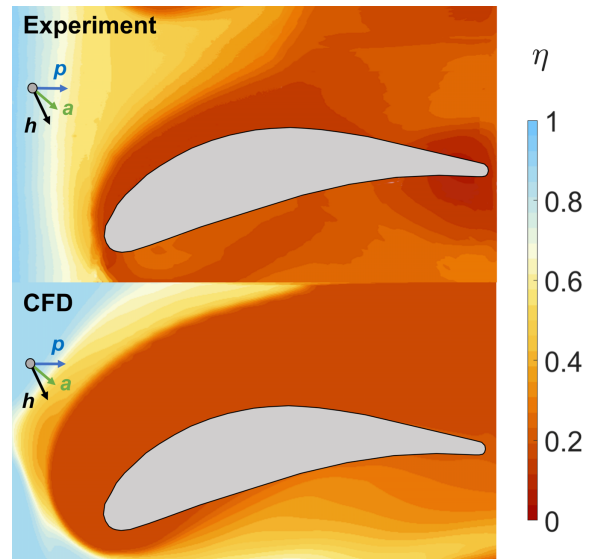


FIGURE 12: Adiabatic film cooling effectiveness on the hub endwall at idle condition ($\phi = 0.355$, $Re = 100,000$).

5. SUMMARY AND CONCLUSIONS

This study presents the first comprehensive thermal investigation of a state-of-the-art turbine rear structure. Measurements were performed for Reynolds numbers and flow coefficients representative of modern turbofan engines. For the first time, in a fully coupled LPT–TRS environment, the heat transfer distributions on an OGV with realistic geometry and the hub film cooling effectiveness downstream of a purge slot were obtained.

At on-design condition, the heat transfer distribution on OGV sides is greatly affected by the laminar–turbulent transition, the spanwise variation of aerodynamic loading, the LPT tip leakage, and the passage vortex. The boundary layers on both OGV sides undergo laminar–turbulent transition, which results in a characteristic heat transfer variation. Moreover, the higher loading toward the hub causes the formation of the passage vortex and reduced heat transfer in the vane–hub region. This results in a non-uniform spanwise heat transfer distribution near the hub. On the pressure side, the heat transfer distribution is more uniform compared to the suction side. The CFD captures the heat transfer distribution well, however predicts an earlier onset and end of the laminar–turbulent transition on vane suction side while a later onset and end on vane pressure side. From a design perspective, these results show that accurate modeling of laminar–turbulent transition is a prerequisite for obtaining realistic OGV heat transfer distributions. In addition, the used numerical method predicts lower overall heat transfer over the vane including leading edge and a stronger amplitude of the passage vortex. These discrepancies demonstrate that reliable aerothermal predictions in a TRS require reproduction of both the global aerodynamic field and the near-wall region.

At zero-swirl off-design condition, which is representative for the sea-level take-off, the heat transfer on the vane suction side is changed moderately while on the pressure side reaches significantly larger values than at on-design which results in an elevated thermal load of the OGV. Midspan data show that the transition onset on the vane suction side moves downstream, while the peak-to-peak variation in the heat transfer is lower compared to the on-design case.

Under idle off-design condition, the changes of the OGV heat transfer are more pronounced: the heat transfer on the suction side is elevated by a factor of 1.5–2, while a severe flow separation on the pressure side leads to moderately elevated heat transfer. These results highlight the strong sensitivity of the heat transfer distribution in the TRS to inlet swirl variations. In comparison, the effect of the Reynolds number is more limited and leads to expected heat transfer variation. Therefore, the results show that severe off-design conditions should be considered as critical cases in design practice for heat transfer assessment.

The endwall film effectiveness measurements provide the first experimental dataset obtained in an engine-representative TRS. It was found that the resulting film effectiveness distribution in the inter-OGV region is dominated by the tangential pressure gradient across the passage and by the horseshoe vortex developing in the vicinity of the OGV.

Under the on-design condition, the film remains partially mixed near the endwall over a large part of the inter-OGV passage, leading to a relatively high film effectiveness. This demonstrates that

purge flow cooling performs well when the TRS operates close to its intended swirl range. Under the off-design idle condition, however, the inlet swirl angle deviates strongly from the design value. The horseshoe vortex develops around a large part of the suction side, leading to a substantially reduced film effectiveness on the hub around the vane’s suction side. Consequently, the film cooling becomes less effective under this severe off-design condition.

In the immediate vicinity of the OGV, the film cooling is dominated by the horseshoe vortex, which lifts the coolant away from the surface. This results in consistently low effectiveness both in the leading edge region and further downstream. For the off-design idle case, this low-effectiveness zone becomes more pronounced, with the horseshoe vortex displacing an even larger portion of the film into the inter-OGV passage. As a result, the hub-OGV region is poorly cooled, especially near the leading edge, where the strong horseshoe vortex prevents the film from reaching the OGV surface.

Although the CFD reproduces most of the film cooling pattern very well and in large extent of the passage agrees with the measured values, in some critical regions the CFD predicts a too low film effectiveness, which means a conservative prediction, and in some regions a too high film effectiveness, which means an optimistic prediction. This limitation highlights a modelling gap and indicates that measures are required, as unbalanced predictions can result in insufficient local film cooling. Consequently, this can lead to elevated thermal loads in critical regions such as the leading edge, where material degradation may occur. Therefore, the experimental dataset obtained in this work provides a high-fidelity benchmark for advancing the predictive capability of TRS thermal loads using CFD and for enabling a more robust thermal design of the TRS.

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