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DRAFT: INVESTIGATION OF ISOTHERMAL CONVECTIVE HEAT TRANSFER IN AN OPTICAL COMBUSTOR WITH A LOW-EMISSIONS SWIRL FUEL NOZZLE

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ABSTRACT

Modern combustor design optimization is contingent on the accurate characterization of the combustor flame side heat loads. The present work focuses on the experimental measurement of the isothermal (non-reacting) convective heat transfer along a fused silica optical can combustor liner for Reynolds numbers ranging between 11 500 and 138 000. The model combustor was equipped with the SoLoNOx swirl fuel nozzle from Solar Turbines Incorporated, subjecting the liner walls to realistic isothermal flow and turbulence fields. Infrared (IR) imaging through fused silica was demonstrated, and a novel estimation of the three-dimensional conduction heat losses for steady state constant heat flux experiments was developed. A maximum heat transfer augmentation of ~18 was observed with respect to fully developed turbulent pipe flow correlations. Contrary to other investigations, the augmentation magnitude and distribution are shown to be approximately constant with Reynolds number (particularly away from the impingement location). Particle Image Velocimetry (PIV) was included to support the heat transfer measurements, suggesting that peak heat transfer occurred 0.12 nozzle diameters upstream of the jet reattachment point along the liner. Reynolds-Averaged Navier Stokes (RANS) computations are shown to yield peak heat transfer predictions within 17.4% of the experimental results when using the realizable $k-\epsilon$ turbulence model and enhanced wall treatment. The measurements were further analyzed in the context of results from other heat transfer studies on gas turbine combustors.

KEYWORDS:

Combustor, heat transfer, swirl flow, infrared thermography, conduction correction

NOMENCLATURE

A	Area [m ²]
a	Absorptance
B_λ	Spectral radiance from Planck's Law [W m ⁻³ sr ⁻¹]

24	c	Speed of light [$299\,792\,458\text{ m s}^{-1}$]
25	D	Diameter [m]
26	f	Frequency [Hz]
27	h	Heat transfer coefficient [$\text{W m}^{-2}\text{ K}^{-1}$]
28	h_p	Planck's constant [$6.62607004 \times 10^{-34}\text{ m}^2\text{ kg / s}$]
29	K	Absorption coefficient
30	k	Conductivity
31	K_B	Boltzmann constant [$1.38064852 \times 10^{-23}\text{ m}^2\text{ kg s}^{-2}\text{ K}^{-1}$]
32	L	Thickness of the quartz liner [4 mm]/reference length or direction [m]
33	Nu	Nusselt number
34	P	Power [W]
35	P_4	Pressure at the combustor outlet
36	P_3	Pressure at the combustor inlet
37	Pr	Prandtl number
38	Q	Heat rate [W]
39	Q''	Heat flux [W m^{-2}]
40	R	Radiance [$\text{W m}^{-2}\text{ sr}^{-1}$]
41	R_Ω	Resistance [Ω]
42	Re	Reynolds number
43	T	Temperature
44	t	Transmittance
45	V	Voltage [Volts]
46	α	Thermal diffusivity [$\text{m}^2\text{ s}^{-1}$]
47	ϵ	Emissivity
48	λ	Wavelength [m]
49	μ	Dynamic viscosity [$\text{kg m}^{-1}\text{ s}^{-1}$]

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51

52 Subscripts

53	C	Evaluated with respect to the combustor dimension
54	F	Evaluated at the film temperature
55	IR	Refers to the infrared camera
56	$LOSS$	Refers to thermal conduction heat losses
57	N	Evaluated with respect to the nozzle dimension
58	nat	Refers to natural convection
59	Q	Refers to the quartz liner
60	R	Radial direction (with respect to the combustor axis)

61	W	Evaluated at the wall or at the wall temperature
62	X	Axial direction (with respect to the combustor)
63	Y	Vertical direction (for a 2D image)
64	λ	Spectral
65	∞	Evaluated at the flow bulk temperature

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67

68 1. INTRODUCTION

69 The continued development of gas turbine engines for improved efficiency and performance requires operating at
70 higher turbine inlet temperatures, while simultaneously avoiding any increase in pollutant emissions or failure of the
71 components exposed to the hot gas path. To improve efficiency, modern gas turbines operate at increasingly higher
72 compression ratios, thus increasing the air temperature at the inlet to the combustion system and reducing its cooling
73 potential. To limit the production of NO_x , combustors burn at lean equivalence ratios, which in turn demands excess
74 air in the fuel-air mixture and further decreases the amount of air available for cooling. Modern combustor liners
75 moreover limit the use of film cooling, to prevent quenching of the intermediate combustion products and the
76 subsequent formation of unburned hydrocarbons and carbon monoxide near the walls. For these reasons, thermal
77 management has become a critical aspect during the design phase of gas turbine combustors. In order to optimize
78 coolant utilization, detailed heat transfer data for gas turbine combustion chambers is required. In particular,
79 information on the convective heat loads is limited, due to the complex flows generated by industrial swirl fuel nozzles
80 and the challenging conditions at which combustors operate.

81 The first attempts to semi-empirically characterize the heat transfer along the combustor liner walls were
82 carried out by Lefebvre and coworkers [1, 2]. The radiative component of the thermal load is relatively well
83 understood [3,4]. For internal convection, Lefebvre and Ballal [1] pointed out that the "uncertainties regarding the
84 airflow pattern, the state of the boundary layer development, and the effective gas temperature make the choice of a
85 realistic model almost arbitrary" [1]. The authors estimated the convective heat transfer according to early correlations
86 for fully developed turbulent pipe flow, arguing that a similar formulation should apply within combustor flows when
87 no film cooling is used. Lefebvre and Ballal [1] suggested Eq. 1 to estimate the convective heat fluxes within
88 combustors.

89

$$Q_c'' = 0.02 \text{ Re}_{C,\infty}^{0.8} \frac{k_\infty}{D_c} (T_\infty - T_w) \quad (1)$$

90

91 The authors further proposed a reduction of the 0.02 coefficient to 0.017 in the primary zone to account for
92 reduced near wall gas temperatures. Equation 1 is approximately equivalent to evaluating the Dittus-Boelter heat
93 transfer correlation, given by Eq. 2, with $\text{Pr} \approx 0.706$ (air). The gas properties for the Dittus-Boelter equation are
94 typically evaluated at the local film temperature and the exponent n_T is equal to 0.4 for $T_w > T_\infty$ and 0.3 for $T_w < T_\infty$.

95

$$\text{Nu}_{\text{Dittus-Boelter}} = 0.023 \text{Re}_{c,f}^{0.8} \text{Pr}_f^{n_T} \quad (2)$$

The Dittus-Boelter equation is accurate within $10\,000 < Re < 120\,000$, applicable only for wall to gas temperature differences of no more than 56 K, and cannot properly account for the temperature dependence of the gas properties near the wall [5,6]. Since the time of the work by Lefebvre, there have been significant improvements to turbulent flow heat transfer correlations [7-10] and a more comprehensive understanding of the flow within combustors [11-13]. Work on heat transfer enhancement in swirling flows and sudden expansions has also provided insight that can be applied to understanding convective loads within modern gas turbine burners. Dellenback et al. [14] for instance, worked on the experimental characterization of the heat transfer downstream of a sudden pipe expansion with swirling flow. While their swirl generator was not representative of combustor fuel injectors, it provided important details on the effects of swirl and Reynolds number ($Re_c < 51\,000$). The authors showed that increasing the swirl number resulted in larger heat transfer to the wall, and that the location of peak heat transfer consistently occurred upstream of the location of flow reattachment. Similar heat transfer investigations on turbulent decaying swirl flow have also been conducted without the sudden expansion to study the effects of swirl number and type of swirl generator [15]. Work on heat transfer in an abrupt expansion without swirling flow, such as that conducted by Baughn et al. [16], is also partly relevant to combustor flows as it exhibits flow reattachment and the formation of a corner recirculation region.

While combustor cooling technologies have been studied extensively in the open literature [17,18], information on combustor heat loads is lacking. Combustor convective heat transfer has been studied by Ekkad and coworkers for both can [19,20,21] and annular [22,23] combustor geometries, using approximated swirlers representative of industrial fuel nozzles. The work of these authors included isothermal convective heat transfer measurements supplemented by numerical simulations and flow characterization using particle image velocimetry (PIV). The work by Patil et al. [22] reported Nusselt numbers several times higher than those obtained from the Dittus-Boelter equation, suggesting that the correlation underestimates the convective heat loads within burners. Recent work by Andreini et al. [24] and Lorenzo [25] have also experimentally and computationally studied the isothermal convective heat transfer and flow within combustors using a three swirl nozzle combustor model. Their results include the effects of neighboring nozzles and of the cooling film at the start of the liner. Detailed PIV by the authors provided insight into the interaction between nozzles and the relationship between flow and convective heat transfer.

In the present investigation, the isothermal (non-reacting) heat transfer along an optical can combustor equipped with an industrial fuel nozzle (SoLoNOx nozzle by Solar Turbines Incorporated) was studied over a broad range of Reynolds numbers at atmospheric pressure. Previous investigations to date have relied on approximate nozzle models fabricated via additive manufacturing. Using a fully functional industrial fuel nozzle yields realistic velocity and turbulence fields within the research combustor. Moreover, the test setup used in the current study is capable of operation at reacting conditions, which will be the focus of further research efforts. Performing non-reacting steady state heat transfer measurements in a fused silica (quartz) optical burner required adaptations to traditional heat transfer techniques to account for the relatively high thermal conductivity and limited infrared (IR) transmissivity of the liner. These modifications have the potential to expand the applicability of IR thermography to experiments relying on fused silica for optical access and experiments where larger conduction heat losses are unavoidable.

Measurements were taken for Reynolds numbers between 11 500 to 138 000 with respect to the combustor inner diameter (Re_C). These values correspond to combustor static pressure drops ($1 - P_4/P_3$) between 0.61% and 30.42%. The lower mass flow cases reproduce typical combustor pressure drops ($< 3\%$), while the higher mass flow cases reproduce typical combustor Reynolds numbers ($Re_C > 100\,000$). The experimental results were supplemented with Particle Image Velocimetry (PIV) measurements to relate the flow field to the heat transfer distribution observed. Industry often relies on computational fluid dynamics (CFD) for the analysis of heat transfer within their combustors, supplemented with thermocouple and thermal paint measurements [26, 27]. For this reason, Reynolds-Averaged Navier Stokes (RANS) calculations using the k- ϵ turbulence model were also carried out to evaluate the performance of commercial codes in predicting the heat transfer characteristics. Emphasis however was placed on the experimental heat transfer data and procedures.

2. EXPERIMENTAL PROCEDURE

The experiments were conducted at the Advanced Propulsion and Power Laboratory (APPL) at Virginia Tech. A portion of the experimental facility is shown in Figure 1. Upstream of the main control valve there was a flow metering section and a pressure regulator (not shown). The flow metering was done using a turbine flow meter, coupled with downstream pressure and temperature measurements, yielding a total accuracy in the mass flow of $\sim 2\%$. The pressure reducing regulator lowered the pressure to the required operating conditions and the main control valve throttled the flow to the desired mass flow rate. An inline 192 kW heater was also installed but was not used for the current experimental results.

Figure 1 Diagram of the experimental facility.

The test section houses an industrial low emission swirl fuel nozzle provided by Solar Turbines Incorporated (SoLoNOx). The nozzle consists of an axial swirler, followed by fuel injectors downstream that introduce fuel into the nozzle annular channel for premixing. Since the experiments and computations for the current study were conducted at isothermal (non-reacting) conditions, no fuel was required. The liner consisted of a quartz tube 4 mm thick, with an internal diameter of 203 mm and a length of 216 mm. The test section and coordinate system used are shown in Figure 2. The coordinate system was defined at the axis of the combustor but is shown displaced in the figure for clarity. The heat transfer characteristics were observed to be axisymmetric and hence only a small portion of the liner was used for the measurement of the convective heat transfer. The area used for the heat transfer measurement was plane A as

highlighted in Figure 2. To support the heat transfer observations, PIV measurements were taken along Plane (B), also shown in Figure 2; capturing the swirling jet expansion into the primary zone and reattachment on the liner wall.

Figure 2 Test section. The coordinate system was defined at the axis of the combustor but is shown displaced for clarity. Measurement planes: (A) Heat transfer with IR thermography, (B) flow field with PIV.

2.1 Steady State Heat Transfer through the Optical Quartz Liner

To measure the heat transfer along the liner wall, a thin surface heater (0.3 mm thick Polyimide Thermofoil™ heater from Minco, 5 cm × 28 cm) was attached to the inner wall of the quartz liner providing a constant and uniform wall heat input (Q_w). The wall heat input was determined from the electrical resistance of the heater and the voltage supplied ($P = Q_w = V^2/R_\Omega$). The steady state (60 min) inner wall temperatures (T_w) were derived from IR measurements using a FLIR SC6700 camera, with a detector spectral range of 1 to 5 μm .

Figure 3. Top view schematic of the experimental setup for the heat transfer measurements.

A schematic of the heater placement on the liner is shown in Figure 3. The heater was covered with aluminum tape (~0.08 mm thick) to uniformly distribute the heat generated and coated with a high emissivity paint (~0.05 mm coating thickness) to increase the IR emission from the inner wall. The IR camera was positioned on the acquisition side, past the liner wall opposite to where the heater was placed. The inner wall thermal radiation from the heater was partially transmitted through the quartz glass, reaching the IR detector along with the emission of the quartz liner itself. An important challenge of the measurement was to ensure that the quartz emission would not overwhelm the signal received by the IR detector.

During the experiment, air at ambient conditions was supplied to the test setup, and the heater power was adjusted to achieve the maximum allowable temperature at the given air flow. The heat transfer coefficient (HTC), h , along the combustor liner wall was determined from Eq. 3.

$$h = \frac{Q''_W - Q''_{LOSS}}{(T_W - T_\infty)} \quad (3)$$

Where the wall temperature T_W was calculated from the IR acquisitions, Q''_W was the heat flux applied to the surface, T_∞ was the air flow temperature, and the heat flux lost to conduction Q''_{LOSS} was obtained from experiments detailed in section 2.3. T_∞ was estimated from the average of two thermocouples, one immediately at the outlet of the fuel nozzle and another at the outlet of the combustor.

2.2 Calibration of the inner wall emission

It was first critical to theoretically demonstrate that the IR camera was capable of measuring the inner wall emission reliably, without being overwhelmed by the quartz liner emission. Detailed absorption coefficient data for fused silica is given by Dvurechenskii et al. [28] and Loenen and Van der Tempel [29], which can be used to theoretically approximate the contribution from the inner wall to the overall acquired IR emission. The emissivity of the black coating (ϵ_W) was estimated as 0.88 at all wavelengths; this was the lowest emissivity obtained from an independent calibration of the coating over the range of experimental temperatures. The spectral emissivity ($\epsilon_{\lambda,Q}$) and transmissivity ($t_{\lambda,Q}$) of the quartz liner were calculated from Lambert's and Kirckhoff's laws (Equations 4 and 5 respectively), neglecting the reflectivity of the quartz.

$$1 - a_{\lambda,Q} = t_{\lambda,Q} = \exp(-K_{\lambda,Q}L) \quad (4)$$

$$a_{\lambda,Q} = \epsilon_{\lambda,Q} \quad (5)$$

The spectral absorption coefficient K_λ was taken from [28] for their KI glass at 1073 K (absorption increases with temperature, hence this represents a worst case scenario), and L was equal to the quartz liner thickness (4 mm). The total emission from the inner wall was calculated by integrating Planck's Law (B_λ , Eq. 6) of spectral radiance over the IR detector spectral range as given by Eq. 7, accounting for the optical properties of the wall and liner.

$$B_\lambda(\lambda, T) = \frac{2hpc^2}{\lambda^5} \frac{1}{\exp\left(\frac{hpc}{\lambda KB T}\right) - 1} \quad (6)$$

$$R_T = R_i + R_Q = \int_{1\mu m}^{5\mu m} t_{\lambda,Q} \epsilon_W B_\lambda(\lambda, T_W) d\lambda + \epsilon_{\lambda,Q} B_\lambda(\lambda, T_Q) d\lambda \quad (7)$$

Figure 4. Estimated ratio of transmitted to total emission received by the IR detector, to validate that measurements of the inner wall are possible through quartz glass.

The result of the ratio of transmitted emission (R_i) to total radiance received by the detector (R_T) for different quartz and inner wall temperatures are shown in Figure 4. During the experiment, the lowest wall temperature (around the point where the swirling jets impinge on the liner) was ~ 360 K, while the quartz temperature was approximately equal to the combustor flow temperature T_∞ and maintained between 283 - 294 K for all the cases tested (environment temperature). This implies that, per the theoretical analysis, the transmitted inner wall emission contributed at least 50% to the total radiance received by the IR camera. The maximum temperatures observed along the liner were ~ 420 K, corresponding to $> 80\%$ of the acquired signal due to the transmitted emission from the inner wall. As shown by Figure 4, the higher the temperature of the quartz liner, the smaller the relative signal from the inner surface heater.

Figure 5 Calibration of the IR reading to the thermocouple reading of the inner wall temperature.

To calibrate the effective emission from the heated inner wall surface, the IR readings through the quartz liner were compared against in-situ T-type thermocouple measurements (errors of ± 0.5 K). The results, shown in Figure 5, were

fitted with a 6th order polynomial to obtain the calibration curve for the inner wall surface temperature in terms of the IR temperature reading. The total uncertainty band (three standard errors) in the fitted curve for calibration was ± 9.54 K. This overall calibration reduced the error in the measurement, compared to the error that would be incurred by separately measuring the transmissivity of the quartz liner and the emissivity of the black coating. Note that the experimental results are independent of the estimated emissivity assumed throughout the theoretical analysis.

2.3 Three dimensional heat loss characterization

The heat lost to the quartz liner and surroundings (Q_{LOSS}) was measured by applying a heat flux to the wall while there was no flow through the combustor, and acquiring the wall temperature at steady state using the IR camera. Several heat loss experiments at different heat inputs were carried out to obtain a loss curve that relates the heat lost, to the temperature difference between the surface of the heater and the ambient air (the heat sink temperature). Investigators [19,22,23] typically place the heater on a thermally insulating surface (foams/cork with $k \approx 0.04 \text{ W m}^{-1} \text{ K}^{-1}$) to minimize the heat loss and any three dimensional conduction effects. This is unfortunately not applicable for the quartz liner, as its thermal conductivity equals $\sim 1.4 \text{ W m}^{-1} \text{ K}^{-1}$ [30] at ambient conditions. The energy balance for a heat loss (no flow) experiment is given by Equation 8, where the supplied heat is lost to natural convection within the combustor, radiation, and conduction losses in the radial, azimuthal, and axial directions ($Q_{LOSS,R}$, $Q_{LOSS,\theta}$, $Q_{LOSS,X}$ respectively). The radiative loss was accounted for within the radial conduction loss term (the temperature potential driving both mechanisms is approximately the same) and was hence not explicitly calculated.

$$Q_{W,no\ flow} = Q_{LOSS,R} + Q_{LOSS,X} + Q_{LOSS,\theta} + h_{nat}A(T_W - T_{\infty}) \quad (8)$$

For the heat loss calculations, the azimuthal component was approximated as a vertical component to simplify the analysis, i.e. $Q_{LOSS,\theta} \approx Q_{LOSS,Y}$. The natural convection HTC (h_{nat}) within the combustor was estimated according to the correlation for free convection from a vertical surface as given by Equation 9 [31].

$$\overline{Nu}_L = 0.68 + \frac{0.67 Ra_L^{1/4}}{(1 + (0.492/Pr)^{9/16})^{4/9}} \quad (9)$$

Where Ra_L is the Rayleigh number ($L = 51 \text{ mm}$) with properties evaluated at the corresponding film temperature. According to the calculations $h_{nat} \approx 8.4 \text{ W m}^{-2} \text{ K}^{-1}$, and a conservative uncertainty of 20% was considered while calculating the total uncertainty of the heat loss (the experimental data used by [31] deviated from their correlation by $\sim 12\%$ on average).

An example of the heater temperature distribution after a steady state heat loss measurement is shown in Figure 6. Close to the center of the heater a temperature maximum was observed. Since $Q_{W,no\ flow}$ was approximately uniform, this was indicative of significant conduction in the X and Y directions. The point of maximum temperature during any heat loss experiment corresponds to zero conduction heat losses along the azimuthal and axial direction ($Q_{LOSS,\theta} = Q_{LOSS,X} = 0$); there is only conduction into the quartz liner ($Q_{LOSS,R}$, which also accounts for radiative losses) and

natural convection occurring. Therefore, at the area of maximum measured temperatures along the heater $Q_{LOSS,R}$ can be solved for, from the wall temperature data (T_w from the IR camera), the input heat flux ($Q_{W,no\ flow}$), and the surrounding heat sink temperature (T_∞ from thermocouple readings). Figure 7 shows the measured $Q_{LOSS,R}$ curve obtained from the maximum temperature area for the six heat loss measurements taken for this study. The data was interpolated to obtain the radial heat loss at other wall temperatures.

Figure 6. Temperature distribution of the heater surface during a no-flow experiment to calculate the conduction heat losses.

Figure 7. Radial heat flux lost due to conduction.

At any other point on the heater surface during the no-flow experiments, the remainder of the heat supplied after removing the convective loss and $Q_{LOSS,R}$ is due to the azimuthal and axial heat losses. To estimate these components, the locations along the heater where the axial and azimuthal temperature gradients equaled zero were identified, corresponding to a zero heat loss in the respective direction. The zero azimuthal temperature gradient corresponded to the point of maximum temperature at each axial location. This can be similarly done for the azimuthal component of the heat loss.

For the case of the radial heat loss, the difference between the ambient and heater surface temperatures defined the temperature potential driving the heat transfer. This however was not the case for the axial and azimuthal heat loss components as these components depended on the local temperature gradient in those directions. The noise in the raw experimental data complicated the extraction of this temperature gradient. For this reason, the maximum temperature profiles along X and Y (where the alternate heat loss component was zero) were fitted with a 6th order polynomial from which the temperature gradient was determined. Figure 8 shows the resulting heat loss in the axial and azimuthal directions as a function of $T_w dT_w/dL$ (where L is either Y or X for the azimuthal and axial heat loss components respectively) from the different heat loss experiments. The discrepancies between cases and directions are due to non-linear effects (conductivity changes with temperature), errors in the temperature gradient, and non-uniformities in the heat flux applied.

Figure 8. Axial and azimuthal heat flux lost due to conduction.

Using the calculated heat loss characterization for the pixels that were subjected to all conduction heat loss components resulted in less than 2% unaccounted heat flux. While the errors in the axial component of the heat loss are large, its contribution to the total heat loss is one order of magnitude smaller compared to the radial heat loss.

2.4 Supporting Flow Measurements using PIV

Emphasis was placed on the heat transfer measurements, and the PIV data are included to relate the observed heat transfer features to the flow field. Two-Dimensional planar Particle Image Velocimetry (PIV) was taken for $Re = 11\,500$, $23\,000$, and $41\,400$ along plane B (Figure 2). These measurements support the heat transfer results and provide details on the size of the corner recirculation and approximate location of flow reattachment. Further information on the steady and unsteady flow field within the combustor model at isothermal conditions can be found in Gomez-Ramirez et al. [32]. For the measurements, a Nano-L 135-15 Nd:YAG dual laser system with a Flowsense 4M MKII camera and control software from Dantec Dynamics were used. A total of 800 image pairs were acquired in the plane and an adaptive correlation algorithm was used to extract the velocity vectors. The initial and final interrogation areas were 128×128 pixels and 32×32 pixels respectively with a 50% overlap. Median and peak

validations were used to correct unphysical vectors in the dataset. Only velocity fields (PIV snapshots) with less than 30% corrected vectors were retained, to prevent biasing the results to the median validation.

2.5 Experimental Uncertainty

The overall uncertainty in the heat transfer experiment was attributed to errors in T_W , T_∞ , Q''_W , and Q''_{loss} . Errors in the inner wall temperature (T_W) were 9.62 K, accounting for the calibration error (three standard deviations) and the thermocouple accuracy of ± 0.5 K. The total uncertainty for the combustor flow temperature (T_∞) was 1.92 K, including the average difference between the combustor inlet and outlet temperature and the accuracy of the thermocouples. The wall heat flux error was calculated from errors in the heater electrical resistance of $\pm 0.3 \Omega$ and errors in the voltage supplied of ± 0.5 V, this error was also considered during the heat loss calculation. The heat loss errors were estimated according to the errors shown in Figure 7 and Figure 8. The total relative error for the HTC was on average between 5.1% and 10.8% depending on the mass flow case. The majority of the error was due to the inner wall calibration uncertainty. Errors in Q''_{loss} and Q''_W combined accounted for at most 3.2% uncertainty in the HTC.

Experiments were tested for repeatability at different heater settings (lower temperature) with differences smaller than 2%. It was also verified that the steady state condition was reached after 60 minutes by comparing results for data acquisitions after 60, 120, 180, and 240 minutes; the results were on average within 1.3% of the 60 minutes result.

For the PIV dataset, the correlation algorithm yielded an estimated subpixel accuracy of 0.1 pixels, which resulted in an error between $0.02V_{ref}$ for the plane imaged. To further verify consistency, the time-averaged axial velocity profile at $X/D_N = 0.05$ was used to estimate the airflow delivered to the test section (assuming axisymmetric flow) with an average discrepancy with the metered mass flow of 5.6%.

3. COMPUTATIONAL SETUP AND METHODOLOGY

The main purpose for presenting the numerical calculations was to verify that Reynolds-Averaged Navier Stokes (RANS) calculations, as are typically used in industry, yielded consistent results compared to the experiment. A detailed analysis of the computational results, including the impact of the turbulence model used and modeling strategy, will be presented elsewhere. In order to optimize the computational time and the meshing strategy, the computations were performed in two steps. The first step consisted of obtaining the flow profiles at a section downstream of the swirler by simulating flow in the entire swirler-combustor domain. The second step was to use these flow profiles as inlet boundary conditions for a 90 degree combustor sector domain (without the swirler). The experiments predicted near axisymmetric time-averaged flow and steady state heat transfer features; hence simulating the steady flow within the entire three-dimensional combustor domain was considered computationally redundant. Using a sector model with pre-calculated inlet boundary conditions, instead of solving the flow through the swirler in each simulation, favored the optimization of the mesh, improved convergence, and reduced computation time. Patil and Tafti [33] have successfully applied a similar computational strategy. The hexahedral mesh of the combustor sector domain and swirler are shown in Figure 9 (left and right respectively).

Figure 9. Computational meshes used. Left: combustor sector domain. Right: Swirler domain.

3.1 Modeling method

The realizable k - ϵ turbulence model [34] was used for the analysis of the combustor flow for Reynolds numbers up to 73 000 with respect to the combustor diameter. The RNG k - ϵ turbulence model was considered initially as suggested by Patil et al. [19, 22] for swirling flows; however the results using the realizable k - ϵ model showed better convergence and resulted in more realistic velocity and heat transfer distributions. A 3.7 million element grid was used to numerically simulate the combustor sector domain for all Reynolds numbers. The near wall grid was defined to yield a Y^+ smaller than one, in accordance with the requirements of the wall treatment approach used. Grid independence at all Reynolds numbers was also verified. Second order spatial discretization schemes were used for momentum and energy, in conjunction with pressure-velocity coupled equations. The inlet boundary to the sector domain was applied based on the velocity and turbulent kinetic energy profiles obtained from the separate simulation performed for the full swirler-combustor domain. The outlet boundary condition was maintained at atmospheric pressure with backflow turbulence defined by 75% turbulence intensity (from experimental observations, see [32]) with a length scale equal to the diameter of the central recirculation. The liner wall was modeled as applying a constant heat flux of 2500 W/m^2 to the working fluid. The opposite faces of the sector were modeled as rotationally periodic boundaries. In addition to the residuals falling below at least 10^{-5} , the convergence was monitored by ensuring a constant value of mass averaged axial velocity at a combustor cross section capturing the jet expansion and central recirculation for at least 1000 iterations.

3.2 Modeling uncertainty

Three different grids for the highest and lowest Reynolds number cases were used to ensure grid independence for the computational investigation performed in the current study. The grid independence study of the HTC result for the highest Reynolds number case is shown in Figure 10. The overlap in the heat transfer predictions between the medium size grid and the fine grid was on average 2.5% for the low and high Reynolds number cases. All other Reynolds numbers cases were evaluated with at least the medium resolution grid used at the highest Reynolds number.

The Grid Convergence Index (GCI) [35] was used to estimate the discretization error in the calculations (obtained for the lowest and highest Reynolds number cases). In addition to this, the numerical uncertainty was also

estimated empirically by calculating the azimuthal (circumferential) variability. Since experiments predicted an axisymmetric heat transfer distribution, the circumferential deviation in HTC was chosen as an empirical metric of the uncertainty. The axially averaged GCI obtained for the $Re = 11\,500$ and $73\,000$ cases was respectively 1.2% and 4.4%. The maximum GCI was observed in the corner recirculation with values $< 13\%$. Similarly, the average circumferential variability along the liner was 0.4% and 4.4% for the lowest and highest Reynolds number results respectively.

Figure 10. Mesh independence study for the 73 000 Reynolds number case.

4. RESULTS AND DISCUSSION

The heat transfer data are presented in terms of the HTC, h , or Nusselt numbers (non-dimensional HTC) as given by Equation 10.

$$Nu = \frac{h D_c}{k_\infty} \quad (10)$$

The Nusselt number introduces an ambiguity to the result when defining the reference temperature for the calculation of gas properties. In the case of the present results, the bulk flow properties were used for the Reynolds number and Nusselt number, following the convention adopted by modern turbulent pipe flow correlations [7-9]. The Nusselt numbers were normalized with respect to the Dittus-Boelter correlation (Eq. 2) for fully developed turbulent pipe flow and the result is reported as an enhancement (augmentation) along the combustor wall as done by previous researchers [19-24].

4.1 Steady state heat transfer at different Reynolds numbers

The steady state heat transfer distribution on the liner wall was observed to be invariant along the azimuthal dimension (vertical variability was $< 1.5\%$), as expected from the symmetry of the time-averaged swirling flow and the axisymmetric test setup. For this reason, the HTCs obtained from the IR imaging were vertically (azimuthally) averaged and only the variation along the axial dimension is reported. Combustor swirling flows have been shown to

exhibit periodic flow asymmetries ($f > 24$ Hz) such as the precessing vortex core (PVC) [12,13]. However, no significant temporal variability was observed on the temperature measurements at steady state. The reason for this is that the timescale of the flow oscillations is negligible compared to the time scale of heat diffusion through the fused silica liner. The Fourier number can be calculated as a metric of the relative significance between the different time scales. Considering the thermal diffusivity of fused silica ($\alpha \approx 8.4767 \times 10^{-7} \text{ m}^2\text{s}^{-1}$), the time scale of the oscillations ($1/f < 0.042$ seconds), and the liner thickness (0.004 m) yields a $Fo < 0.0022$. Moreover, any periodic flow vortices or asymmetries (such as those generated by the PVC) only represent a fraction of the overall convective transport of energy between the fluid and wall; and are therefore not expected to impact the local HTC's drastically.

The Nusselt numbers obtained for Reynolds numbers extending from 11 500 to 138 000 are shown in Figure 11. The normalized flow field within the combustor for the different mass flows was approximately self-similar for the range of Reynolds numbers measured and for the swirl number of the fuel nozzle (0.74 ± 0.04). The self-similarity in combustor flows has also been reported for other burner geometries and fuel nozzles [36, 24]. For this reason, only the normalized flow field for the $Re = 11\,500$ case is shown in Figure 12. The combustor dump plane (exit of the nozzle) is located at $X/D_N = 0$ and only half of the combustor domain was acquired (Figure 2). The velocities were normalized with respect to the average axial velocity within the fuel nozzle. Further information on the flow field, including the coherent and stochastic vortical structures forming downstream of the nozzle, can be found in Gomez-Ramirez et al. [32].

Figure 11. Experimental Nusselt numbers for several Reynolds numbers.

Figure 12. Normalized flow field for Reynolds number $\sim 12\,000$. Nozzle dump plane is at $X/D_N = 0$. Liner wall is at $R/D_N = 1.45$.

The swirling jet expanded into the combustor primary zone and reattached along the liner. The white outline in Figure 12 indicates the location where the time-averaged axial velocity was zero, which corresponds to the center of the shear layer and point of flow reattachment on the liner. The region upstream of the reattachment point forms a corner recirculation with lower heat transfer rates due to the limited mixing and relative slower flow velocities. Downstream of the impingement point the swirling jet attached to the wall, and subsequent boundary layer growth led

to a decrease in the heat transfer. The reattachment point was observed at $X/D_N \approx 1.15$, downstream of the location of peak heat transfer, which was observed at $X/D_N = 1.03 \pm 0.02$ for all mass flow cases. A consistent observation was also made by Dellenback et al. [14] for different swirl and Reynolds numbers, with their peak heat transfer on average ~ 0.33 step heights ($0.16D_N$ for their setup) upstream of their re-attachment point. This mismatch has not been reported in more recent investigations of annular combustor models [22, 24]. The mismatch could be due to an increased tangential and axial shear, analogous to increased transport of energy per the Reynolds analogy, as the flow turns back from the reattachment point into the corner recirculation.

The Nusselt numbers normalized with respect to the Dittus-Boelter correlation are shown in Figure 13 (fluid properties evaluated at the bulk flow temperature to be consistent with previous studies and other correlations). The overlap between the different mass flows indicates that the correlation approximately captures the scaling of the combustor heat transfer with Reynolds number ($Re^{0.8}$). There is however a suggestion in the data of decreasing enhancement with increasing Reynolds number, as reported in [22, 19, 14]. The peak enhancement decreased from 22.0 ± 2.1 at the lowest Reynolds number to 17.8 ± 0.9 at the highest; slightly outside the uncertainty bands for the measurement. Andreini and coauthors [24] also showed coincident enhancements away from the impingement location, with only a small decrease in peak enhancement with Reynolds number. Dellenback and coworkers [14] observed a peak heat transfer dependence on $Re^{0.46}$ and related the decrease in augmentation to the heat transfer dependence on Reynolds number for reattaching flow of $Re^{2/3}$. The swirling jet reattachment on the liner wall is analogous to annular jet impingement. Jet impingement heat transfer correlations [37] show a direct dependence on $\sim Re^{0.4}$ to $Re^{0.7}$ (varying by geometry). This explains the decreasing enhancement with Reynolds number for the combustor flow around the reattachment (impingement) point.

Figure 13. Normalized Nusselt numbers (enhancement or augmentation) with respect to the Dittus-Boelter correlation for fully developed pipe flow.

The self-similarity in the normalized flow field also explains the constant heat transfer distribution and peak heat transfer location at all mass flows, implying that there are no significant transitions in the fluid dynamics at higher Reynolds numbers. This further indicates that measurements at lower Reynolds numbers capture the location of

maximum heat transfer along the combustor and the approximate distribution of isothermal convective heat loads (independent of Mach or Reynolds number).

The slight decrease in enhancement with increasing Reynolds number suggested by Figure 13 could also be related to the limitations of the Dittus-Boelter equation. Modern correlations for fully developed turbulent pipe flow heat transfer such as those proposed by Gnielinski [8,9] (based on the work of Pethukhov[7]) and Sleicher and Rouse [10] apply to a broader range of conditions and have been validated against more extensive sets of experiments. While the difference in the fully developed heat transfer result is small between correlations (<10% for the conditions in the current investigation), there is a noticeable difference at higher Reynolds numbers. Figure 14 shows the spatially averaged and maximum enhancement with respect to different formulations as reference. The lower enhancement at higher Reynolds numbers as suggested when using the Dittus-Boelter equation, weakens when normalizing with respect to other correlations. This seems to imply that the decrease in enhancement may be partly due to errors in the scaling with Reynolds number as given by the Dittus-Boelter equation ($Re^{0.8}$), compared to that of for instance Sleicher and Rouse [10] ($Re^{0.83}$).

Figure 14. Maximum and axially averaged enhancement with respect to different correlations for fully developed turbulent pipe flow.

4.2 Numerical calculation results

The numerical results show approximately consistent heat transfer distributions and magnitudes compared to the experiment as shown in Figure 15 for Reynolds numbers ranging between 11 500 and 73 000. The average difference ($0.2 < X/D_N < 2.6$) between the experiment and CFD was 25.9%, 10.4%, 5.8%, and 9.6% for increasing Reynolds numbers. For the lowest Reynolds number, the simulation predicted the peak heat transfer location $0.25D_N$ upstream of the peak location observed experimentally. The incorrect prediction of peak heat transfer location is what primarily contributed to the larger average error at the lowest Reynolds number. In terms of the maximum heat transfer magnitude, the discrepancy with increasing Reynolds numbers was -17.1% , 3.8%, 13.3%, and 17.4%.

The agreement between the experimental and numerical heat transfer results suggests that the near wall formulation in the simulation is appropriate. Enhanced wall treatment was used in the calculation, which relies on a

separate model for the turbulent viscosity and dissipation rate near the wall to calculate the turbulent kinetic energy (TKE) [38]. Heat transfer on the other hand was calculated with the law-of-the-wall for temperature [38] (based on the Reynolds analogy); a function of, among other variables, the near wall TKE. The realizable k- ϵ model constants were calibrated with boundary layer flows and rotating homogeneous shear flows [34]. The dissipation rate equation takes into account the effects of mean rotation on the turbulent stresses and hence is better suited to reproduce the conditions in the inertial sublayer for rotating flows (including heat transfer). However, the location of peak heat transfer is dependent on the free flow dynamics. The turbulence model assumptions (isotropic Reynolds stresses) are better satisfied at higher Reynolds numbers [34], which explains the improved prediction of the peak heat transfer location at higher mass flows. Two-equation turbulence models assume small effects due to pressure gradients and anisotropic turbulence, conditions which are not satisfied at the impingement location. It has been shown that the k- ϵ models overpredict the turbulent kinetic energy near the stagnation region of impinging jets. Realizability limits prevent excessively high Reynolds shear stresses, which constrain the overestimation of the stagnation heat transfer to 15-30% in jet impingement flows [37], consistent with the observations in this study.

Figure 15. Comparison between experimental and computational HTC for various flow rates.

4.3 Comparison to previous investigations and discussion of the experimental results

The main challenge when comparing measurements from past investigations is that each has adopted different reference quantities, working fluids, swirl generators, and downstream geometries. The work of Dellenback et al. [14] used water as the working fluid, calculated the Nusselt number in terms of the pipe diameter downstream of the expansion (D_C), and evaluated the Reynolds number in terms of the pipe diameter upstream of the same (D_N). The research by Andreini et al. [24], Patil et al. [22], and Gomez et al. [23] was done for annular combustors. [24] chose the hydraulic diameter of their test section for their calculations, while [22, 23] used the hydraulic diameter of the full annulus.

Figure 16 shows the maximum Nusselt number obtained in different investigations, evaluated with property values at the bulk flow temperature. Only investigations for which sufficient information was available to define the Reynolds number as given by Eq. 11 were considered.

498

$$Re_c = \frac{\dot{m} D_C}{\mu_B A_C} \quad (11)$$

499

500 Where μ_B , $\dot{m}$, D_C , and A_C are the dynamic viscosity of the working fluid at the bulk temperature, mass flow, reference
 501 combustor diameter (hydraulic diameter of the test setup), and cross-sectional area of the combustor setup
 502 respectively. Figure 16 also includes the axial average of the results in this investigation from $0.2 < X/D_N < 2.5$,
 503 encompassing the upstream and downstream decay of the peak in heat transfer.
 504

505

506 **Figure 16.** Comparison between the present investigation Nusselt numbers and the maximum observed in other investigations.

507

*The data of Dellenback et al. [14] has been scaled, see discussion for details.

508 Annular combustor studies [22-24] show a two-dimensional heat transfer distribution on the liner, with the
 509 peak heat transfer decreasing both in the axial and azimuthal (along the annular circumference) directions. Lateral
 510 (circumferential) averages of the data are typically calculated for annular combustors to visualize the heat transfer
 511 enhancement as a function of axial location. The circumferentially averaged heat transfer however, includes the
 512 circumferential decay of the heat transfer peak, and therefore the true maximum heat transfer observed on the liner is
 513 not reflected by the circumferentially averaged data [22-24]. Both the maximum Nusselt number from the lateral
 514 averages and the true maximum observed in the two dimensional heat transfer distribution were available from the
 515 work of Gomez and coauthors [23] and are plotted in Figure 16. They reported different heat transfer distributions for
 516 the inner and outer annulus walls which were averaged for the comparison in the present study. The lateral average
 517 from the data of [23] matches approximately with the axial average of the present results (lower by ~8.7% on
 518 average), whereas the true peak heat transfer from [23] was on average ~ 7% higher than the maximum observed in
 519 the can combustor. Only the laterally averaged peak heat transfer from the work of Andreini et al. [24] was available
 520 and is shown in the figure to be 8.7% lower than the axial average in the current investigation. These results suggest
 521 that the true peak heat transfer in can and annular combustors will be of similar magnitude. The relative agreement
 522 between the laterally averaged maximum in the annular combustor and the axial average in the can combustor could

be explained if the azimuthal decay in heat transfer for the annular models is similar to the axial decay observed in the can model.

An important factor to consider between investigations is the different swirl numbers and expansion ratios used. There are two phenomena by which convective heat transfer is enhanced in isothermal combustor flows compared to fully developed pipe flow: 1) due to swirl, and 2) due to the reattaching flow after the expansion into the primary zone. The addition of swirl to the flow adds a tangential velocity component, introduces a centrifugal acceleration to the flow, modifies the impingement location on the liner, and triggers a central recirculation for swirl numbers greater than 0.6 [13], all of which have an impact on the transport phenomena near the wall. Dellenback and coworkers [14] experimentally characterized the effect of swirling flow on the local magnitudes of heat transfer downstream of a sudden pipe expansion for a fixed D_N / D_C ratio of 0.51. Their geometry is similar to that of can combustors, except for their swirl generator. Their results showed that an increase in swirl number, S , led to an upstream shift in the impingement location and to higher heat transfer enhancement. Based on their data for Re_N between 60 000 and 100 000 (Re_C of ~ 30 000 and ~ 50 000), the heat transfer increased with swirl number according to $S^{0.235}$. The enhancement due to reattaching flow after a sudden expansion has been characterized by Baughn et al. [16], who showed that a decrease in the expansion ratio D_N/D_C corresponded to an increase in peak heat transfer. The work by Dellenback et al. [14] with $D_N/D_C = 0.51$ would hence result in lower heat transfer compared to the present combustor model with a $D_N/D_C = 0.34$. If the expansion ratio for the annular combustor studies is defined as the ratio of the nozzle diameter to the annular height, the work of Andreni and coworkers [24] would correspond to an expansion ratio of 0.42 (a lower maximum heat transfer is expected), and that of Gomez et al. [23] to 0.36. The similar expansion ratio of this investigation and that of Gomez and coauthors [23] may explain the relatively good agreement between the observations in Figure 16. According to the results by Baughn et al. [16], peak heat transfer increased according to $\sim (D_N/D_C)^{-1.053}$, however this relationship may be different in the presence of swirl. The results from Dellenback and coworkers [14] for swirl numbers between 0.6 and 0.77 are shown in Figure 16. However, the data from Dellenback et al. [14] were corrected to account for the different Prandtl numbers of water and air and the difference in expansion ratio. Despite the corrections, the modified results from Dellenback were still $\sim 25.2\%$ lower than the axially averaged Nusselt number obtain in the present study.

Discrepancies in the data could also be attributed to the reference temperature. Modern turbulent pipe flow correlations have shown that evaluating the Nusselt number at the bulk flow temperature requires a correction to the correlations of the form $(T_\infty/T_w)^{n_T}$ to account for fluid property changes near the wall [7, 9]. The experimental data in the present investigation were fitted with Equation 14, following Gnielinski's formulation [8-9] for fully turbulent pipe flow, given by Equation 12, and adding a factor $K_S Re^{n_S}$ to account for the enhancement due to swirl and flow reattachment. Gnielinski's correlation was chosen because it was derived from the Chilton-Colburn analogy, and hence relies on a strong theoretical basis. In-built into the Gnielinski equation are provisions to account for different wall roughness (through the Darcy-Weisbach friction factor f) and the effects due to property changes with temperature.

$$Nu_{Gnielinski} = \frac{(f/8) (Re_{C,\infty} - 1000) Pr_\infty}{1 + 12.7 \sqrt{f/8} (Pr_\infty^{2/3} - 1)} \left(\frac{T_\infty}{T_w} \right)^{n_T} \quad (12)$$

$$f = (1.8 \log_{10} Re_B - 1.5)^{-2} \quad (13)$$

$$Nu_B = K_S Re_{C,\infty}^{n_S} Nu_{Gnielinski} \quad (14)$$

Where Eq. 13 applies to a smooth pipe, and the temperature dependence exponent n_T proposed by Gnielinski [8] equals 0.45, and is applicable for $0.5 < T_W/T_\infty < 1.5$ (Pethukhov's formulation [7] includes expressions for the coefficients and n_T exponent in Eq. 13, which depend on Prandtl and Reynolds numbers). The factor K_S was equal to 39.15 and 61.06 for the mean and maximum Nusselt number trends respectively. Similarly, the exponential dependence on the Reynolds number, n_S , was -0.0684 and -0.0846 for the mean and maximum heat transfer cases respectively, which reflects the weak decrease in enhancement with increasing Reynolds. The exponent n_S has a strong influence on the coefficient K_S ; assuming $n_S \approx 0$, K_S was 17.85 and 23.13 for the mean and maximum heat transfer. The $K_S Re_{C,\infty}^{n_S}$ factor is related to the swirler and combustor geometry, as inferred from the work of Baughn et al. [16] and Dellenback et al. [14]. Therefore, the coefficients calculated are only applicable to the current system, but the formulation is nevertheless included as a potential framework to compare different studies.

Ji and Gore [39] and Stopper et al. [40] have shown that the fluid dynamics within combustors can be significantly different at isothermal and reacting conditions. The increased temperature across the flame front leads to increased velocity magnitudes and fluctuations, subsequently impacting the strength of the swirling jet (swirl number), the location of jet reattachment along the liner, and the turbulence kinetic energy. In reaction, multiple heat transfer modes are present including convection, radiation, and thermochemical heat release. The sharp gradients in temperature and composition lead to difficulties in defining appropriate temperature potentials and flow properties to quantify the convective heat transfer coefficients. In spite of these challenges, the isothermal data presented can be used as a first estimate of the convective component of the heat transfer observed at reacting conditions, provided that the Reynolds number and Nusselt number are calculated according to the appropriate local free flow temperature observed in the reacting burner. The experimental setup in the present work is capable of operation at reacting conditions, and future investigations will explore the effects introduced by reaction. Nevertheless, isothermal measurements are a prerequisite to understand convective heat transfer in a reacting environment, without the added complexity of multiple heat transfer modes and sharp property gradients.

5. CONCLUSIONS

The non-reacting heat transfer for an optical gas turbine combustor model was studied experimentally and computationally. The experiments were carried out for Reynolds numbers with respect to the combustor diameter between 11 500 and 138 000. The present work theoretically demonstrated the feasibility of infrared thermography (1-5 μ m) through fused silica (quartz glass) for low quartz glass temperatures and introduced a novel analysis of the three-dimensional heat losses for constant heat flux experiments. The results obtained were compared against numerical calculations and other experiments.

The normalized heat transfer results (with respect to several correlations for fully developed turbulent pipe flow), showed that the augmentation away from the impingement (jet reattachment) location was approximately constant for different Reynolds numbers. This indicates that the correlations for fully developed turbulent internal flow captured the increase in heat transfer with operating Reynolds number for the can burner configuration. The data supports however that the scaling is slightly different at the stagnation region as observed by others [14, 19, 22, 24]; an observation that is consistent with the scaling with Reynolds numbers observed in jet impingement heat transfer correlations.

The RANS numerical calculation with the realizable $k-\epsilon$ turbulence model yielded heat transfer results in general agreement with the experiment. Both the simulated peak and axially averaged heat transfer had an average discrepancy of 13% with respect to the experiments for the Reynolds numbers compared. Based on these results, the realizable $k-\epsilon$ turbulence model may be acceptable to explore the impact different geometrical parameters (expansion ratio, swirl number) have on heat transfer. The experimental results were also compared with measurements from an annular combustor [23] yielding a difference of $\sim 7\%$ for the maximum observed heat transfer along the liner. The laterally averaged Nusselt numbers from annular combustors at the location of maximum enhancement were within 8.7% of the axially averaged Nusselt numbers observed along the can combustor in this investigation.

The maximum and average heat transfer were also fitted according to a modified Gnielinski formulation to account for the augmentation within the combustor primary zone. The fit coefficients are only applicable to the current combustor geometry, but provide a framework to compare different experiments. The study presented highlights the need for additional data to characterize and understand combustor heat transfer for different burner geometries. In particular, there is no comprehensive study on the simultaneous impact of the burner expansion ratio and swirl number on liner heat transfer.

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