Experimental investigation of unsteady convective heat transfer under airflow velocity and temperature variations

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Abstract
An experimental approach to study unsteady local heat transfer characteristics due to airflow velocity and/or airflow temperature variations is presented. It uses controlled electrical heaters and rotating vanes to independently vary the flow and thermal boundary conditions. Time-resolved surface temperatures are measured using an in situ calibrated infrared thermography camera. Those surface temperatures are analyzed by modeling the transient conjugate heat transfer process in the wall to obtain locally resolved surface heat flux distributions. The applicability is illustrated for a flow and heat transfer behind a tetrahedral vortex generator on a flat plate.

Keywords: conjugate, convective, transient heat transfer, 3D heat conduction, vortex generator, infrared

(Some figures may appear in colour only in the online journal)

1. Introduction
Unsteady heat transfer processes are an intrinsic part of gas turbines. They occur due to the operating principle itself, changing heat loads, varying cooling air mass flows and unsteady flow phenomena. In addition, for stationary gas turbines the increasing amount of renewable energy leads to a higher proportion of unsteady operating conditions with frequent start-up and shut-down processes as well as operation in different partial load conditions (Magni et al 2016, Kuželka et al 2019, Abudu et al 2021). Hence, unsteady heat transfer processes are an area of great interest and will gain further importance in the future. To ensure even safer and more efficient operation under transient conditions, it is of particular interest to numerically and experimentally generate, capture and evaluate the effects of these operating conditions on gas turbine cooling.

To experimentally study unsteady heat transfer, an experimental setup is required that allows a controlled and repeatable variation of the boundary conditions. In case of airflows, many researches applied a mesh heater to change the airflow temperature (Ireland et al 1999, Newton et al 2003, Poser et al 2005, Liu et al 2014). This type of heater is characterized by a homogeneous temperature field over the entire cross-section and responds quickly to changes due to their low thermal mass.

To vary the airflow velocity, a wide variety of designs exist and were applied in experimental studies already. All designs use either a volume change or a cross section change to vary the flow velocity. Volume change designs use adjustable reservoirs which are connected to the main flow path. The volume change is achieved by, for example, a piston (Fallen 1982, Park et al 1982), a membrane pump (Persoons et al 2012) or a subwoofer membrane (Moon et al 2005, Ji et al 2008, Lalizel et al 2012, Li et al 2013). The cross-section change can be realized with a rotating valve (Sheriff and Zumbrunnen 1994, Habib et al 2002, Zohit et al 2005, Elshafei et al 2008), a sliding gate
The complex flow field with up- and downwash regions of the longitudinal vortices leads to a complex heat transfer pattern. Starting at the tip, the two main vortices lead to two streaks of high heat transfer in their downwash regions. The corner vortices arise at the two sharp ends of the VG. In between a main vortex and its corresponding corner vortex a so-called upwash region leads to a particular streak of low heat transfer. In addition, around the VG a horseshoe vortex is induced. A typical heat transfer pattern in the wake of the VG is visualized in figure 2 (top view). Blue areas represent regions with low heat transfer and yellow areas represent regions with high heat transfer.

The top wall of the test section includes three openings for IR radiation transmitting windows (F) which enables flexible access for the IR camera (H) to capture the surface wall temperature at different locations of the flat plate. As IR transmitting window material serves calcium fluoride (CaF₂). It reaches with its thickness of 0.005 m a transmittance of 95% in the sensitive wavelength range from 1.5 to 5.1 μm of the applied IR camera1.

Additionally, the test section top wall includes 0.205 m upstream the leading edge of the flat plate a port for hot-wire anemometry and thermometry2 (D). The hot-wire system measures locally the temporally varying inlet conditions (inlet velocity $u_{in}$, inlet temperature $T_{in}$). Due to the installation situation, the orientation of the two single-sensor probes with a wire diameter of 10 μm is 90°. One probe operates in constant temperature mode (CTA) and measures the airflow velocity. The other probe operates in constant current mode (CCT) and measures the local airflow temperature.

Further downstream above the flat plate, fine wire thermocouples (E) with a wire diameter of 80 μm measure the local airflow temperature at mid channel height.

2.2. Airflow control system

The mesh heater together with its two DC power supply units and the adjustable vanes are the two components of the airflow control system and enclose the test section. If one of the two control variables heater voltage $U_H$ or angle of attack of the vanes Ω is changed, the pressure drop on the flow path to the vacuum pump changes and thus also the airflow velocity $u$ and airflow temperature $T_f$ within the test section. In conclusion, a controlled variation of the airflow velocity or airflow temperature requires a necessary controlled change of $U_H$ as well as Ω.

The mesh heater itself is built up with six steel meshes, arranged one after the other downstream and with a square cross-sectional area of 0.19 × 0.19 m², a mesh width of 40 μm and a wire diameter of 25 μm.

Downstream the test section three adjustable vanes are placed above each other. By varying the angle of attack Ω at the same time, the gap between the vanes and to the lower and upper wall varies and so does the pressure drop. Each 0.01 m

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1 FLIR SC7600.
2 SVMtec hot-wire system eCTA and eCCA with probe type HWP10/90.
thick vane has a wingspan of 0.12 m and a chord length of 0.049 m. All blades are connected by gears to the same shaft, which is driven by an electric motor via a reduction gearbox. A picture of the complete assembly is shown in figure 3.

With the aid of a PI controller the measured $\Omega$ is used to vary the rotational speed of the electric motor and therefore to control $\Omega$. An automatic calibration in three steps, shown in figure 4, assigns the measured $\Omega^\ast$ to the actual $\Omega$. In the first step, the measured $\Omega^\ast_1$ is set to zero in an arbitrary fixed position. Then the vanes are rotated by 200° at constant speed. At the same time, the hot-wire probe records $u_{in}$ as a function of $\Omega^\ast_1$. When $\Omega^\ast_1$ reaches the position of the minimum gap between the vanes, a sharp minimum of $u_{in}$ is obtained, which can be determined exactly. The vane angle $\Omega$ to this minimum is defined as 90° and provides the assignment of $\Omega^\ast_2$ to $\Omega$.

The second step of calibration optimizes the obtained assignment of $\Omega^\ast_2$ to $\Omega$ with additional measurements of $u_{in}$ at different angles and non rotating vanes. Between 80° and 100°, $u_{in}$ is measured with a resolution of 1.25°. The minimum obtained in this step is considered as exact for the assignment of $\Omega^\ast$ to $\Omega$.

While the first two steps provide the exact assignment of $\Omega^\ast$ and $\Omega$, the third step of calibration determines the characteristics of $u_{in}$ versus $\Omega$. For $\Omega$ greater than 140°, $u_{in}$ hardly changes at all. So only the range between 90° and 140° is used for experiments.

The airflow control system aims to reproducibly control the individual histories of $u_{in}$ and $T_{f,in}$ for each experiment by continuously changing $U_H$ and $\Omega$. However, since the physical relationship

$$ (u_{in}, T_{f,in}) = (U_H, \Omega) $$

is not linear and the response time of the airflow control system devices is hardly faster than the desired changes, a PID controller cannot be used to control the process variables $u_{in}$
Figure 3. Adjustable vanes mechanism (legend: (I) asynchronous electric motor, (II) optical singleturn shaft encoder, (III) worm gearbox, (IV) gearbox, (V) vanes, (VI) test section).

Figure 4. Initial calibration of the adjustable vanes.

and $T_{f,\text{in}}$. For this reason, a steady-state operating range is experimentally determined and serves as the basis to calculate $U_H$ and $\Omega$ from the required course of $u_{\text{in}}$ and $T_{f,\text{in}}$. In the steady-state case, two equations describe the physical relationship between the control variables and the process variables. On the one hand the steady and incompressible Bernoulli equation

$$p_\infty = p_{\text{in}} + \frac{1}{2}\rho_f (1 + \zeta_H) u_{\text{in}}^2$$  (2)

with a Reynolds number dependent loss coefficient $\zeta_H$ for the mesh heater from the ambient $(p_\infty, T_\infty)$ to the inlet section provides a connection. On the other hand, the energy equation

$$\dot{m}_f c_p (T_{f,\text{in}} - T_{f,\infty}) = P_H$$  (3)

defines the influence of the heater power on the flow variables at the inlet. If these equations are further transformed using Ohm’s law

$$P_H = \frac{U_H^2}{R_H}$$  (4)

and the ideal gas equation

$$p = \rho_f R_s T_f,$$  (5)

the following approximate relationships are obtained:

$$\frac{1}{u_{\text{in}}^2} \propto f\left(\frac{1}{T_{f,\text{in}}^2}\right), \quad \Omega = \text{const.}$$  (6)

$$\frac{T_{f,\text{in}} - T_\infty}{u_{\text{in}}} \propto f\left(U_H^2\right), \quad \Omega = \text{const.}$$  (7)

While $R_H$ represents the constant heater mesh resistance, $R_s$ is the specific gas constant for dry air.

By determining the required $U_H$ to reach different temperature levels for different $\Omega$ and then plotting the measured data against each other as in the two equations (6) and (7), one obtains the figures 5 and 6. Both figures visualize the functional relationship which is linearly approximated. In conclusion, the operating range is mathematically described and can be used to calculate $U_H$ and $\Omega$ for predefined histories of $u_{\text{in}}$ and $T_{f,\text{in}}$ in advance. The procedure to calculate the histories of $\Omega$ and $U_H$ was implemented in LabVIEW™ with a stochastic solver for systems of non-linear equations. The coefficients for
2.3. Inlet velocity profiles

During each experiment Ω varies according to the required histories of $u_{in}$ and $T_{f,in}$. As the flow velocities in the channel are in areas significantly below the speed of sound, these geometric changes at the outlet might also change the velocity profile in front of the flat plate and above the flat plate. To exclude such an influence, the velocity profile at the inlet was traversed for different vane angles. Figure 7 shows the profiles normalized with the mean velocity $\bar{u}$ over the channel height for different Ω. The inlet profile is independent of Ω.

Two aspects cause a non-symmetrical velocity profile. First, the reduction of the cross-section between the mesh heater and the test section (see figure 1). Second, the traversing procedure has an influence. During the acquisition of the velocity profiles, the probe was pulled back stepwise to the probe ports.

The measurement of the velocity profile at two further positions perpendicular to the main airflow direction (0.03 and −0.03 m) yielded the same result as the profiles shown in figure 7 for the center of the channel.

Independent of Ω, the local velocity measured at $z = 0.02$ m always corresponds to $\bar{u}$. This fixed position was selected for the experiments and the measured local velocity has been defined as $u_{in}$.

2.4. In situ calibration of IR camera

As shown in figure 1, an IR camera captures the surface temperature of the evaluation area with temporal and spatial resolution. Many influencing factors, such as ambient radiation, transmittance of the window and emittance of the observed surface influence the measurement result. Therefore, each of these quantities would have to be known exactly in order to get accurate surface temperature measurements with the IR camera. In order to take all these factors into account and to obtain quantitative measurement results, the IR camera was calibrated in situ with the aid of one type T surface thermocouple in the field of view (FOV). The graph of figure 8(a) includes three pictures in top view of the surface thermocouple glued on a perspex cylinder and mounted into the flat plate. The top left picture shows the cylinder with surface thermocouple without black paint coating. The top right picture shows again the same cylinder with black paint coating. The complete flat plate as well as the cylinder and the surface thermocouple are coated with black paint in order to get a homogeneous emittance of the surface. The bottom left picture represents a detail of one IR image during an experiment. One can distinguish

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3 Omega™ CO2-T.
4 Omegabond OB101-2.
5 Hallcrest type SPB100.
the copper leg of the thermocouple from the surrounding. Due to different material properties of copper \((\rho = 8920 \text{ kg m}^{-3}, c = 385.1 \text{ J (kg K)}^{-1}, k = 377.05 \text{ W (m K)}^{-1})\) compared to constantan \((\rho = 8920 \text{ kg m}^{-3}, c = 393.5 \text{ J (kg K)}^{-1}, k = 21.1 \text{ W (m K)}^{-1})\) and Perspex, the local temperature of the copper leg rises more slowly.

For the evaluation area the experimentally determined heat transfer coefficient is always lower than 200 W (m² K)⁻¹. Together with the average thermal conductivity \(k_{tc}\) of 199.08 W (m K)⁻¹ and the thickness \(L_{tc}\) of 10 µm, this leads to a Biot number \((Bi = h L_{tc} / k_{tc})\) smaller than 2.5 × 10⁻⁶. Hence, the assumption of a homogeneous measuring tip temperature of the thermocouple is justified (lumped body assumption).

The graph of figure 8(a) shows an exemplary calibration curve received from a single experiment. The experiment started with an airflow temperature increase from ambient conditions to a higher temperature level. Then an airflow temperature pulsation with a frequency of 0.1 Hz is superimposed. At the end the airflow temperature as well as the airflow velocity have been increased to their maximum values.

To obtain the calibration curve, the radiation measured locally above the measuring tip of the surface thermocouple is plotted against the measured temperature of the surface thermocouple \(T_{tc}\). The output quantity of the IR camera is given in Digital Level DL. The maximum detected radiation of a pixel in the FOV of the IR camera as well as the measured radiation at the measuring tip of the thermocouple are plotted against \(t\) in the graph of figure 8(b). Due to the varying heat transfer distribution over the surface, both histories differ from each other.

The airflow temperature and airflow velocity increase at the end of the experiments leads to a strong temperature and radiation increase at the measuring tip. Thus, this final sequence of the experiment enables a wide span of measured radiation at the measuring tip of the surface thermocouple and the longest possible evaluation time \(t_{max}\).

### 3. Evaluation method

The previous sections focused on the generation and acquisition of the unsteady convective heat transfer situation. Another main object is the temporally and spatially resolved evaluation of the heat transfer situation schematically visualized in figure 9. For the evaluation the method according to Estorf (2006) was applied. This method is based on the analytical evaluation of the transient three-dimensional heat transfer equation within the wall

\[
\frac{\partial T_s}{\partial t} = \frac{k_s}{\rho_s c_s} \left[ \frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} + \frac{\partial^2 T_s}{\partial z^2} \right] \tag{8}
\]

for constant material properties. Estorf (2006) also shows that linear temperature dependent material properties of the wall, resulting in a temperature dependent thermal diffusivity \(\alpha_s\), can be taken into account with a Kirchhoff transformation. A transient surface heat flux

\[
z = 0: \quad -k_s \frac{\partial T}{\partial z} \bigg|_{z=0} = \dot{q}_u(t) \tag{9}
\]

captures the heat transfer at the interface between fluid and solid.

As special requirements for the experimental setup and the experimental procedure, on the one hand the entire setup has to be in thermal equilibrium

\[
t = 0: \quad T = T_0 \tag{10}
\]
at the beginning of the experiment. On the other hand, the wall in \(z\)-direction is considered to be semi-infinite

\[
z \to +\infty: \quad T = T_0. \tag{11}
\]

This condition can be sufficiently realized by limiting the experiment duration \(t_d\)

\[
t_d < L^2 \frac{\rho_s c_s}{16 k_s} \tag{12}
\]

according to Schultz and Jones (1973). It ensures for a one dimensional heat conduction situation with an initial step change of \(\dot{q}_u\), a temperature change at the penetration depth \(z = L\) of less than 1% compared to the surface temperature change.
To find an analytical solution for the heat transfer situation of the experimental setup, Estorf (2006) considered a similar situation in the first step (figure 9, evaluation model step 1). However, this similar situation only contains heat conduction processes. For this heat conduction problem within an infinite cube with an initial heat release $2q(x,y,z=0)$ in the $xy$-plane, an analytical solution exists in the Fourier space

$$\tilde{\theta}(u,v,z,t) = \frac{1}{\rho c_\varepsilon \sqrt{\pi \alpha_f t}} \int_0^t q(u,v,\tau) \times \exp \left[ -\alpha_s (u^2 + v^2) t - \frac{z^2}{4 \alpha_f t} \right] \mathrm{d}\tau.$$ \hspace{1cm} (13)

The frequencies $u$ and $v$ belong to the respective spatial directions $x$ and $y$ and $\tilde{\theta}$ represents the normalized temperature $\theta = T_r - T_0$ in Fourier space. In order to consider the symmetry to the $xy$-plane and to apply the obtained results to the experimental setup, the initial heat release contains the factor two.

Using the Duhamel principle (figure 9, evaluation model step 2), Estorf (2006) transformed the solution obtained for the initial condition of an initial heat release into a wall temperature field

$$\tilde{\theta}(u,v,z,t) = \frac{1}{\rho c_\varepsilon \sqrt{\pi \alpha_f t}} \int_0^t \hat{q}(u,v,\tau) \times \exp \left[ -\alpha_s (u^2 + v^2) (t - \tau) - \frac{z^2}{4 \alpha_f (t - \tau)} \right] \mathrm{d}\tau$$ \hspace{1cm} (14)

for a time-dependent heat flux $2\hat{q}(x,y,z=0)$ in the $xy$-plane. For example, Özisik (1985) explains the Duhamel principle in detail. The inverse of the temperature is

$\hat{q}(u,v,t) = \frac{k_s}{\sqrt{\pi \alpha_f}} \times \int_0^t \frac{1}{\sqrt{t - \tau}} \tilde{\theta}(u,v,z=0,\tau) e^{-\alpha_s (u^2 + v^2)(t-\tau)} \mathrm{d}\tau$ \hspace{1cm} (15)

and corresponds to the surface heat flux $\hat{q}_w$ of the experimental setup in the Fourier space.

A numerical approximation of the surface temperature $T_w$ is necessary to solve the integral of equation (15). Applying the piecewise linear approximation of $T_w$

$$t_i < \tau < t_{i+1} :$$

$$T_w(\tau) = \frac{T_w(t_{i+1}) - T_w(t_i)}{t_{i+1} - t_i} (\tau - t_i) + T_w(t_i)$$ \hspace{1cm} (16)

with a constant time step $\Delta t$ leads to the finite sum of the surface heat flux in Fourier space

$$\hat{q}_{w,lm}^n = k_s \sum_{k=0}^{n-1} \left\{ \left( \frac{1}{2} \omega_{lm}^2 + \omega_{lm} (t_n - t_k) \right) \times \frac{\Delta t}{\sqrt{\pi \alpha_f}} \tilde{\theta}_{w,lm,k+1} - \tilde{\theta}_{w,lm,k} \right\} \times \left( \text{erf}[f_{lm}(t_k)] - \text{erf}[f_{lm}(t_{k+1})] \right)$$

$$+ \frac{\tilde{\theta}_{w,lm,k+1} - \tilde{\theta}_{w,lm,k}}{\sqrt{\pi \alpha_f} \Delta t} \left( \sqrt{t_n - t_k} e^{-\alpha_s (z_n - z_k)} \right) - \sqrt{t_n - t_k} e^{-\alpha_s (z_n - z_{k+1})} \right\}. \hspace{1cm} (17)$$

The discrete frequencies $u_i$ and $v_m$ result from the uniform discretization ($L$ pixel $\times M$ pixel) of the evaluated surface and appear in equation (17) with the terms $\omega_{lm} = \sqrt{u_i^2 + v_m^2}$ and $f_{lm}(\tau) = \omega_{lm} \sqrt{\alpha_f (t_n - \tau)}$. The indices $l$ and $m$ relate to the
discretization (pixel-wise) in x- and y-direction. The superscript \( n \) is the discrete time point of the evaluation of \( \hat{q}_w (x, y, t) \). Using a cosine transformation to calculate the normalized surface temperature \( \hat{\theta}_w \) instead of a Fourier transformation leads to adiabatic sidewalls at the borders of the evaluated area. An inverse Fourier transformation or cosine transformation respectively delivers the spatially and temporally resolved surface heat flux \( \hat{q}_w (x, y, t) \).

The spatially and temporally resolved heat transfer coefficient results from the application of the Newton’s law of cooling

\[
h(x, y, t) = \frac{\hat{q}_w (x, y, t)}{T_w (x, y, t) - T_{ref}(x, y, t)}
\]

in a post processing step.

The reference temperature \( T_{ref} \) within this study is the free stream temperature resolved in time and space. To get the complete \( T_{ref} \)-field, the interpolation method according to Poser and von Wolferdors (2011) has been applied.

The influence of spatial and temporal discretization has been discussed by Estorf (2006). A comparison to heat transfer measurements using narrowband thermochromic liquid crystals is given in Brack et al (2017).

### 4. Uncertainty analysis

The evaluation quantities \( \hat{q}_w \) and \( h \) are affected by the uncertainty of the measurement quantities. Calibration procedures enable to determine the uncertainty of the measurement quantities which are given in table 1. For the applied thermocouples a stationary calibration was carried in a calibrated dry block calorimeter\(^6\). The CTA hot-wire probe is calibrated by the manufacturer, the CCT probe is calibrated in situ. A calibrated fine wire thermocouple, placed next to the CCT probe, provides the measurement values for the in situ calibration during the determination of the operating range of the experimental setup.

The method according to Moffat (1982) has been used to calculate the uncertainties of the evaluation quantities. Applying this method to \( \hat{q}_w \) in Fourier space leads to the following equation for the uncertainty of the surface heat flux amplitudes

\[
\delta \hat{q}_w = \sum_{k=0}^{n-1} \left[ \left( \frac{\partial \hat{q}_w}{\partial \hat{\theta}_w} \right)^2 \left[ \delta \hat{\theta}_w \right]^2 + \left( \frac{\partial \hat{q}_w}{\partial \rho_s} \right)^2 \left[ \delta \rho_s \right]^2 + \left( \frac{\partial \hat{q}_w}{\partial c_s} \right)^2 \left[ \delta c_s \right]^2 \right]^{1/2}
\]

An inverse Fourier transform yields then \( \delta \hat{q}_w \).

To obtain the uncertainty of the heat transfer coefficient \( \delta h \), Moffat’s method was applied too and results in

\[
\delta h = \sqrt{\left( \frac{\partial h}{\partial \hat{q}_w} \right)^2 \left[ \delta \hat{q}_w \right]^2 + \left( \frac{\partial h}{\partial T_{ref}} \right)^2 \left[ \delta T_{ref} \right]^2}
\]

\[
= \sqrt{\left( \frac{h}{\hat{q}_w} \right)^2 \left[ \delta \hat{q}_w \right]^2 + \left( \frac{h}{T_{ref} - T_w} \right)^2 \left[ \delta T_{ref} \right]^2}
\]

\[
+ \left( \frac{h}{T_{ref} - T_w} \right)^2 \left[ \delta T_w \right]^2.
\]

\( \delta h = \sqrt{\left( \frac{\partial h}{\partial \hat{q}_w} \right)^2 \left[ \delta \hat{q}_w \right]^2 + \left( \frac{\partial h}{\partial T_{ref}} \right)^2 \left[ \delta T_{ref} \right]^2}
\]

\[
= \sqrt{\left( \frac{h}{\hat{q}_w} \right)^2 \left[ \delta \hat{q}_w \right]^2 + \left( \frac{h}{T_{ref} - T_w} \right)^2 \left[ \delta T_{ref} \right]^2}
\]

\[
+ \left( \frac{h}{T_{ref} - T_w} \right)^2 \left[ \delta T_w \right]^2.
\]

Table 1. Independent input quantities with their typical measured values and standard deviation.

| Qty         | Meas. range | Std. dev. \( \delta x \) |
|-------------|-------------|---------------------------|
| \( u_0 \)   | 5–25 m s\(^{-1}\) | 0.1 m s\(^{-1}\) + 0.05\( u_0 \) |
| \( T_0 \)   | 293.15 K    | 0.1 K                     |
| \( T_{ref} \) | 293.15–333.15 K | 0.1 K                     |
| \( T_w \)   | 293.15–333.15 K | 0.1 K                     |
| \( \rho_s \) | 1190 kg m\(^{-3}\) | 10 kg m\(^{-3}\)         |
| \( c_s \)   | 1470 J (kg K\(^{-1}\)) | 20 J (kg K\(^{-1}\))    |
| \( k_s \)   | 0.19 W (m K\(^{-1}\)) | 0.01 W (m K\(^{-1}\))   |

\( \Delta x \) 0.4 mm Pixel\(^{-1}\) 0.04 mm Pixel\(^{-1}\)

The quantities \( \hat{q}_w \) and \( T_w \) are considered as independent of each other. For the experiments discussed in the results section, the relative uncertainties \( \delta \hat{q}_w / \hat{q}_w \) and \( \delta h / h \) vary between 10% to 15% on average. They are dependent on position and time. Furthermore, smaller \( \hat{q}_w \) and \( h \) lead to larger relative uncertainties. The relative uncertainty \( \delta h / h \) is affected also by the driving temperature difference \( T_{ref} - T_w \). If \( T_{ref} - T_w \) approaches zero, \( \delta h / h \) tends to infinity. Further details are given in Brack (2020).

### 5. Results

To demonstrate the functionality and performance of the experimental setup as well as to visualize the possibilities of the evaluation method, the results of two different experiments are presented in the following. First, an experiment with a constant airflow velocity at a periodically pulsating airflow temperature is discussed. The frequency \( \nu \) of the pulsation was set to 0.1 Hz. Then a second experiment is presented in which the airflow temperature maintained constant in sections and the airflow velocity pulsed continuously with a frequency \( \nu \) of 0.05 Hz.

Both experiments represent generic ones. They are not derived from any explicit periodic phenomenon, e.g. in the secondary air system in gas turbine steady-state operation. However, both are particularly suited to demonstrate the performance, the repeatability, and the limitations of the experimental framework in single experiments. The frequencies of the two experiments with periods of several seconds are in the range in which unsteady, periodic and/or non-periodic, operating point changes can occur in turbomachinery (LaGraff et al 2006).
Figure 10. Inlet airflow condition histories as well as control variable histories for the pulsing airflow temperature experiment.

5.1. Pulsating airflow temperature

Figure 10(a) shows the predefined set point history for the pulsating airflow temperature experiment. Up to 152 s the airflow velocity keeps constant at 22 m s$^{-1}$. Then it increases linearly to the maximum value of 25.47 m s$^{-1}$ and stays constant again. The temporal set point variation of the airflow temperature starts with an S-shaped increase from ambient conditions to a value of 306.15 K after 14 s. After reaching the higher temperature level, the airflow temperature pulsates with an amplitude of 9 K around the average value of 306.25 K at a frequency of 0.1 Hz. This pulsation lasts up to 152 s. Then the airflow temperature linearly increases to the maximum temperature level of 321.36 K.

This predefined set point history served as input to calculate the temporal variation of $\Omega$ and $U_H$, which are shown in figure 10(c). The main varying control variable is $U_H$. However, even if only $T_{f,in}$ changes up to 152 s, this requires not only a variation of the $U_H$, but also a variation of $\Omega$ at the same time. The changes of $\Omega$ and $U_H$ are antagonistic. An increase in $U_H$ gives a reduction in $\Omega$ and vice versa.

The measured histories of $u_{in}$ and $T_{f,in}$ are plotted in figure 10(b). A comparison with the set point history of figure 10(a) confirms a good agreement with only minor deviations. The inlet velocity $u_{in}$ reaches a mean value of 22.6 m s$^{-1}$ up to 152 s and pulsates slightly with an amplitude smaller than 0.4 m s$^{-1}$. The airflow control system also reproduces the rapid linear rise afterwards. A slight overshoot is visible. As described in section 2.4, this part of the experiment only aims at evaluating the complete experiment duration with a pulsating flow temperature. An evaluation of the heat transfer of the section after 152 s is not carried out.

Between the measured value and the set point value of $T_{f,in}$ a constant offset of 0.82 K exists. It results from an increased ambient temperature between the point in time of the measurement of the operating range and the point in time of the actual experiment. During pulsation the maxima reach an average value of 315.88 K and the minima are on average 299.63 K.
Figure 11. Local heat transfer data for the pulsation airflow temperature experiment with \( \nu_{\text{T, in}} = 0.1 \text{ Hz} \).

The maxima deviate from the average value not more than 0.1 K and the minima not more than 0.2 K. Thus the average amplitude is 8.13 K which deviates 0.87 K from the set point.

The effects of the airflow temperature pulsation at a constant airflow velocity on the heat transfer are illustrated in figure 11. Figure 11(a) visualizes the position of the the local points for evaluation with a contour plot of the surface.
temperature at 50 s. Due to the longitudinal vortex system with main vortices, corner vortices as well as a horseshoe vortex, the surface temperature in the wake of the VG has a specific characteristic. It consists of two streaks of high temperature and a streak of lower temperature in between on each side of the symmetry plane. Position P1 is located in the upwash area between the main and the corner vortex. In this area the heat transfer is in principle lower than in the downwash areas of the vortexes where the evaluation point P2 is located. Therefore, downwash areas have a higher surface temperature than other areas and are visible behind the VG as high temperature streaks. Location P3 is in a region of lower heat transfer on the symmetry plane between the two main vortices.

Independent of the local position, \( T_w \) starts to increase as soon as \( T_{f,in} \) starts to increase from the temperature of thermal equilibrium to a higher temperature level (see figure 11(b)). Dependent on the local strength of the heat transfer, the strength of the local temperature rise of \( T_w \) varies. Due to the fact that P2 is located directly in the downstream region of the main vortex and close to the VG, \( T_w \) rises significantly faster than at P1 and P3. The strength of the \( T_w \) temperature rise at P1 and P3 is almost equal.

As the surface heat flux \( q_w \), shown in figure 11(c), is the root cause for the temporal change of \( T_w \), it changes the strongest where \( T_w \) also changes the strongest. In conclusion, at the beginning \( q_w \) shows the strongest increase at P2 and with superimposed pulsation of \( T_{f,in} \) also the strongest amplitude.

Figure 11(d) shows the histories of the calculated heat transfer coefficient \( h \) at the positions P1 to P3. Regardless of the position, \( h \) cannot be determined up to 14 s after the beginning of the experiment, since the driving temperature difference and \( q_w \) are zero. After 14 s \( q_w \) increases, but the driving temperature difference is still small, which leads to strongly fluctuating heat transfer coefficients. Smallest measurement errors have a strong effect on \( h \). After approximately 25 s the heat transfer coefficient stabilizes at a constant value. Here, the already discussed differences in the strength of the heat transfer become apparent. At P2, \( h \) reaches a value of 130 W \((m^2 \text{K})^{-1} \), while P3 has a value of 60 W \((m^2 \text{K})^{-1} \) and P1 a value of 56 W \((m^2 \text{K})^{-1} \), which is significantly lower.

With starting pulsation of \( T_{f,in} \), \( h \) changes in time. Due to the large change in flow temperature, two different phases of surface heat flux direction occur. A phase with a negative driving temperature difference results in a negative surface heat flux and a phase with a positive driving temperature difference results in a positive surface heat flux. During phases with a positive driving temperature difference \( h \) stabilizes for P1 and P3. However, for P2 \( h \) continuously changes.

5.2. Pulsating airflow velocity

The inlet conditions of the pulsating airflow experiment and the therefore required variation of \( \Omega \) and \( U_H \) are visualized in figure 12 up to 120 s. Figure 12(a) shows the defined set point variation of \( u_m \) and \( T_{f,in} \). Right at the beginning of the experiment \( u_m \) starts to pulse with a frequency of 0.05 Hz and an amplitude of 5 m s\(^{-1} \) around the mean inlet velocity of 15 m s\(^{-1} \). As for the pulsating airflow temperature experiment, \( T_{f,in} \) increases in an S-shaped form from ambient conditions to a value of 310.65 K after 25 s. When reaching a \( T_{f,in} \) of 310.65 K it keeps constant.

Figure 12(c) presents the calculated temporal variation of \( \Omega \) and \( U_H \) using the defined set point histories of \( u_m \) and \( T_{f,in} \) as well as the operating range of the experimental setup. Compared to the pulsing airflow temperature experiment, the required pulsation amplitude of \( \Omega \) increases. Furthermore, the non-sinusoidal shape of the \( \Omega \) pulsation visualizes the non-linear relationship between \( u_m \) and \( \Omega \).

When reaching the constant \( T_{f,in} \) of 310.65 K, the pulsation of \( u_m \) requires a significant pulsation of \( U_H \) too. This pulsation of \( U_H \) reaches with an amplitude of approximately 6 V the same range as for the pulsating temperature experiment.

The measured \( u_m \) and \( T_{f,in} \) are plotted in the graph of figure 12(b). Also for this experiment a good agreement between set point histories and measured histories exists. Minor deviations are visible for the extrema of \( u_m \). The average maximum of \( u_m \) is 20.25 m s\(^{-1} \) and the average minimum is 10.05 m s\(^{-1} \). These average values are met in a stable manner each pulsation period.

Furthermore, \( T_{f,in} \) does not reach a fully constant value of 310.65 K but pulsates slightly due to the interaction of changing velocity and electrical heater control.

Figure 13 visualizes the position of five different line profiles in the wake of the VG in order to present the influence of the pulsation airflow velocity on the heat transfer.

Figure 14 shows the corresponding heat transfer coefficient distributions along the five different line profiles and their variation with time for six different points in time. With figure 14(b) the illustration starts at the maximum \( u_m \) and ends with figure 14(g) one period further at the maximum \( u_m \).

In general and independent of the considered line, the heat transfer coefficient decreases with decreasing \( u_m \) and increases with increasing \( u_m \). In addition, for the lines 1 to 3 the shape of the \( h \)-profile is not influenced too. Each profile characterizes four main maxima and three main minima which vary in their absolute value and their \( y/W_{vg} \)-position. For the lines 4 and 5 the general shape changes with time and \( u_m \). At \( t = 55 \) s the \( h \)-profile of line 5 has a clear maxima at \( y/W_{vg} \approx 0 \). The \( h \)-profile of line 4 shows for same point in time and position a plateau. With increasing \( u_m \) the maxima of the \( h \)-profile of line 5 almost vanishes and the plateau of line 4 gets shorter. As the local maximum in heat transfer at line 5 in the center position occurs only at the maximum \( u_m \), it indicates a change in local flow field structures.

After a period, the same \( h \)-profiles result as at the beginning of the period.
Figure 12. Inlet airflow condition histories as well as control variable histories for the pulsing airflow velocity experiment.

Figure 13. Surface temperature distribution at 50 s and evaluation position of line profiles.
6. Conclusion

This paper summarizes two major contributions. First, an experimental setup for unsteady convective heat transfer experiments is presented that allows these experiments to be conducted in a deterministic and repeatable manner. Second, a measurement and evaluation methodology is discussed which provides measurement data with high accuracy and heat transfer evaluation data which is temporally and spatially resolved.

The airflow control system, consisting of a mesh heater and adjustable vanes, provides a simple and easy-to-implement...
setup to independently change the airflow temperature and the airflow velocity of a channel flow. The presented experimental results show that the precalculation of the temporal course of the control variables using the steady-state operating range works well. The deviations are smaller than 0.4 m s$^{-1}$ for $u_{in}$ and 0.2 K for $T_{in}$. Also, the pulsation frequencies set before the experiment are accurately matched. The described airflow control system does not completely succeed in keeping either $u_{in}$ constant during a $T_{in}$ change or vice versa. Some transient effects are not taken into account in the precalculation of the control variables, as e.g. the thermal inertia of the mesh heater. Further, the power supply units and the electric motor exhibit inertia in their control.

With the presented evaluation method, evaluation quantities of the heat transfer are obtained whose uncertainty of 10%–15% is comparable to other transient evaluation methods, for example the transient TLC method. The resolution of all evaluation values enables further analysis of the time dependent heat transfer characteristics.

**Data availability statement**

The data that support the findings of this study are available upon reasonable request from the authors.

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