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EXPERIMENTAL AND COMPUTATIONAL INVESTIGATION OF FLOW IN GAS TURBINE BLADE COOLING PASSAGES
Harald Roclawski, Jamey D. Jacob, Tiangliang Yang, & James M. McDonough
Mechanical Engineering Dept.
University of Kentucky

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EXPERIMENTAL AND COMPUTATIONAL INVESTIGATION OF FLOW IN GAS TURBINE BLADE COOLING PASSAGES

Harald Roclawski,∗ Jamey D. Jacob,† Tiangliang Yang,‡ & James M. McDonough§

Mechanical Engineering Dept.
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Results of experimental and numerical investigations into gas turbine cooling passage flows are presented. EXPAND ONCE ENTIRE PAPER IS COMPLETE.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tbody>
<tr>
<td>b</td>
<td>Bar height, 25.4mm</td>
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<tr>
<td>c_f</td>
<td>Skin friction coefficient</td>
<td></td>
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<tr>
<td>D</td>
<td>Channel width, 406mm</td>
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<tr>
<td>FFP</td>
<td>Forward flow probability</td>
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<tr>
<td>h</td>
<td>Backward facing step height, 28.6mm</td>
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<tr>
<td>H</td>
<td>Channel height, 203mm</td>
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<tr>
<td>Re</td>
<td>Reynolds number</td>
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<tr>
<td>U_∞</td>
<td>Freestream velocity</td>
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<tr>
<td>ν</td>
<td>Viscosity</td>
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<tr>
<td>ρ</td>
<td>Density, kg/m³</td>
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Introduction

Continuing requirements for ever increasing thrust/weight ratio in high-performance aircraft gas turbine engines necessitates increasing operating temperatures to gain improved thermodynamic efficiency. Even a fractional improvement in performance can offer significant savings (see, for example, Mayle†). Currently available materials for turbine blades are unable to withstand long periods of exposure to these high temperatures while maintaining structural integrity, even with thermal barrier coatings (TBC), implying need for active cooling strategies. Over the past 30 years turbine temperatures have been continually increased, and continuing improvements in cooling techniques have been a major contribution to this.

Several different approaches to cooling are usually employed in cooling even a single turbine blade, as can be inferred from a sample blade circuit shown in figure 1. In general, high-performance turbine blades are cooled by a combination of exterior flow film cooling which limits the heat flux from the combustion gases to the blade material and interior cooling air circuit flows which extract heat from the interior surfaces of the blade. A key factor in this interior heat removal process is maintaining turbulent flow. At the same time this results in greater pressure losses, so there must always be a tradeoff between cooling circuit heat transfer effectiveness and pressure loss in turbine blade design studies. A significant portion of the design effort is devoted to this problem, and it is especially difficult for both experimental and computational fluid dynamics (CFD) because of the highly complex geometries of interior cooling air circuits, the turbulent flow, and in the case of experiments, the need to account for high rates of blade rotation.

Past experiments on turbine blade cooling have fo-
cused on the flows within simple or complex channels and the effect of turbulence generating devices on the flow field and heat transfer rates. The effect of rotation, which is a requisite parameter in turbine flows, has generally been ignored due to the complexity of the experimental apparatus. This has proved to be a major gap in experimental testing. Also, the majority of experimental measurements have been single point measurements, not allowing a full field analysis. This point has proved difficult when trying to validate numerical simulations which benefit from full field data. The maturation of modern optical field measurements have allowed these hurdles to be overcome, though any experiments must require extreme care in their formulation.

In the case of numerical simulations, turbulence modeling based on Reynolds-averaged Navier-Stokes (RANS) approaches has generally proven inadequate for this problem, and typical Reynolds numbers \((Re)\) encountered in internal cooling air flows \((\sim \mathcal{O}(10^5))\) preclude use of direct numerical simulation (DNS) in most cases. But recent advances in computing power and in numerical procedures associated with large-eddy simulation (LES) suggest that certain forms of this approach may be applicable. However, a significant amount of laboratory experimentation is needed for their validation, and (probably) tuning. In the present paper we will focus on one particular aspect of constructing synthetic velocity models. We remark that there are many alternative approaches,\(^{15,16}\) and we shall not attempt a thorough review of the subject. Instead, we will concentrate on the Hylin and McDonough\(^{17}\) formalism.

**Previous Work**

Previous experimental research in turbine blade cooling has been primarily focused on the external cooling effects (e.g., Wang et. al\(^8\)). This is due to the relative ease of the external measurements as compared to the complex apparatus or gross simplifications required for internal duct measurements, particularly when it applies to measurements of the flow field. Previous research efforts on internal cooling have been limited in scope, typically focusing on a single aspect of the multivariable problem. Bunker and Metzger\(^3\) examined the local heat transfer from internal impingement cooling using temperature sensitive paint. General relations showed increased heat transfer with increased jet Reynolds number. Bohn et. al\(^4\) numerically and experimentally examined trailing edge cooling in turbine blades. The experiments were conducted in a scaled test rig and showed anisotropic turbulence profiles resulting in non-symmetrical coolant distribution. The numerical predictions compared reasonably well with the experimental data. Johnson et. al\(^5\) examined the heat transfer within rotating serpentine passages. Geometry and orientation were found to have large effects on the maximum local heat transfer. Specifically, the angle to which the passage was inclined respective to the axis of rotation was found to vary the heat transfer ratio up to 50%. Flannery et. al\(^6\) examined the heat transfer properties of various heat transfer enhancement devices within a cooling circuit using napthalene sublimation. Dimensional analysis and scale modeling showed improvement in heat transfer in vortex flow cavities and sand-dune shaped turbulators, though actual measurements in a channel or rotating cavity were not performed. Morris and Chang\(^7\) investigated the heat transfer properties of a circular cooling channel. Full field heat transfer data were obtained through a combination of measurements and solution of the channel wall heat conduction equation. The resulting internal heat flux distribution over the full inner surface was subsequently used to determine the local variation of heat transfer coefficient. Effects of the Coriolis force and centripetal buoyancy on the forced convection mechanism were investigated and found to be of sufficient order to warrant consideration in further experimental and numerical studies.

Cakan and Arts\(^8\) studied the flow in a straight, rectangular, rib-roughened internal cooling channel. At a Reynolds number of 6500 and 30000 and a rib blockage ratio of 0.133, DPIV measurements were taken. They found that the flow through the ribbed channel can be characterized by a series of accelerations, decelerations with separation, reattachment and redevelopment due to the sudden changes in cross-section. The ribs induce a separation and recirculation bubble. The flow reattaches at \(X/e = 4.5\) \((Re = 6500)\). Comparing both studies, they claim that the reattachment distance is not strongly dependent on the Reynolds number. Upstream of the ribs, the flow impinges on the rib, moves to the sidewalls of the channel and produces to vortices. Behind the rib a similar motion occurs due to the recirculation region. In the span wise flow direction; two counter rotating secondary flow cells are observed.

The flow in a straight, rectangular channel with ribs on two opposite walls was investigated by Lio et. al\(^9\) by means of LDV. The Reynolds number based on the channel hydraulic diameter was 33000. The ribs were perforated and the effect of the rib open area ratio was investigated. They also found a periodic accelerating and decelerating flow behavior. In contrast to the previous paper, only one secondary flow cell is observed in the span wise flow direction. Furthermore, it was discovered that the reattachment length downstream of a rib pair is shorter than in the case of a backward-facing step. The maximum heat transfer rate was found to be dependent upon a critical range of the open area ratio governed by whether the flow treated the ribs as permeable or impermeable. A PIV Investigation of the flow in a rectangular channel with a 90° and 45° rib arrangement and a 180° bend was done by Schabacker and Bölcs\(^10\) at \(Re = 45700\) and a rib height equal to
0.1 hydraulic diameters. Two counter rotating vortices in the span wise flow direction were observed. The development length to achieve a fully developed flow condition is longer for the case of a 45° rib arrangement. Furthermore, the 45° ribs prevent the development of zones of recirculating flow in the upstream outer corner of the bend and the curvature-induced secondary flows a weakened in this section of the channel. Compared to a smooth channel, the flow recovers faster from the bend effect. Results for the case of a stationary and rotating, rectangular, ribbed channel with a 180° bend were obtained by Servouze\textsuperscript{11} using LDV. The flow conditions were Re=5000, Ro=0.33 and a rib aspect ratio of 10. In the stationary case a periodic accelerating and decelerating flow behavior was found. In contrast to other papers, secondary flow structures (vortices) in the span wise flow direction were not observed. Iacovides\textsuperscript{12} did an LDA study on the flow in a ribbed channel with a 180° bend. He investigated a stationary case at a Re=100000 and two rotating cases at Ro=(+/-)0.2. The rib-height to duct diameter ratio was 0.1. He also observes a periodic flow behavior. Because of the ribs, turbulence increases at the bend entry and an additional separation bubble over the first rib interval downstream of the bend exit is formed. Nevertheless, in agreement with Schabacker and Bölcs, it is claimed that the flow recovers faster from the bend effect in a ribbed channel. Especially the separation bubble along the inner wall is reduced. Lastly, Hwang and Lai\textsuperscript{13} examined laminar flow within a rotating multiple-pass channel with bends from a computational standpoint. Rota-
tion was found to have a large impact on the wall friction factor. Validations were only made with stationary experiments, however. Heat transfer rate or turbulators were not examined.

**Previous Modeling Efforts**

Put previous modeling in here.

**Experimental Portion**

**Setup & Diagnostics**

The wind tunnel arrangements for the backward facing step and turbulator experiments are shown in figure 11(a) and 11(c), respectively. Both test sections were installed in a low-turbulence open-circuit blow-down wind tunnel. A 7.5 hp motor powers a radial fan at the inlet. A vibration damper, flow straightener, and turbulence dampening screens precede the nozzle which has a contraction ratio of 6.7. The maximum test section velocity is 35 m/s with an open exhaust and approximately 10 m/s when a filter is installed to capture seeding particles. The test sections have a channel height $H$ of 0.2 m (203 mm) and width $D$ of 0.4 m (406 mm). The backward facing step has a step height $h$ of 28.6 mm. The turbulator test section is arranged so that multiple square ribs of various sizes can be placed in different locations in the channel. The current paper presents results for ribs of equal sizes with sides $b = 24.5$ mm. Ribs arrangements of 1, 2, 3, and 4 bars are examined with equal separation distances of 152 mm in the multiple turbulator runs. Re based on channel height, step size, and rib size are reported in table 1.

In both the backward facing step and turbulator test sections, PIV, HiWA, and static pressure measurements were made along the tunnel centerline downstream of the step and last rib, respectively. For PIV, the laser sheet was generated by a 25 mJ double-pulsed Nd:YAG laser with a maximum repetition rate of 15 Hz. Pulse separations varied from 100 $\mu$s to 1 ms based upon the tunnel velocity. A 10 bit CCD camera with a 1008 x 1018 pixel array was used to capture images. Uniform seeding was accomplished using either zinc stearate or talc injected at the fan inlet; a 1 micron filter at the tunnel exhaust was required to capture the particles thus reducing the maximum tunnel velocity during PIV. A predictor-corrector algorithm with an interrogation area of 32 x 32 was used to generate displacement vectors and velocity gradients.\textsuperscript{13} For the hot-wire measurements, which are required to obtain details of random fluctuations in velocity for the modeling efforts, a single-component boundary layer hot-wire probe was used capable of wall measurements within 0.3 mm of the surface. 40,000 points or greater were recorded at a sampling rate 10 kHz. For the static pressure measurements, pressure taps were placed 25.4 mm apart downstream of the backward facing step and slightly upstream and downstream of the last turbulator. Pressure measurements were made up to 14.5 $b$ downstream of the turbulator in the results presented here.

**Results**

**Backward Facing Step**

- insert discussion files

**Turbulators**

- insert discussion files

**Discussion**

**Modeling**

In the present paper we will focus on one particular aspect of constructing synthetic velocity models. We remark that there are many alternative approaches,\textsuperscript{15,16} and we shall not attempt a thorough
review of the subject. Instead, we will concentrate on the Hylin and McDonough formalism.

We begin by noting that synthetic velocity turbulence models offer the potential of a much closer connection to laboratory measurements than can be obtained with other modeling approaches because primitive variables (e.g., velocity components) are directly modeled and used instead of attempting to model flow statistics—Reynolds stresses.

Within the Hylin and McDonough framework, a subgrid-scale quantity is expressed as a product of three factors, e.g.,

\[ u^* = A_u \zeta_u M_u. \]  

Here \( u^* \) is a SGS velocity component; \( A_u \) is an amplitude factor; \( \zeta_u \) is an anisotropy correction, and \( M_u \) is a temporal fluctuation. Each of the three factors on the right-hand side of (1) varies across the spatial grid, and from one resolved-scale time step to the next.

Expressions for the first two factors have been derived from first principles employing the Kolmogorov theory of homogeneous turbulence (see Frisch, for a good overview), as given in Hylin and McDonough and Sagaut. The third factor received little specific attention in early investigations. Hylin associated this factor with Kolmogorov’s “stochastic variable,” and considered the logistic map, “absolute value” logistic map and tent map for realizations. All appear to give similar results when the same map is used for all velocity components.

Two-dimensional simulations of turbine blade cooling presented by McDonough et al. suggest that using the same map for all solution components is not correct (especially for passive scalars). McDonough and Huang provide a derivation of appropriate maps for use in (1) for reduced-kinetics \( H_2-O_2 \) combustion and show that the maps (discrete dynamical systems, DDSs) arising from the Navier-Stokes equations are basically logistic maps while those corresponding to thermal energy and species concentrations are not. Finally, McDonough and Huang present a detailed analysis of the 2-D “poor man’s N.–S. equations” derived as in the previous reference. They show that essentially every temporal behavior seen in actual N.–S. flows can be produced by this simple 2-D DDS:

\[
\begin{align*}
a^{(n+1)} &= \beta_1 a^{(n)} (1 - a^{(n)}) - \gamma_1 a^{(n)} b^{(n)}, \\
b^{(n+1)} &= \beta_2 b^{(n)} (1 - b^{(n)}) - \gamma_2 a^{(n)} b^{(n)},
\end{align*}
\]

where \((a, b)^T\) can be viewed as high-wavenumber Fourier coefficients of the velocity field, \( \mathbf{U} = (U, V)^T \).

This DDS contains four bifurcations parameters \((\beta_1, \beta_2, \gamma_1, \gamma_2)\). The first two are directly related to the Reynolds number, \( Re \), or possibly the integral scale or Taylor microscale \( Re \), depending on details of implementation, and thus might reasonably be set equal as in McDonough and Huang. The other two parameters are most closely associated with shear stress: \( u_p \) and \( v_x \) in 2D. As noted in McDonough and Huang, in order to construct reliable synthetic velocities by employing nonlinear DDSs it is essential to find a mapping between physical parameters, say \( Re \) and \( \nabla \mathbf{U} \) and the above bifurcation parameters of the map(s). The present paper documents our initial attempts to accomplish this for a specific flow geometry.

We begin by noting that in order to establish such a mapping we must first be able to accurately express any appropriate set of physical data in terms of bifurcation parameters of the DDS. This is nothing more than a curve-fitting problem, and McDonough et al. and Mukerji et al. provided an approach to solving it. This consists of first recognizing that an “exact” polynomial fit is not appropriate (cf., Casdagli and Eubank, for examples where it is appropriate), and that a global least-squares fit is needed. The next step is to determine a set of data characterizations by means of which to compare the fit with the original data. McDonough et al. provide a long list of these but make no claims as to either the necessity of any single one, or sufficiency of the set as a whole.

Once such a set of characterizations is chosen, we construct the model in the form given in McDonough et al.

\[
M^{(n+1)} = (1-\theta)M^{(n)} + \theta \sum_{k=1}^{K} \alpha_k S_k \left( d_k \omega_k, m_k(\beta_k) \right) .
\]

Details of this expression can be found in the cited references, but we briefly note the following. The parameters \( \theta, \alpha_k, \beta_k, \omega_k, d_k \) and \( K \) must all be found in the course of the curve-fitting process. The \( \alpha_k \) are amplitude factors and can be considered analogous to Fourier coefficients (although they do not arise from a formal inner product). \( S_k \) is a nonlinear discontinuous function of the behavior of the map; its purpose is to allow existence of frequencies of oscillation not found in the DDS, itself (i.e., lower frequencies) in addition to permitting modeling of very irregular intermittencies. The parameters \( \omega_k \) are the frequencies at which iterations of the \( k^{th} \) map will be initiated; \( d_k \) is the duration of evaluation once initiated; \( \beta_k \) is the bifurcation parameter, and for purposes herein, the form of the \( k^{th} \) map is

\[
m^{(n+1)} = \beta_k m^{(n)} (1 - m^{(n)}),
\]

the basic logistic map (see, e.g., May, 1976, for an interesting and detailed treatment).

We will use the scalar map (3) in the present study, rather than the 2-D map described above, because the temporal resolution of the current 2-D PIV data is insufficient for capturing details of subgrid-scale turbulent fluctuations. On the other hand, the single
velocity component obtained from hot-wire measurements at 10 KHz is sufficient. Our goal is thus to determine the variation of each of the parameters in Eq. (2) at a specific location in the physical flow field as a function of $Re$. 

For each value of $Re$ considered we will construct two separate curve fits of the data. The first will be based on the complete velocity signal and thus, up to subtracting out the mean, corresponds to a Reynolds-averaged N.–S. (RANS) quantity:

$$u'(x, t) = U(x, t) - \bar{u}(x),$$

where the overbar denotes time average. The second will be obtained from high-pass filtered data analogous to the SGS behavior in a LES formalism. In this case

$$u^*(x, t) = U(x, t) - \tilde{u}(x, t),$$

where tilde denotes (formally) a spatial filtering.

Both cases are of interest because the ability to fit both types of data shows that synthetic velocity fields are applicable to both LES and RANS models. But in addition, with both fits available we will see that they are very different—as one should expect. This raises very serious questions regarding validity of certain forms of “very” large-eddy simulation (VLES) in which “time-accurate” RANS equations are solved, and conversely to forms of LES in which RANS formalisms are employed to construct the SGS models. At the fundamental mathematical level RANS and LES are quite different, and the results we present below provide a direct empirical demonstration of this. Thus, considerable care must be taken when attempting to bridge the gap between these two modeling approaches.

The cases we consider correspond to $Re = 4 \times 10^4$ and $Re = 1 \times 10^5$ for the turbulator flows discussed above in the section on experiments. For both values of $Re$ the measurement location was $27$ mm upstream of reattachment, and at a height of $3.5$ mm, behind the first turbulator. The curve-fitting process was carried out essentially as described in McDonough et al. and Mukerji et al., and as in those references once the process is complete we compare three additional features: i) the “appearance” of the time series, ii) the power spectral density, and iii) the delay map.

As emphasized in these references, we consider the appearance of the modeled time series to be essential to a good fit of the data (just as would be the case in an exact fit), and the characterizations employed in the least-squares fit are selected to guarantee this. But appearance in this case is a nontrivial notion, first because it is difficult to define precisely, and second because the objective function for the least-squares fit is highly nonlinear, discontinuous, contains both real and integer variables, and thus generally has multiple local minima.

McDonough et al. discuss appearance of the time series in some detail, indicating that close examination of most such data will lead to identification of a finite number of types of “structures.” The number of these is typically used as the initial guess for the value of $K$, the number of terms in the model representation, Eq. (2). In comparing the model with the data it is important to observe, as already noted, that the fit is not intended to be exact because an exact fit is intrinsically incompatible with the physics (and mathematics) of a turbulent flow. Our rule of thumb for checking that the model has preserved the appearance of the data is that if the modeled time series and data are juxtaposed, it should be impossible to determine where the data ends and the model begins.

Use of power spectra and delay maps as additional tests of goodness of fit is recommended because appearance is at best only semi-quantitative. On the other hand, neither of these characterizations alone will guarantee a good fit. Many different time series exhibit very similar power spectra, and the delay map provides only basic topological information associated with the underlying attractor (if there is one). Nevertheless, both of these can be of value in “fine tuning” a fit of data.

**Results for Complete Velocity**

Figure ? displays three complete velocity time series in parts (a) through (c). The first corresponds to experimental data for $Re = 4 \times 10^4$; the second represents evaluation of the model, Eq. (2), with and initial guess of the parameters to be determined in the least-squares fit, and the third shows the final fit of data. We display part (b) of the figure to emphasize that finding the correct parameters is a nontrivial process (and one which requires considerable CPU time). Table ? displays the initial guess of the parameter values and the associated objective function value, as well as the same data for the final fit, for both values of $Re$ considered. Part (c) of the figure indicates a significant improvement over the initial guess, and in fact a fairly good representation of the data.

Figures ?? and ??? display comparisons of power spectra and delay maps, respectively, with part (a) corresponding to measured data and part (b) to modeled results in both cases.

Similar results are presented in Figs. ?, ?, and ??? for the $Re = 1 \times 10^5$ case, except that we have not shown the initial guess result in Fig. ??.

**Results for High-Pass Filtered Velocity**

[Repeat above figures (no Fig. ?(b)); indicate suitability for LES SGS models. Discuss $Re$ dependence of data.]

[Compare model results in the two cases]
emphasizing differences between complete and high-pass filtered behaviors—e.g., variation of parameters with $Re$.]

Discussion

Continuing Work

Acknowledgements

This work is supported by AFOSR grant F49620-00-1-0258 under the supervision of Dr. Tom Beutner.

REFERENCES

Note: CHECK ALL REFERENCES HERE AND IN TEXT FOR CONSISTENCY, ACCURACY, and TIMELINESS. THEY WILL BE UNIFORMLY REFORMATTED ONCE COMPLETE.

References


a) Backward facing step geometry.

b) Turbulator geometry.

Fig. 2 Channel geometry for experimental studies.
Fig. 3  Channel geometry for experimental studies.
Fig. 4 Channel geometry for experimental studies.
a) Backward facing step geometry.

b) Turbulator geometry.

Fig. 5 Channel geometry for experimental studies.
Fig. 6 Channel geometry for experimental studies.
Fig. 7 Channel geometry for experimental studies.
Fig. 8 Channel geometry for experimental studies.
Fig. 9 Channel geometry for experimental studies.
Fig. 10 Channel geometry for experimental studies.
a) Backward facing step geometry.  
b) Turbulator geometry.  

c) Turbulator geometry.  

Fig. 11 Channel geometry for experimental studies.
Fig. 12  Backward facing step PIV: $U_\infty = 0.89$ m/s, $Re = 1,702$. 

a) Velocity.  
b) Vorticity.  
c) RMS turbulence intensity.  
d) Forward flow probability.
Fig. 13  Backward facing step PIV: $U_\infty = 2.56$ m/s, $Re = 4,880$.  

a) Velocity.  

b) Vorticity.  

c) RMS turbulence intensity.  

d) Forward flow probability.
Fig. 14  Backward facing step PIV: $U_\infty = 7.69$ m/s, $Re = 14,657$.

Fig. 15  Detail of reattachment for backward facing step: $U_\infty = 2.56$ m/s, $Re = 4,880$.
a) Velocity.

b) Vorticity.

c) RMS turbulence intensity.

d) Forward flow probability.

Fig. 16  Turbulator - single rub PIV: $U_\infty = 0.71$ m/s, $Re = 1,350$. 
June 7, 2001

CFD figures to go here.