Challenges in Measuring the HTC for Turbine Cooling

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DoE – NETL & Ames Laboratory
Our research group’s focus is on turbine cooling. 

Current efforts include:

• LES of ingress/egress through rim seals that accounts for rotor-stator interactions.
• LES of internal & film cooling aimed at generating statistics to guide the development of RANS models.
• Inflow BC for LES and BC at the interface between LES and RANS for hybrid methods.
• Develop reduced-order methods from CFD for system-level tools.
• Examine fundamental issues in computing & measuring heat-transfer relevant to turbine cooling.

Liu & Shih

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Outline of Talk

Revisit
- HTC measured by transient methods
- HTC measured by steady-state methods

What else?
- $\text{Nu} = F(\text{Re}, \text{Pr}, \text{geometry})$?

Summary & Current Work

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Measurements of HTC by Transient Methods

Measurements are needed to validate CFD tools.

A widely used method is the **thermochromic liquid crystal technique**.

Surface temperature is measured as a function of time.

The heat transfer coefficient is then inferred from either 1-D or 0-D exact solutions.

**1-D Model**

- $T(x, t)$ initially uniform at $T_{\text{initial}}$
- Semi-infinite solid

**Assumptions:**
- constant $h$, $k$, $C_p$
- constant $T_{\text{bulk}}$
- $h_{\text{transient}} = h_{\text{steady-state}}$

**Experimental Technique:**
- Measures $T = f(t)$
- Calculate $h$ at time when $T = 37.6 \, ^\circ\text{C}$

**1-D Model**

\[
\frac{T_{\text{wall}}(t) - T_{\text{initial}}}{T_{\text{bulk}} - T_{\text{initial}}} = 1 - \exp \left[ \frac{h^2 \alpha t}{K^2} \right] \text{erfc} \left[ \frac{h \sqrt{\alpha t}}{k} \right]
\]

\[
\frac{T_{\text{wall}}(t) - T_{\text{initial}}}{T_{\text{initial}} - T_{\text{bulk}}} = \exp \left[ -\frac{h A_s}{\rho V C_p} t \right]
\]

**Two equation method**

\[
\frac{T_{w_1} - T_i}{T_b - T_i} = 1 - \exp \left[ \frac{h^2 \alpha t_1}{K^2} \right] \text{erfc} \left[ \frac{h \sqrt{\alpha t_1}}{k} \right]
\]

\[
\frac{T_{w_2} - T_i}{T_b - T_i} = 1 - \exp \left[ \frac{h^2 \alpha t_2}{K^2} \right] \text{erfc} \left[ \frac{h \sqrt{\alpha t_2}}{k} \right]
\]

- Constant $T_{\text{bulk}}$, Variable HTC
- Variable $T_{\text{bulk}}$ and HTC

**Schematic of a modern gas turbine blade with common cooling techniques**

- Impingement cooling
- Rib turbulators
- S-shaped internal cooling passage
- Finned cooling

$T_{\text{bulk, inlet}} = 350 \, \text{K} = 67.85 \, ^\circ\text{C}$
$T_{\text{surface, initially}} = 300 \, \text{K} = 26.85 \, ^\circ\text{C}$

‘$h$’ can be calculated with known values of $T_{\text{bulk}}$.
‘$h$’ & ‘$T_{\text{bulk}}$’ can be found by solving the two equations.
HTC: 1-eq vs 2-eq vs URANS & RANS at the time $T_{\text{surface}} = 37.6 \, ^\circ C$

Error due to 3-D variation
- $< 4\%$ for 1-eq
- up to $30\%$ for 2-eq.

Error due to unsteady $T$ on wall
- up to $16\%$

$E-\text{URANS} = (h-\text{URANS} - h-\text{RANS}) / h-\text{RANS}$

$E-\text{1DTb} = (h-\text{1DTb} - h-\text{URANS}) / h-\text{URANS}$

$E-\text{1D} = (h-\text{1D} - h-\text{URANS}) / h-\text{URANS}$
Measurements of HTC by Transient Methods

- HTC measured by transient methods differ from the HTC under steady-state conditions with isothermal walls (up to 17% for the pin-fin problem and up to 26% for the rib problem).
- HTC measured by the transient methods vary appreciably in time (up to 31% over 10 seconds). Thus, it is unclear which HTC measured is the correct or the meaningful one.

Sathyanarayanan, Ramachandran, Shih, et al. (AIAA 2015-1195)
Issues:

• Lab vs engine-relevant conditions? Will data from lab conditions be useful?

• Since HTC = f(wall BC: T or q”), what are the effects of imposing constant q”, which is what is typically used?

• What are the effect of the plate thickness and its material properties on the HTC?
**Problem Description**

**Cooling condition**

\[ \text{Re}_D = 50,000 \]
\[ T_c = 400 \degree C \]
\[ P_b = 25 \text{ atm} \]

\[ \text{Case Heating BCs} \]

<table>
<thead>
<tr>
<th>Case</th>
<th>Heating BCs</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>HTC_hot = infinity ( (T_{wall}=T_{hot}=1755K) )</td>
</tr>
<tr>
<td>II</td>
<td>HTC_hot = average of HTC_cold</td>
</tr>
<tr>
<td>III</td>
<td>specified ( q'' ) without HTC_hot</td>
</tr>
<tr>
<td>IV</td>
<td>specified ( q'' ), isothermal, or HTC_hot with different Bi (k or t/D varies)</td>
</tr>
</tbody>
</table>

- \( h/D = 1 \)
- \( S_x/D = 2.5 \)
- \( S_y/D = 5 \)
- \( t/D = 0, 0.5, 1 \)
### Heating BCs on Hot-gas Side

<table>
<thead>
<tr>
<th>Case</th>
<th>Description</th>
<th>HTC_hot = infinity</th>
<th>T&lt;sub&gt;w,hot&lt;/sub&gt; = 1,273 K</th>
<th>T&lt;sub&gt;w,hot&lt;/sub&gt; = 1,755 K</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case I</td>
<td>const. temperature</td>
<td>HTC_hot = infinity</td>
<td>T&lt;sub&gt;w,hot&lt;/sub&gt; = 1,273 K</td>
<td>T&lt;sub&gt;w,hot&lt;/sub&gt; = 1,755 K</td>
</tr>
<tr>
<td>Case II</td>
<td>HTC_hot = average of HTC_cold</td>
<td>h&lt;sub&gt;h&lt;/sub&gt; = 100 W/m&lt;sup&gt;2&lt;/sup&gt;-K, T&lt;sub&gt;hot&lt;/sub&gt; = 1,755 K</td>
<td>h&lt;sub&gt;h&lt;/sub&gt; = 1,000 W/m&lt;sup&gt;2&lt;/sup&gt;-K, T&lt;sub&gt;hot&lt;/sub&gt; = 1,273 K</td>
<td>h&lt;sub&gt;h&lt;/sub&gt; = 1,000 W/m&lt;sup&gt;2&lt;/sup&gt;-K, T&lt;sub&gt;hot&lt;/sub&gt; = 1,755 K</td>
</tr>
<tr>
<td>Case III</td>
<td>specified q&quot; without HTC_hot</td>
<td>adiabatic</td>
<td>q”&lt;sub&gt;w,hot&lt;/sub&gt; = 0 W/m&lt;sup&gt;2&lt;/sup&gt;</td>
<td>q”&lt;sub&gt;w,hot&lt;/sub&gt; = 253,012 W/m&lt;sup&gt;2&lt;/sup&gt;</td>
</tr>
<tr>
<td>Case IV</td>
<td>specified heating BCs with Bi=0.01, 0.1, 1 (varied k)</td>
<td>const. heat flux</td>
<td>Bi=0.01</td>
<td>Bi=0.1</td>
</tr>
<tr>
<td></td>
<td>(q”=462,483 W/m&lt;sup&gt;2&lt;/sup&gt;)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>const. temperature</td>
<td>Bi=0.01</td>
<td>Bi=0.1</td>
<td>Bi=1</td>
</tr>
<tr>
<td></td>
<td>(T&lt;sub&gt;hot&lt;/sub&gt; = 1,755 K)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>const. convective environment</td>
<td>Bi=0.01</td>
<td>Bi=0.1</td>
<td>Bi=1</td>
</tr>
<tr>
<td></td>
<td>(h=600 W/m&lt;sup&gt;2&lt;/sup&gt;-K, T&lt;sub&gt;hot&lt;/sub&gt; = 1,755 K)</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Temperature Distribution w/ Bi=0.01, 0.1, and 1

- Bi = 0.01
- Bi = 0.1
- Bi = 1

Isothermal constant q'

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HTC on Cold Wall w/ Bi=0.01, 0.1, and 1

- Isothermal
- Constant q"
Nu on Cold Wall w/ Bi=0.01, 0.1, and 1

isothermal

constant $q"$

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Relative Error Distribution of HTCs on Cold Wall w/ Bi=0.01, 0.1, and 1

$\varepsilon = \frac{|h_{Bi=0.01 \text{ at } 1755K} - h_{isothermal \text{ at } 1755K, Bi=0.01}|}{h_{isothermal \text{ at } 1755K, Bi=0.01}}$

$\varepsilon = \frac{|h_{Bi=0.1 \text{ at } 1755K} - h_{isothermal \text{ at } 1755K, Bi=0.01}|}{h_{isothermal \text{ at } 1755K, Bi=0.01}}$

$\varepsilon = \frac{|h_{Bi=1 \text{ at } 1755K} - h_{isothermal \text{ at } 1755K, Bi=0.01}|}{h_{isothermal \text{ at } 1755K, Bi=0.01}}$

$\varepsilon = \frac{|h_{Bi=0.01 \text{ at } 1755K} - h_{isothermal \text{ at } 1755K, Bi=0.1}|}{h_{isothermal \text{ at } 1755K, Bi=0.01}}$

$\varepsilon = \frac{|h_{Bi=0.1 \text{ at } 1755K} - h_{isothermal \text{ at } 1755K, Bi=0.1}|}{h_{isothermal \text{ at } 1755K, Bi=0.01}}$

$\varepsilon = \frac{|h_{Bi=1 \text{ at } 1755K} - h_{isothermal \text{ at } 1755K, Bi=0.1}|}{h_{isothermal \text{ at } 1755K, Bi=0.01}}$

isothermal

constant q’’
Zonally Averaged HTC w/ Bi=0.01, 0.1, and 1

 Isothermal

 Constant $q''$

 $T_{w,h} = 1755$ K

 $q''_{w,h} = 462.483$ W/m$^2$

 DoE – NETL & Ames Laboratory
Relative Error of Zonally Averaged HTCs w/ Bi=0.01, 0.1, and 1

\[
\varepsilon = \left| \frac{\bar{h} - \bar{h}_{\text{isothermal,1755K}}}{\bar{h}_{\text{isothermal,1755K}}} \right|
\]

- **Isothermal**: \( T_{w,h} = 1755 \, K \)
- **Constant q'':** \( q'_{w,h} = 462,483 \, W/m^2 \)

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Total Averaged HTC & Nu of the Entire Endwall Section

\[ h = \frac{1}{A_t} \int_{A_0} h(x, y) dA \]

\[ Nu = \frac{1}{A_t} \int_{A_0} Nu(x, y) dA \]

HTC

Nu

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Temperature Distribution w/ t//D=0, 0.5, and 1 (isothermal)

Temperature

\( \text{Nu} \)

HTC

PURDUE UNIVERSITY
Energy-balance $T_{\text{bulk}}$ v.s. linear $T_{\text{bulk}}$

For this problem, approximating $T_{\text{bulk}}$ by linear interpolation is OK because the variation in $T$ is small.
Revisit
• HTC measured by transient methods
• HTC measured by steady-state methods

What else?
• Bulk Temperature?
• $\text{Nu} = F(\text{Re}, \text{Pr}, \text{geometry})$?

Summary & Current Work
Bulk temperature is almost always approximated in experiments but rarely documented.

- What are the consequences of the approximations?
- How should HTC be measured and used in V&V and in design?

$T_b$ is hard to measure. Why?
- an integral!
- Which cross section around the bend?
- How to handle flow separation, ribs, pin fins, …?

Chi & Shih (AIAA 2012-0807)

In turbine cooling, the local Re can vary appreciably along the duct.

\[ T_b = \frac{A \cdot u C_p T dA}{A \cdot u C_p dA} \]

\[ \text{Re} = \rho_{in} V_{in} D_n / \mu, \quad \mu = f(T_b) \]

\[ q'' = h(T_w - T_b) \]

\[ \text{Nu} = hD/k, \quad k = k(T_b) \]

Shih, et al. (ASME HT-2013-17114)
Problem Description

Cases | cooling condition | heating load |
--- | --- | --- |
1 | | q" = 10 KW/m² |
2 | Re_D = 5,000 | q" = 100 KW/m² |
3 | T_c = 400 °C | q" = 200 KW/m² |
4 | P_b = 25 atm | q" = 400 KW/m² |
5 | | q" = 10 KW/m² |
6 | Re_D = 10,000 | q" = 100 KW/m² |
7 | T_c = 400 °C | q" = 200 KW/m² |
8 | P_b = 25 atm | q" = 400 KW/m² |

\[ Re_D = Re_{D,x} \text{ at } x = D, \quad Re_{D,x} = \frac{\rho x U_x D}{\mu(T_b,x)} \]
Dimensionless Bulk Temperature

\[ S_x/D = 2.5, \ S_y/D = 2.5, \ H/D = 1 \]

\[ \phi_{b,x} = \frac{T_{b,x} - T_{c,in}}{T_{c,in}} \]

\[ T_{c,in} = 400^\circ C \]

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Local Reynolds Number = \( \text{Re}(T_b) \)

\[ S_x/D = 2.5, \ S_y/D = 2.5, \ H/D = 1 \]

\[ \text{Re}_{D,x} = \frac{\rho_x U_x D}{\mu(T_b)} \]

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Local Prandtl Number = $\text{Pr}(T_b)$

$S_x/D = 2.5$, $S_y/D = 2.5$, $H/D = 1$

$\text{Pr} = \frac{\mu(T_b)C_P(T_b)}{k(T_b)}$

$\text{Pr}_t = \frac{\mu_tC_P(T_b)}{k_t}$

$Re_D = 5,000$

$Re_D = 10,000$

$q'' = 10 \text{ KW/m}^2$
$q'' = 100 \text{ KW/m}^2$
$q'' = 200 \text{ KW/m}^2$
$q'' = 400 \text{ KW/m}^2$

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Lateral Averaged Nusselt Number

\[
\overline{Nu_i} = \frac{\bar{h}_i D}{k}
\]

Regional average about each row (i) of pins in streamwise direction

\[\text{Re}_D = 5,000\]

\[\text{Re}_D = 10,000\]

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Mean Nusselt numbers ($\overline{Nu}$) for the entire ten row array as functions of the mean Reynolds numbers ($\overline{Re_D}$)

\begin{align*}
\overline{Nu} &= C \overline{Re_D^M} \\
\text{Exp. Metzger, 1982} & \quad \overline{Nu} = 0.069 \overline{Re_D}^{0.73} \\
& \quad \overline{Nu} = 0.286 \overline{Re_D}^{0.59} \\
& \quad \overline{Nu} = 0.226 \overline{Re_D}^{0.61} \\
& \quad \overline{Nu} = 0.115 \overline{Re_D}^{0.68} \\
& \quad \overline{Nu} = 0.009 \overline{Re_D}^{0.95}
\end{align*}
Measuring HTC by transient methods could have some issues.

Measuring HTC by steady-state methods is relatively independent of the BC imposed on the wall surface.

The Re in an internal cooling passage may vary appreciably along the passage because of the rise in coolant temperature from 400 °C to 700 or 800 °C.

HTC is currently obtained under conditions, where $T_w/T_b$ is near unity. Thus, scaling of HTC is needed. The question is how?