Improving Durability of Turbine Components Through Trenched Film Cooling and Contoured Endwalls

DOE Award Number DE-FE0005540
UTSR Project Number 07-01-SR127

Principal Investigator:
Prof. David G. Bogard
University of Texas at Austin
Graduate Research Assistant:
David Kistenmacher

Co-Principal Investigator:
Prof. Karen A. Thole
Pennsylvania State University
Graduate Research Assistant: **Amy Mensch**

UTSR Workshop, October 2-4, 2012

Background image: [Hamed, A., Tabakoff, W., and Wenglarz, R., 2006]

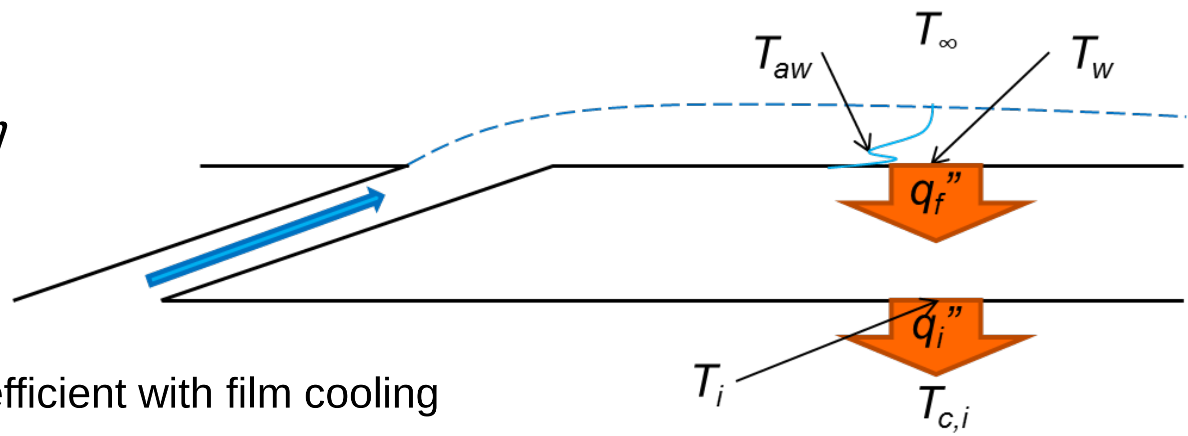
Tasks completed this year for the University of Texas and Penn State

Design and test realistic trench configuration embedded in TBC	Completed Q5-6
Determine the effect of depositions on overall cooling effectiveness with trench configurations	Completed Q7
Develop conducting endwall model	Completed Q5-6
Measure overall effectiveness with and without film cooling	Completed Q7

Measurement of ϕ requires a matched Biot number model

A simplified 1-D analysis using T_{aw} as the driving temperature shows:

$$\phi = \frac{T_\infty - T_w}{T_\infty - T_{c,i}} = \frac{1 - \chi\eta}{1 + Bi + \frac{h_f}{h_i}} + \chi\eta$$



where

$$Bi = \frac{h_f t}{k}$$

h_f = heat transfer coefficient with film cooling

t = wall thickness

k = conductivity of the solid

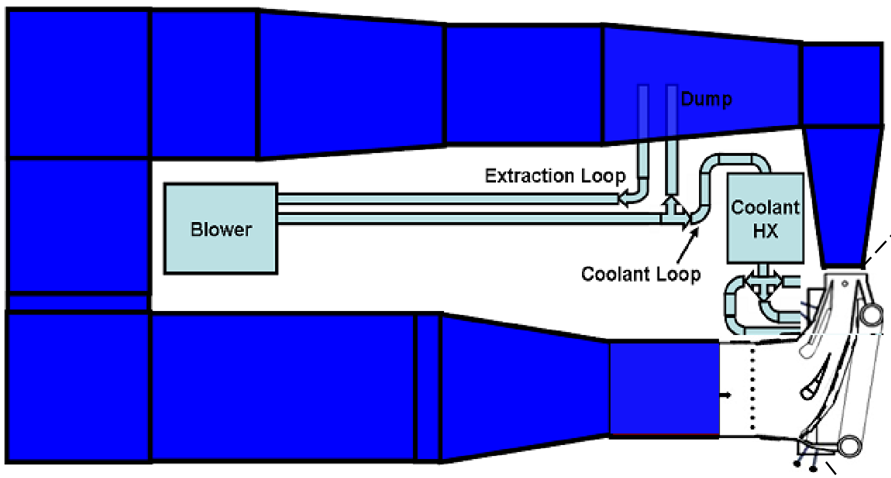
$$\chi = \frac{T_\infty - T_{c,hole\ exit}}{T_\infty - T_{c,vane\ inlet}}$$

It is also important to match h_f/h_i, the ratio of internal to external heat transfer coefficients. Though the magnitudes of both are much lower, the ratio is the same.

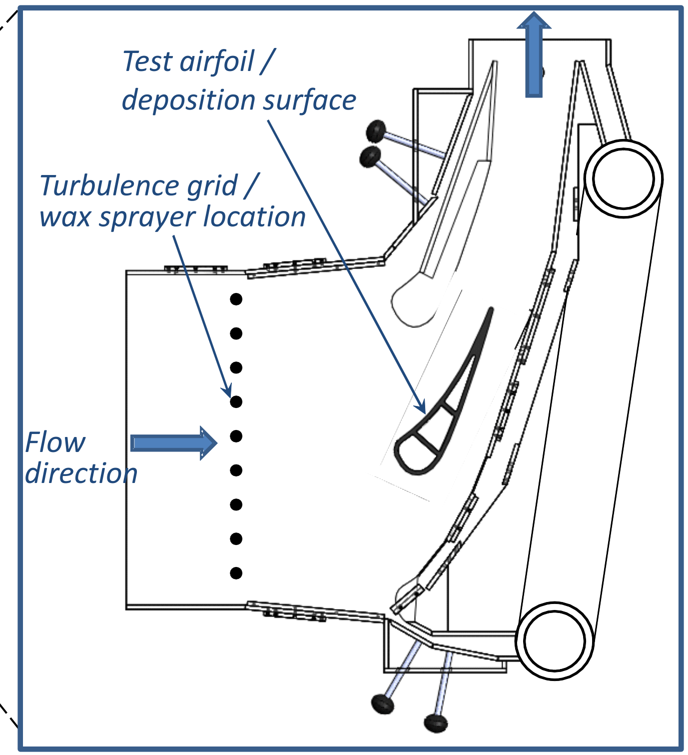
Matching these nondimensional parameters to engine conditions will result in engine-like results for ϕ.

TTCRL

Experiments were conducted to investigate effects of TBC and contaminant depositions on film cooling performance for a vane.

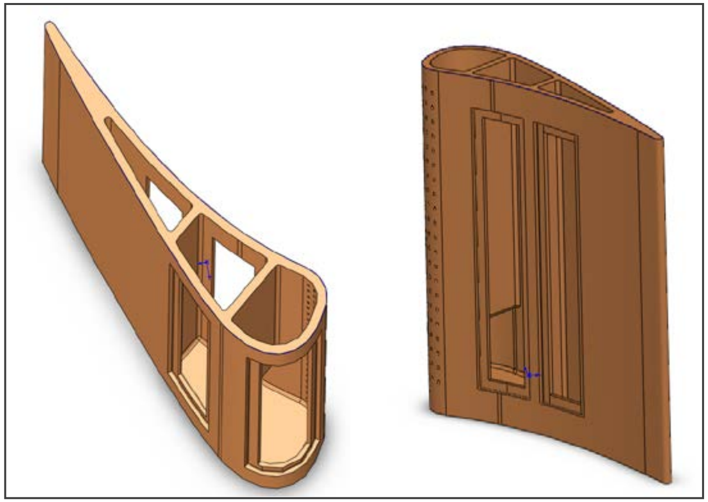


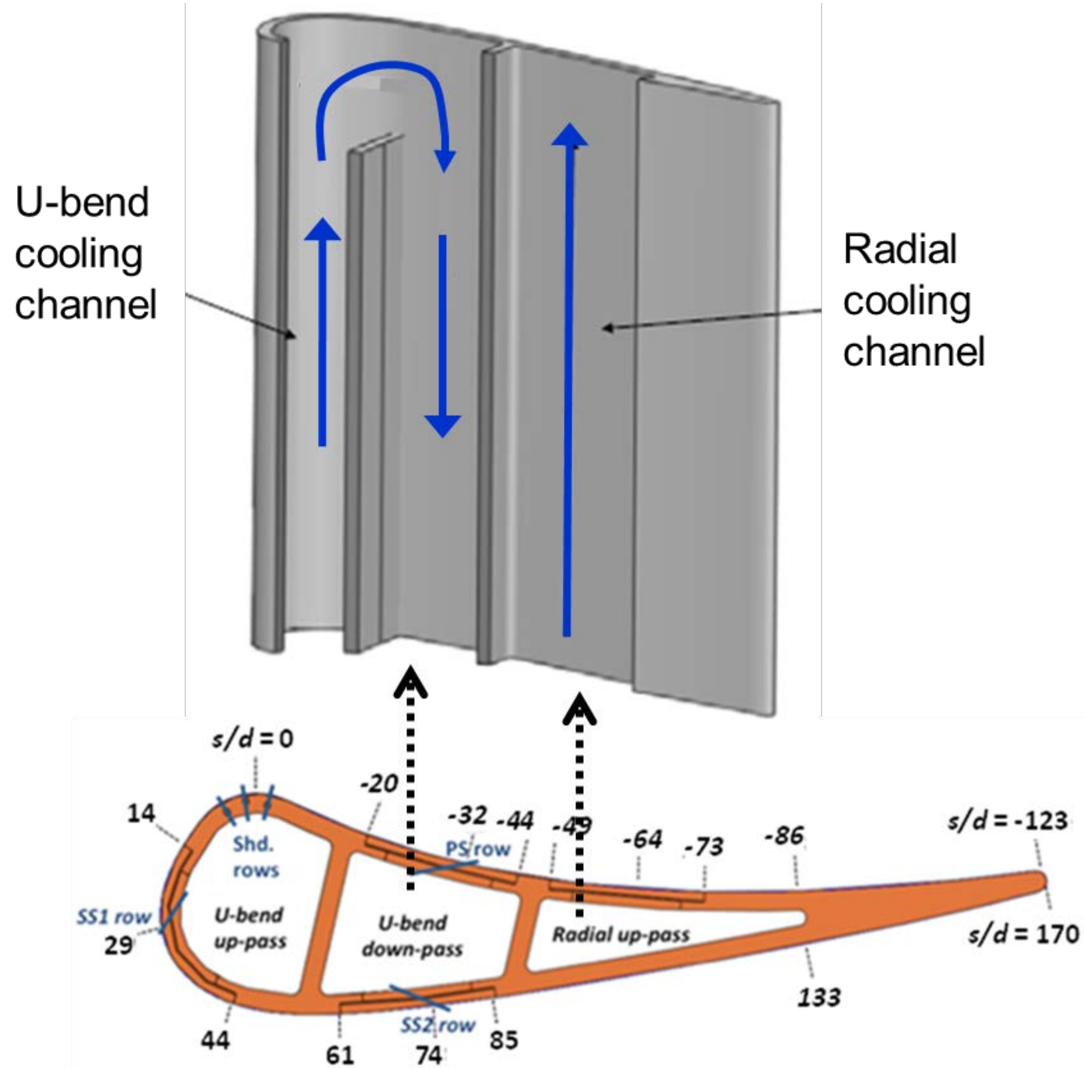
Test section:



- Performance was quantified in terms of overall effectiveness.
- Multiple hole geometries were investigated.

Schematics of the internal cooling configuration for the vane models:

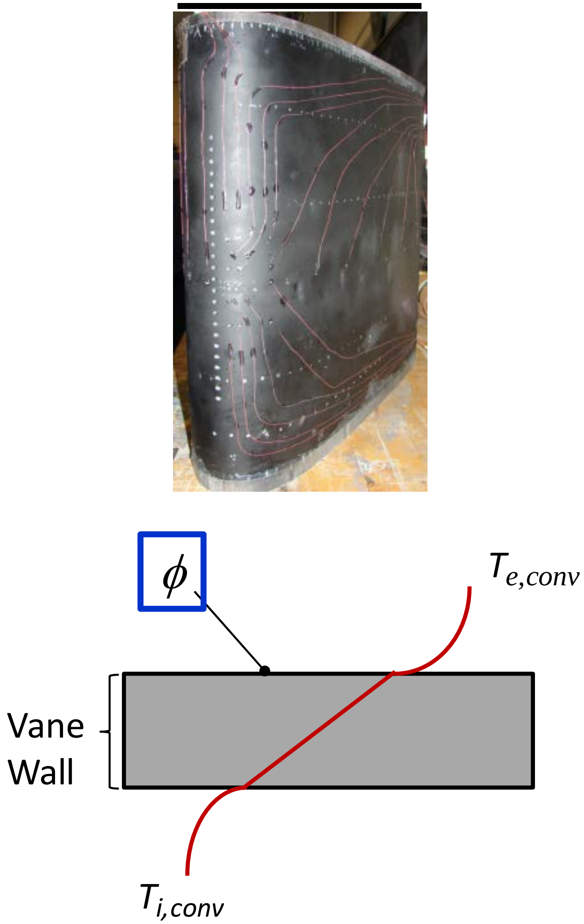




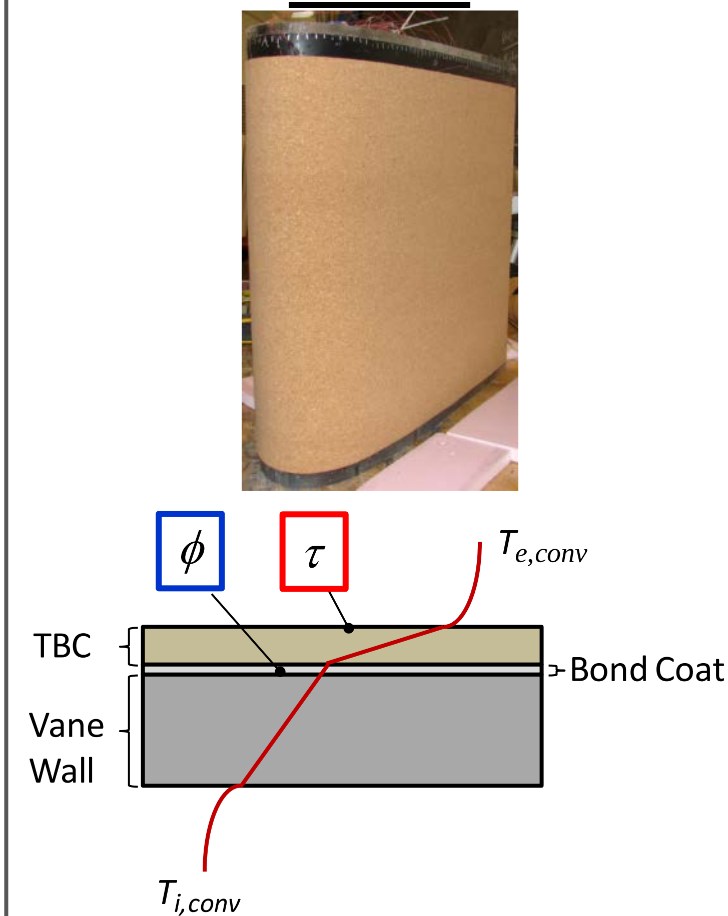
Model vane is designed to match thermal behavior of real vane.

Simulated thermal barrier coating:

Uncoated



Coated



$$\phi = \frac{T_\infty - T_{w,e}}{T_\infty - T_{c,vane\ inlet}}$$

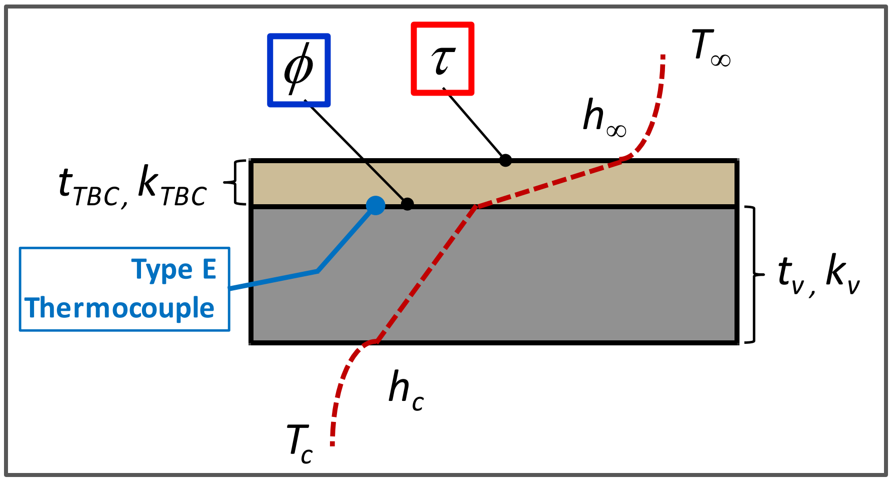
$$\tau = \frac{T_\infty - T_{TBC,e}}{T_\infty - T_{c,vane\ inlet}}$$

Simulated TBC was chosen to match thermal behavior of real TBC.

TTCRL

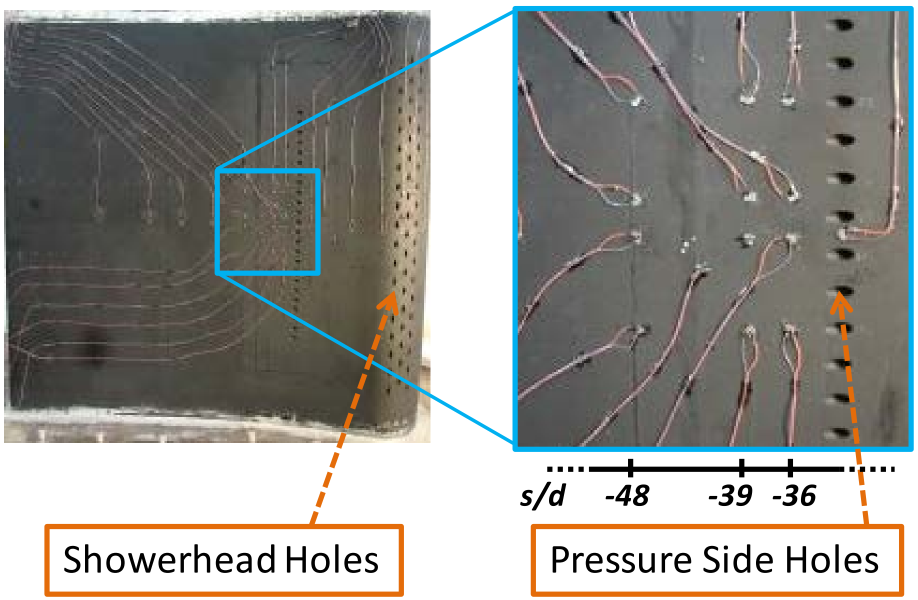
External temperatures measured using an IR camera, and interface temperatures measured with array of surface TC's:

- IR Thermography
- Type E Thermocouples



Surface Temperature: τ

Interface Temperature: ϕ



Scaling of TBC thickness

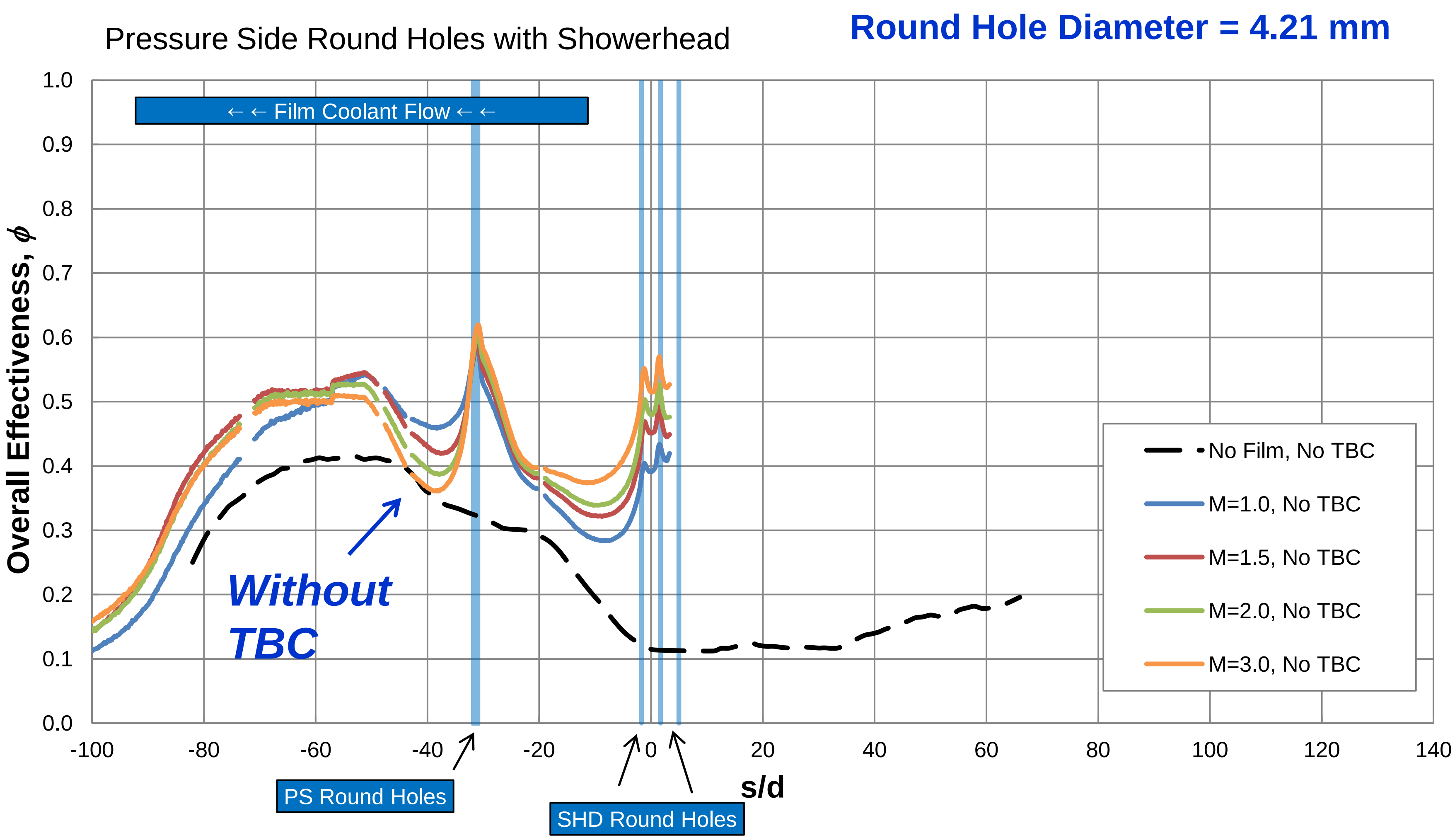
- To correctly simulate the thermal effects of TBC, it is important that the simulated TBC we use in our models is scaled appropriately

Parameter	Real Turbine	Model Turbine	Units
Vane Thickness	1.3-3	12.7	mm
TBC Thickness	0.14 – 0.72	1.92 & 5.1	mm
Vane Conductivity	20	1.02	W/m-K
TBC Conductivity	0.83 – 1.7	0.065	W/m-K
Heat Transfer Coeff.	1500 – 5000	25 – 90	W/m^2-K
TBC/Vane Thickness	0.14 – 0.72	0.15 & 0.40	-
TBC/Vane Conductivity	0.04 – 0.08	0.06	-
Vane Bi	0.1 – 0.6	0.3 – 1.1	-

A range of TBC thicknesses being tested that simulate relatively thin and thick TBC.

TTCRL

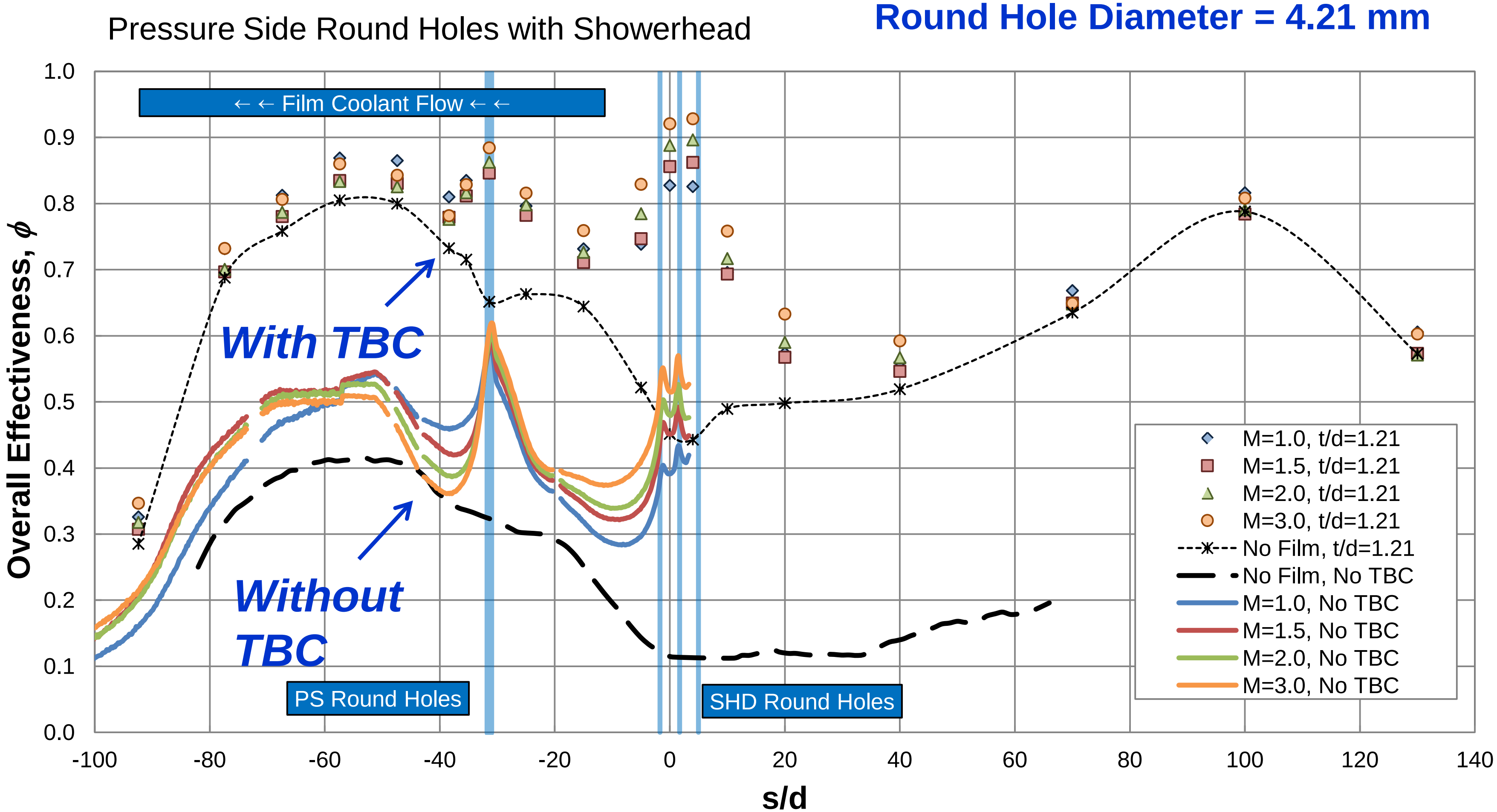
Effect of TBC on metal temperature, ϕ



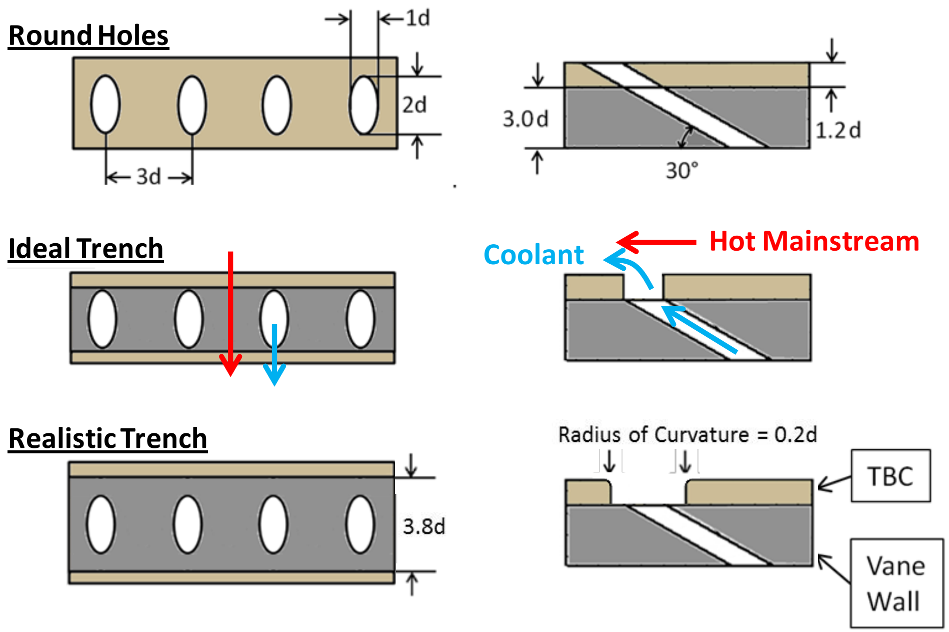
TTCRL

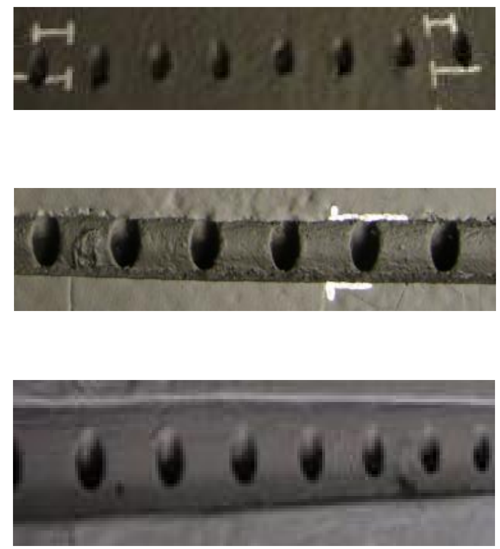
Effect of TBC on metal temperature, ϕ

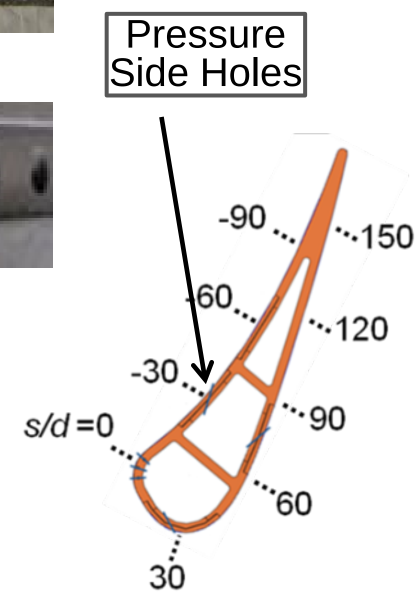
Round Hole Diameter = 4.21 mm



Film cooling configurations used with TBC on the pressure side of the vane model

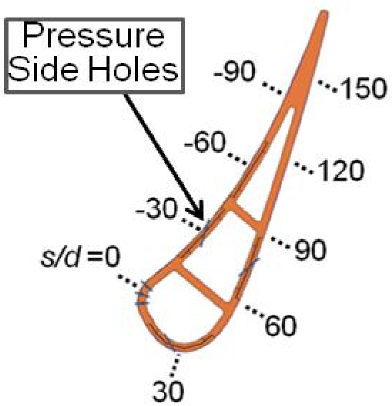




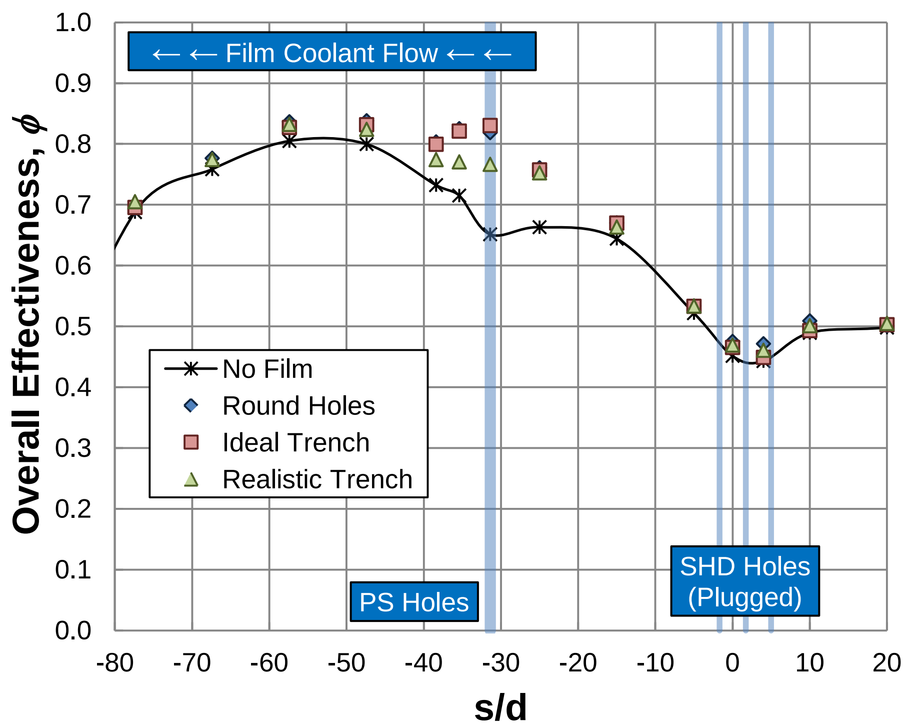


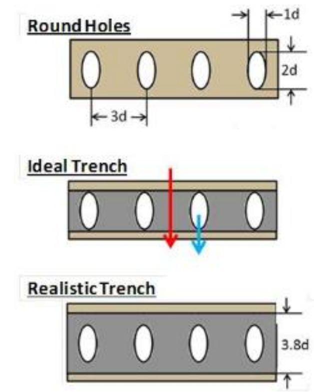
Round Hole Diameter = 4.21 mm

Effect of configuration with TBC on ϕ :



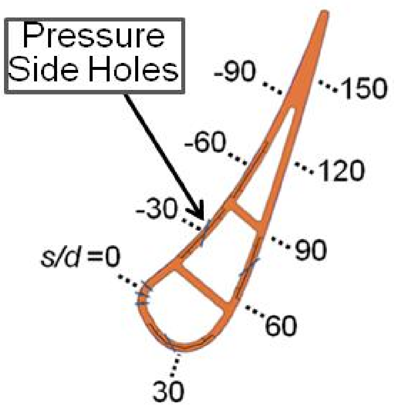
M = 0.5, t/d = 1.21

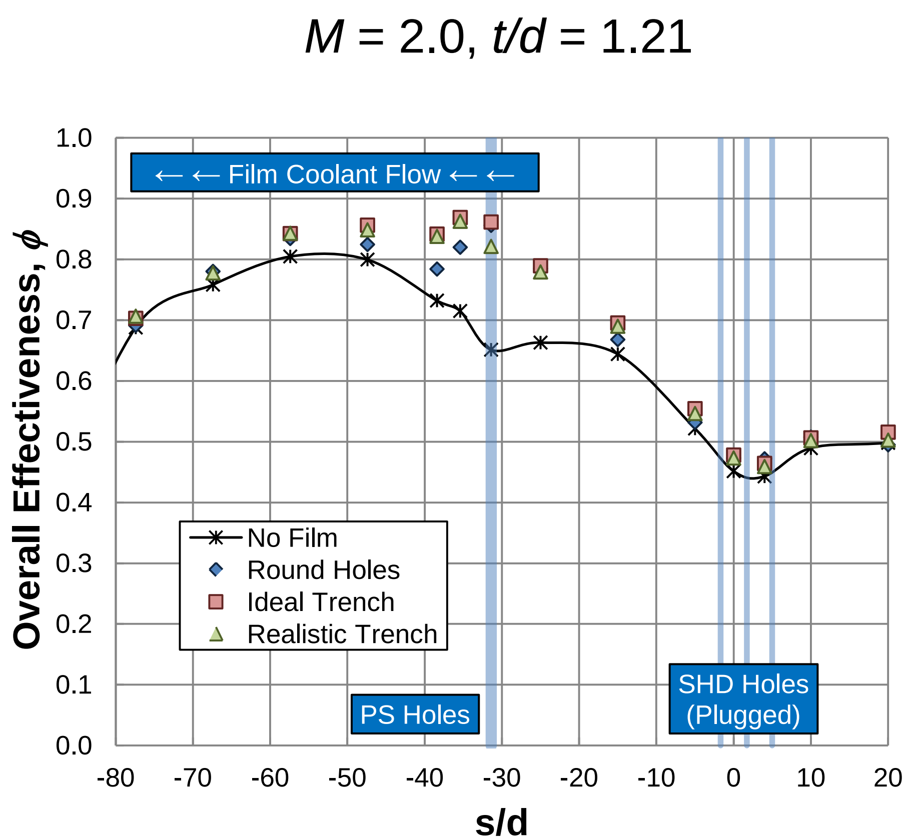


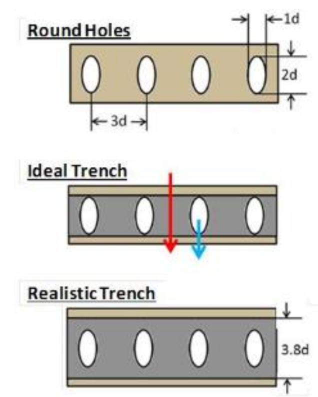


Minimal change in ϕ with changes in configuration.

Effect of configuration with TBC on ϕ:





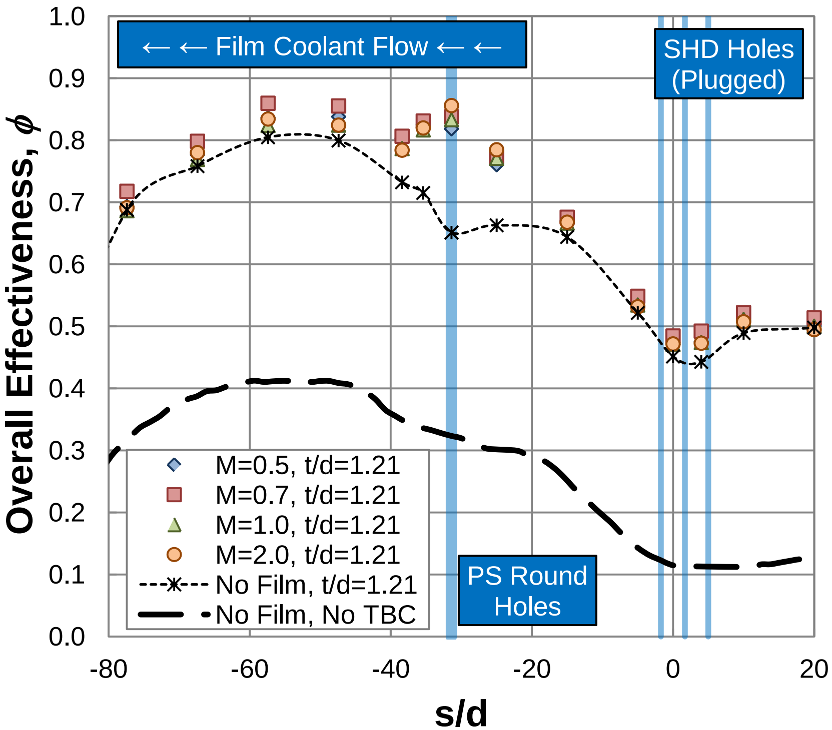


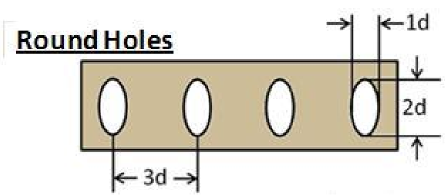
Minimal change in ϕ with changes in configuration.

TTCRL

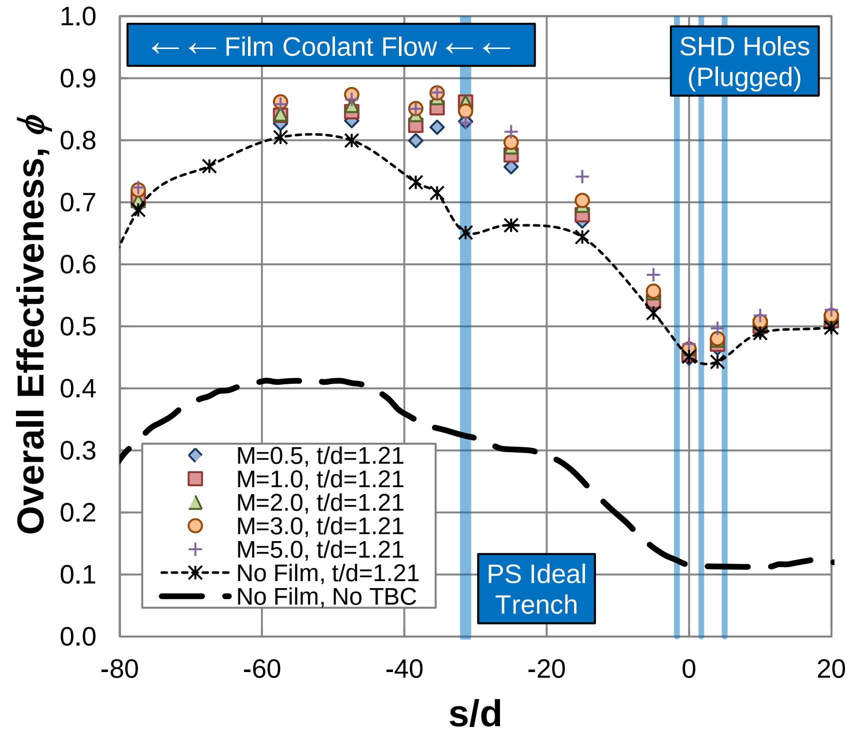
Effect of blowing ratio on metal temperature, ϕ with TBC

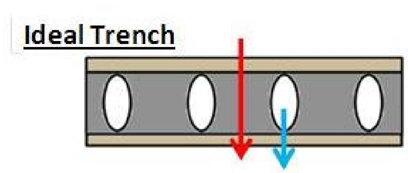
Round Holes, *t/d*=1.21

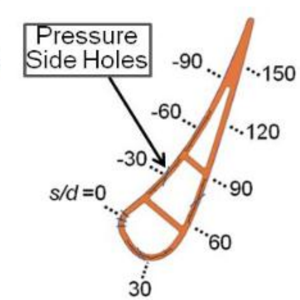




Ideal Trench, *t/d*=1.21





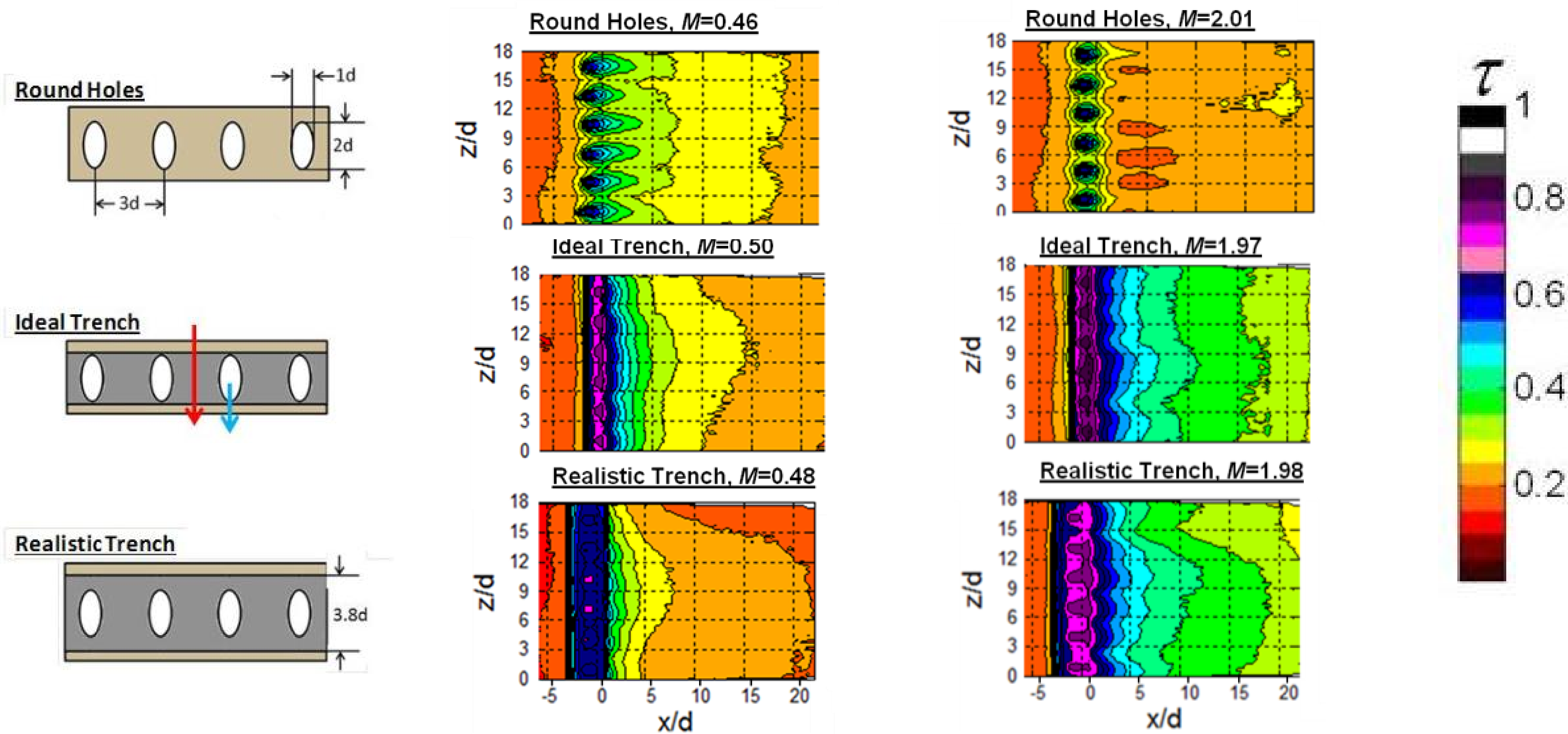


Marginal increase in ϕ with increasing *M*

TTCRL

Effect of blowing ratio on normalized exterior TBC temperature, τ

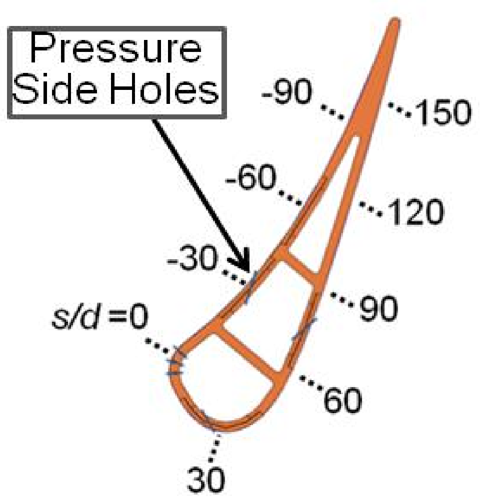
M = 0.5 **M = 2.0**

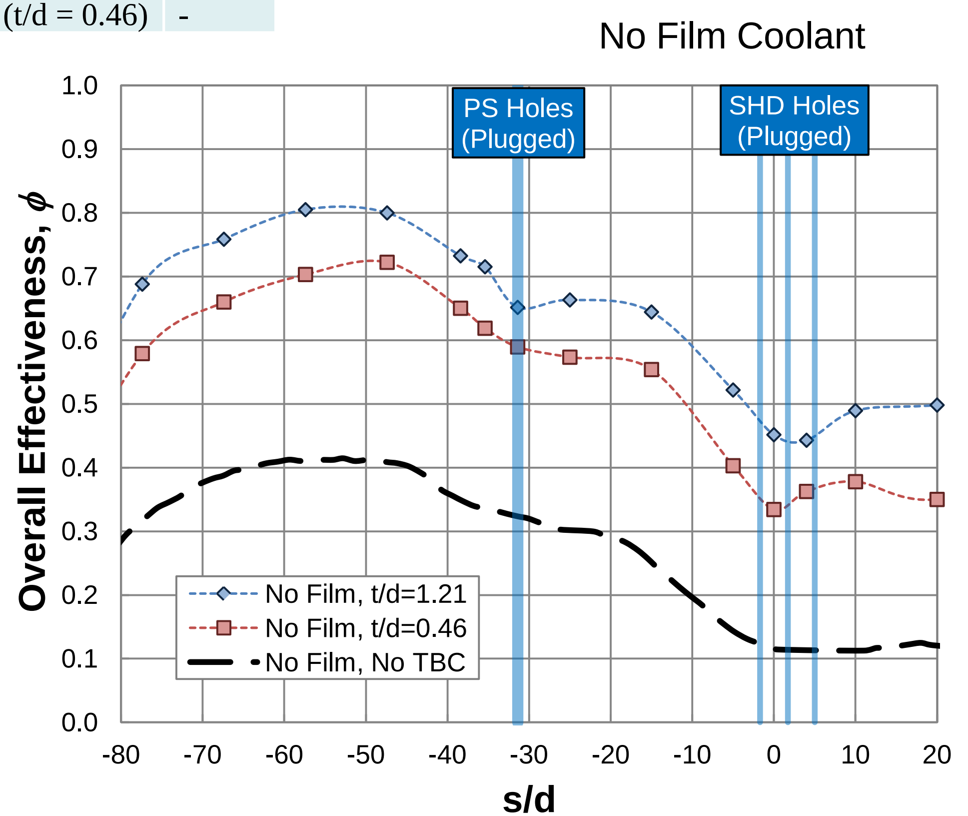


These results confirmed that the trench was providing good external film effectiveness.

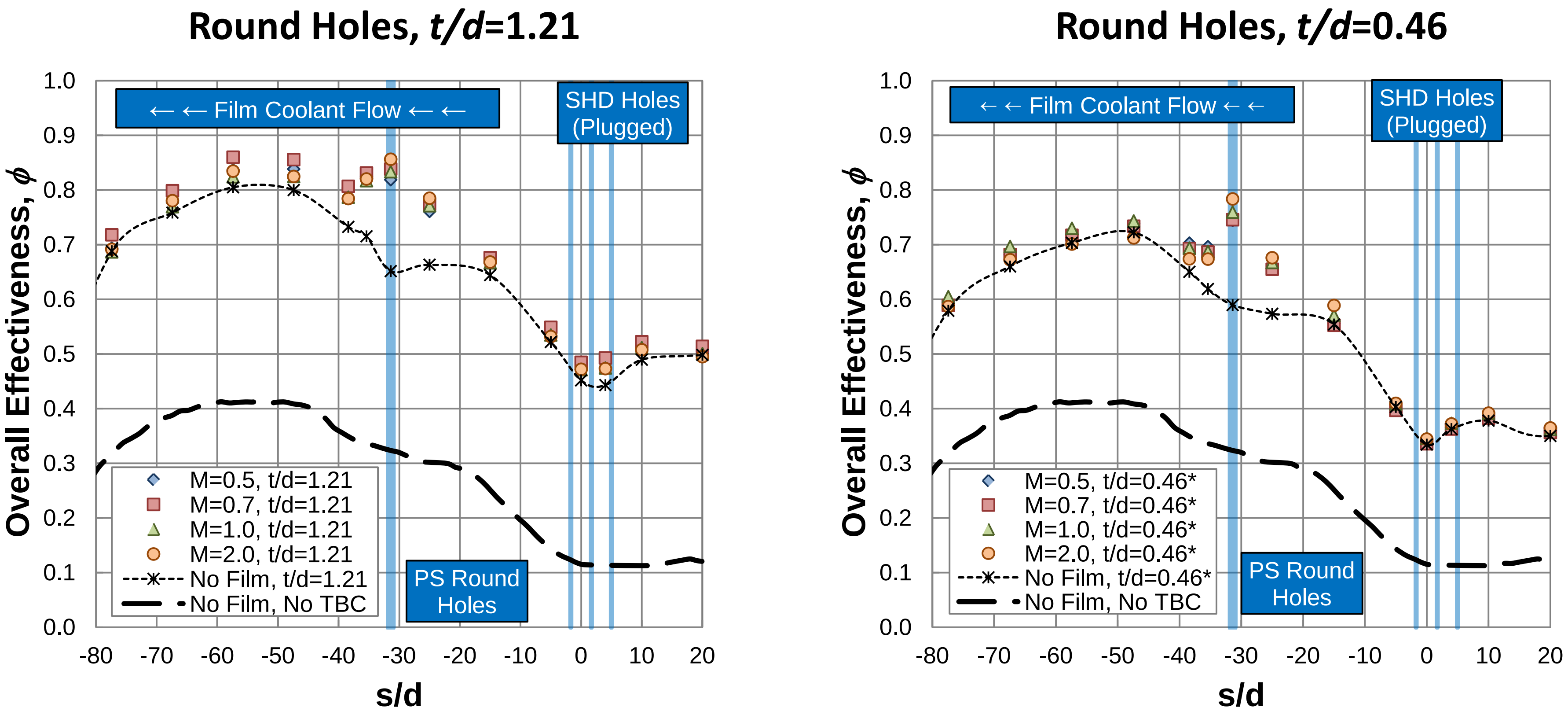
Effect of TBC thickness on ϕ with no film cooling

Parameter	Real Turbine	Previous Model	Current Model	Units
Vane Thickness	1.3-3	12.7	12.7	mm
TBC Thickness	0.14 – 0.72	5.1	1.92	mm
TBC/Vane Thickness	0.14 – 0.72	0.40 (t/d = 1.21)	0.15 (t/d = 0.46)	-





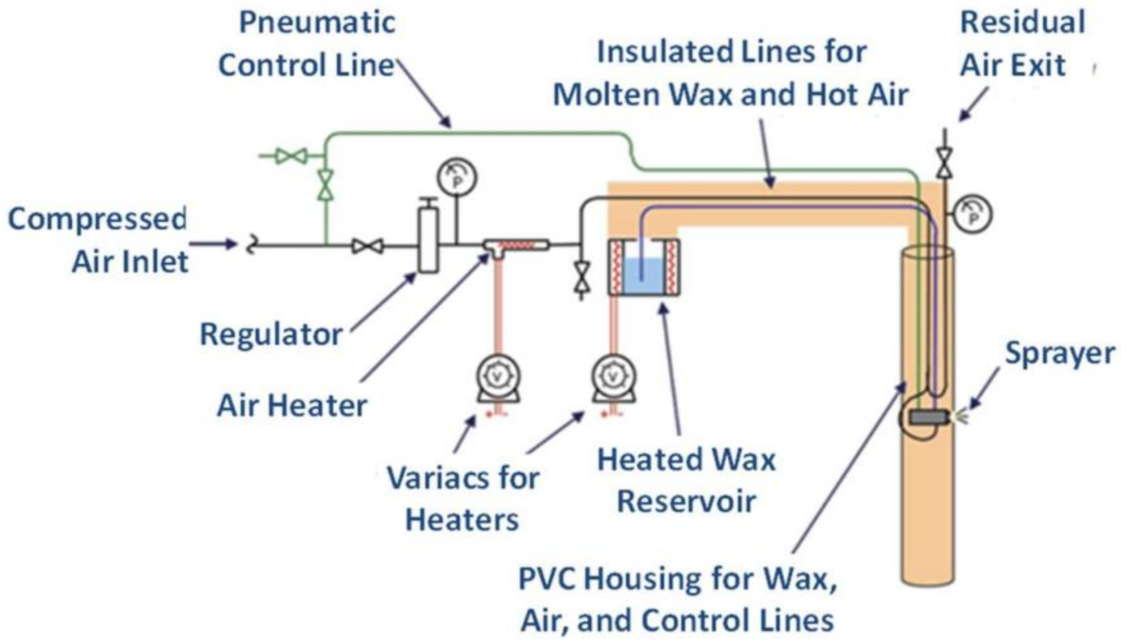
Effect of TBC thickness on metal temperature, ϕ, with film cooling using round holes



As expected, ϕ decreases with decreasing TBC thickness, but the increase in ϕ is similar.

TTCRL

Deposition of contaminants was simulated using the molten wax techniques we have described previously

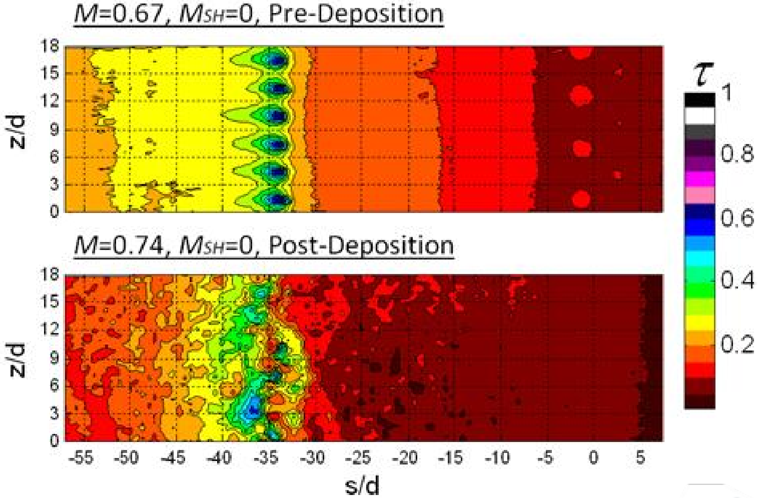


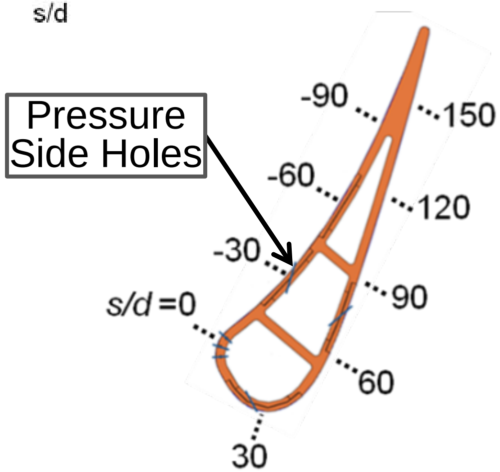
System generates properly scaled particles.

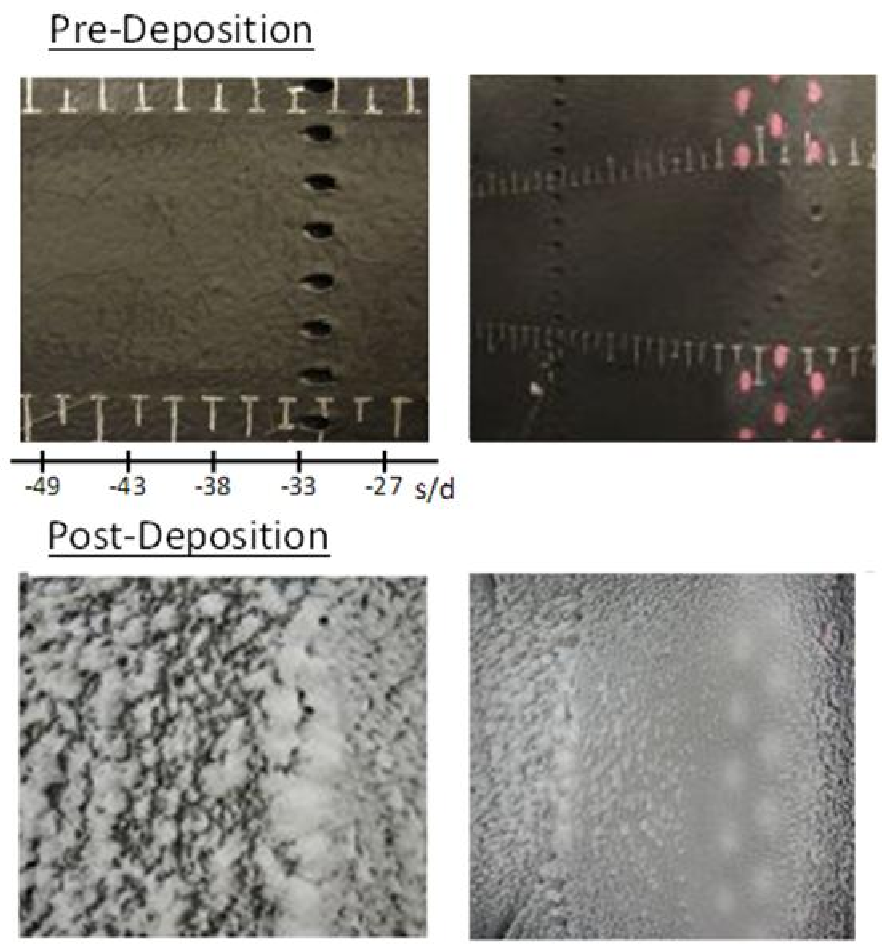
TTCRL

Deposition for round holes, *M* = 0.7:

← ← Film Coolant Flow ← ←

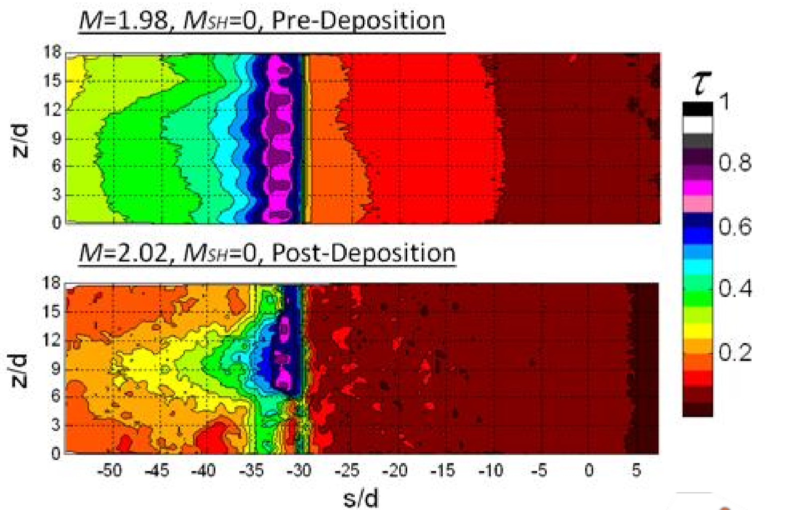


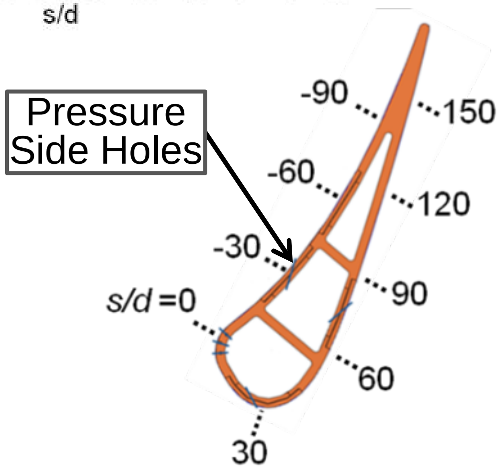


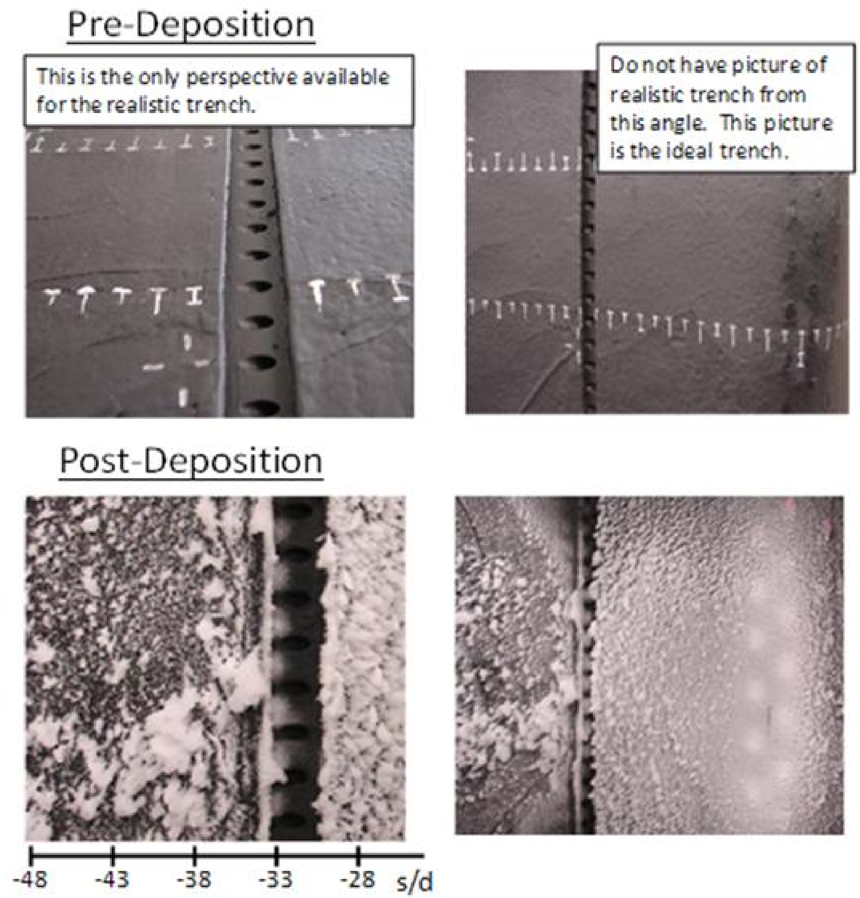


Deposition almost buried round holes at *M* = 0.7.

Deposition for realistic trench, M = 2.0:



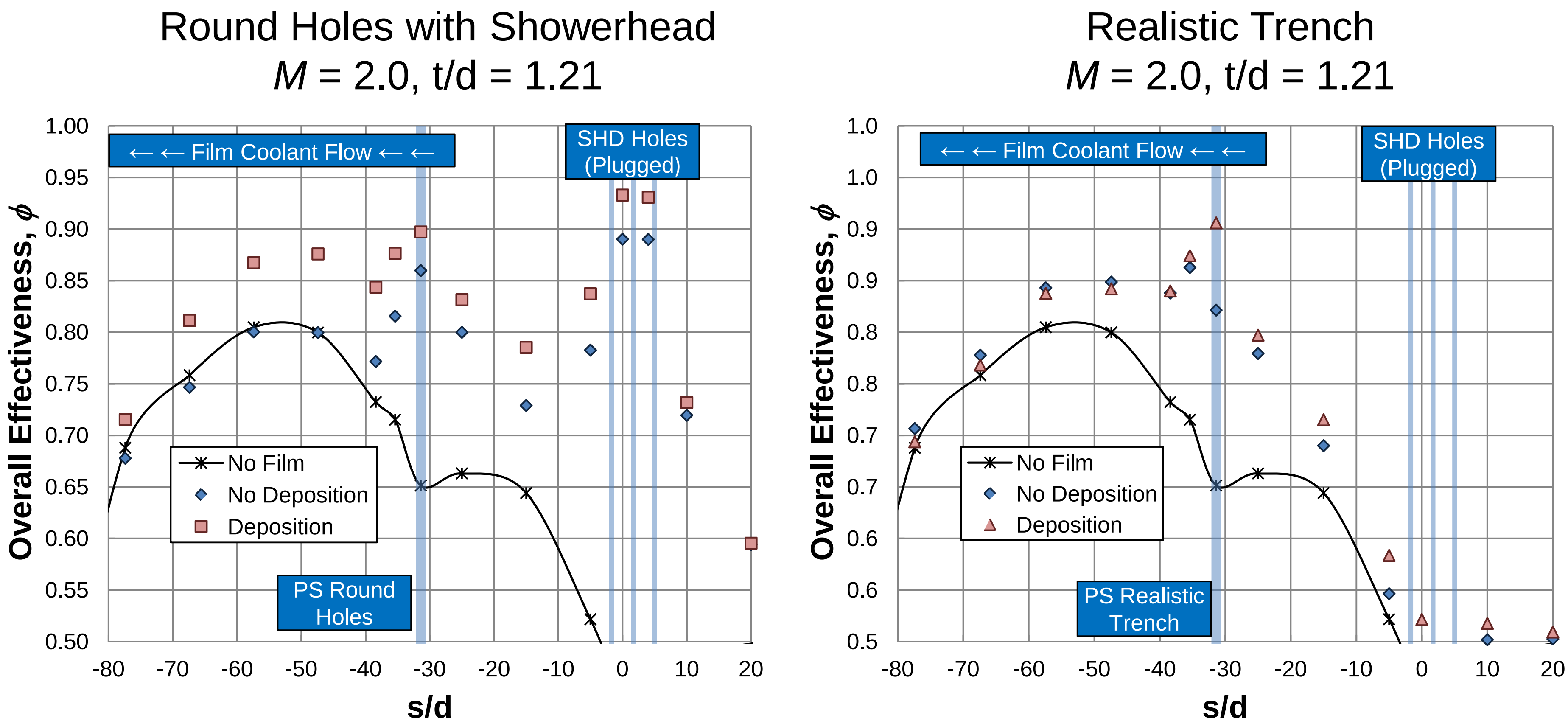




Large growth on trench lip (not shown); Minimal growth elsewhere.

TTCRL

Effect of deposition on ϕ for round holes and the realistic trench

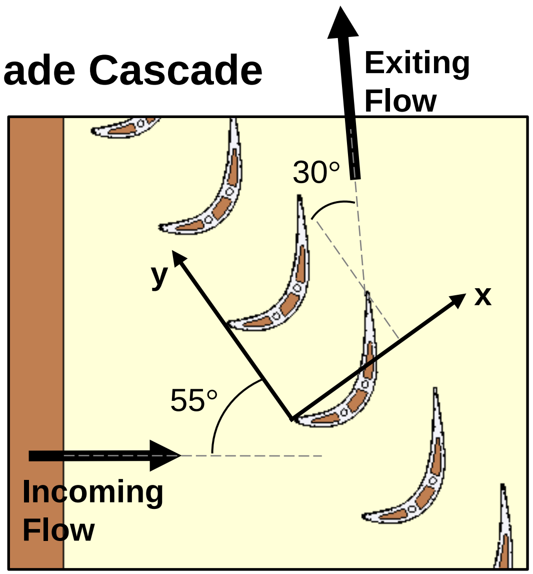


Depositions increase ϕ for round holes due to the additional insulation provided by the layers of deposits.

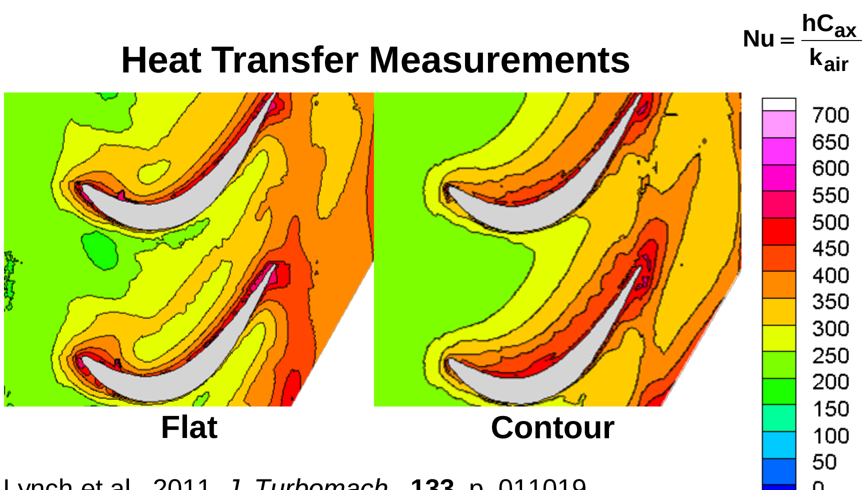
Conjugate endwall heat transfer studies using the Pack-B blade cascade build on previous work

Pack-B Blade Cascade

No. of airfoils	7
Scale	8.6x
Pitch/chord (P/C_{ax})	0.89
Aspect ratio (S/C_{ax})	2.5

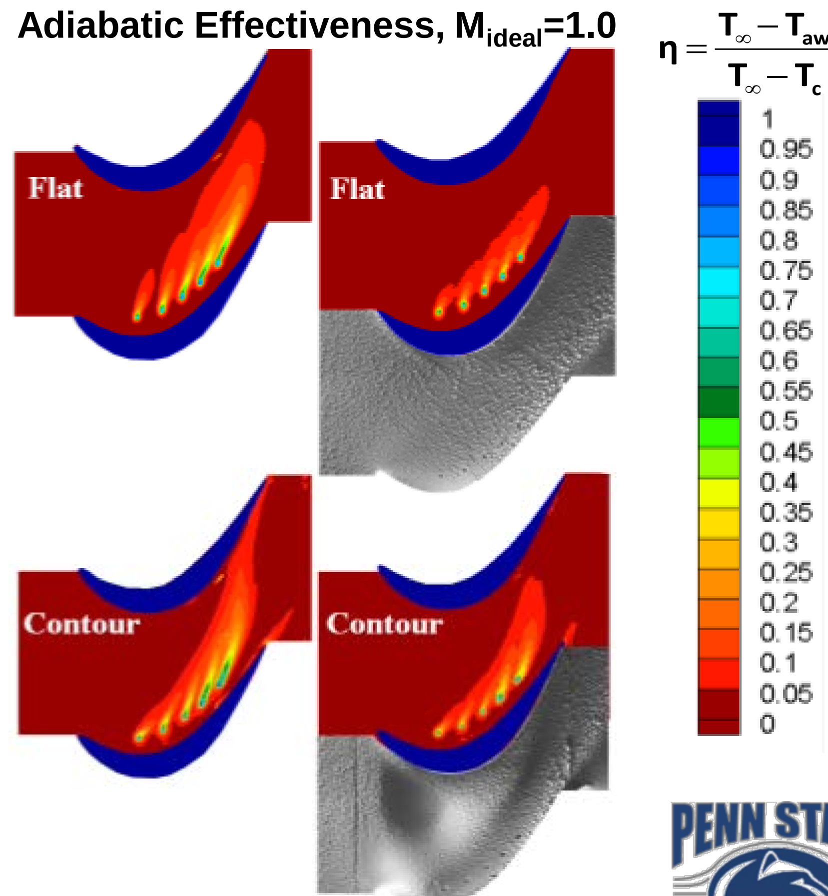


Heat Transfer Measurements



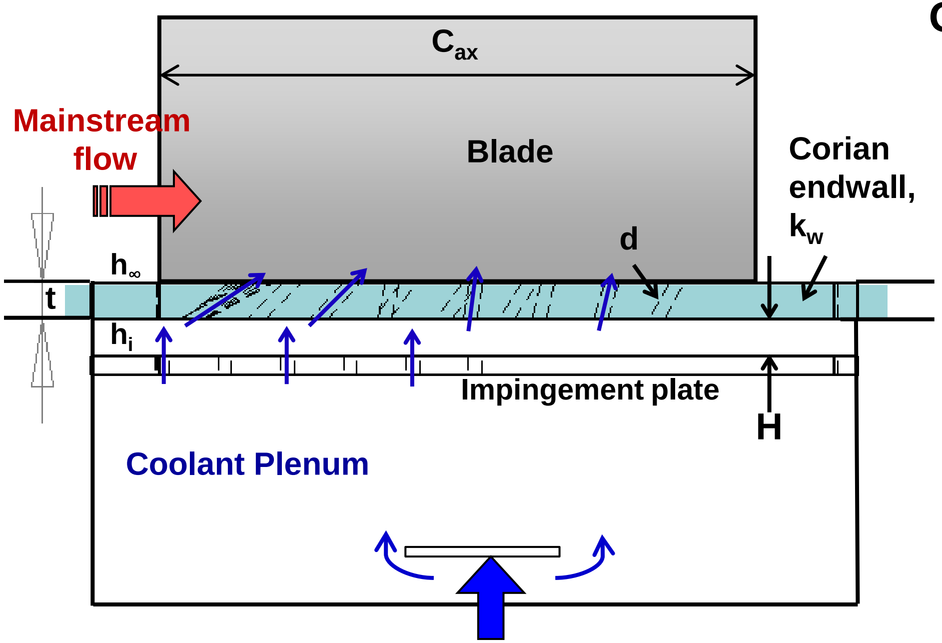
Lynch et al., 2011, *J. Turbomach.*, **133**, p. 011019

Adiabatic Effectiveness, M_{ideal}=1.0



Lawson et al. 2012, ASME Turbo Expo, GT2012-68174

Matching Bi and other non-dimensional parameters to engine conditions allows direct measurement of ϕ



Overall effectiveness (metal temperature):

$$\phi = \frac{T_\infty - T_w}{T_\infty - T_c} = \frac{1 - \left(\frac{T_\infty - T_{film}}{T_\infty - T_c}\right)}{1 + Bi + h_\infty / h_i} + \left(\frac{T_\infty - T_{film}}{T_\infty - T_c}\right)$$

where, $Bi = h_\infty t / k_w$ and

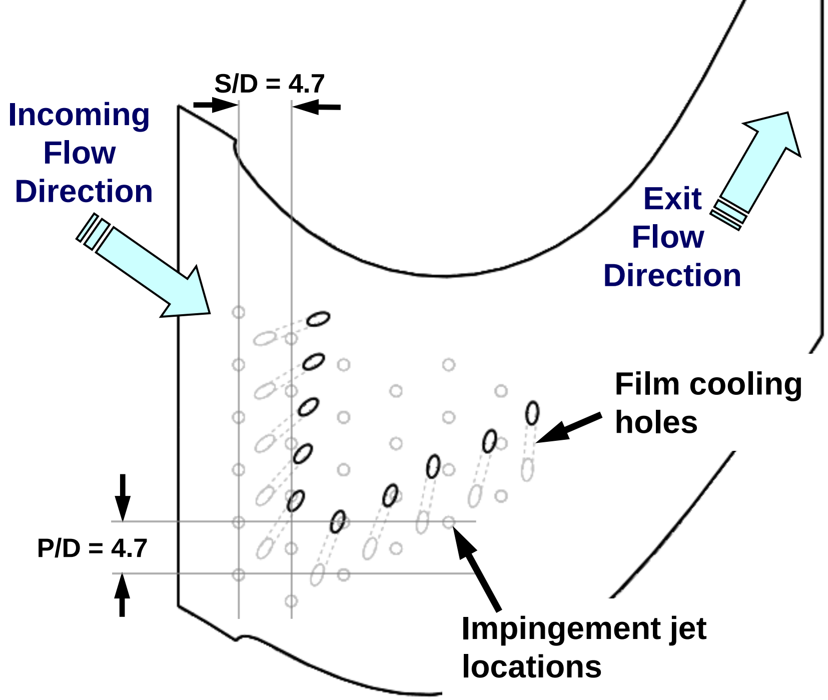
T_{film} = the driving temperature for external convection (T_{aw} is normally used)

Matched Parameters	Engine	Model
$Re_{\infty,in}\ (C_{ax})$	1.25×10^5	1.25×10^5
$M_{filmcooling} = \frac{\rho_c U_c}{\rho_\infty U_\infty}$	1.0 - 2.0	0.6, 1.0, 2.0
h_∞ / h_i	1.0	0.4 - 2.5
$Bi = h_\infty t / k_w$	0.27	0.30 - 0.77
H/d	3	2.9

Using DuPont Corian® endwall

Film and impingement cooling were applied separately and together at three blowing ratios

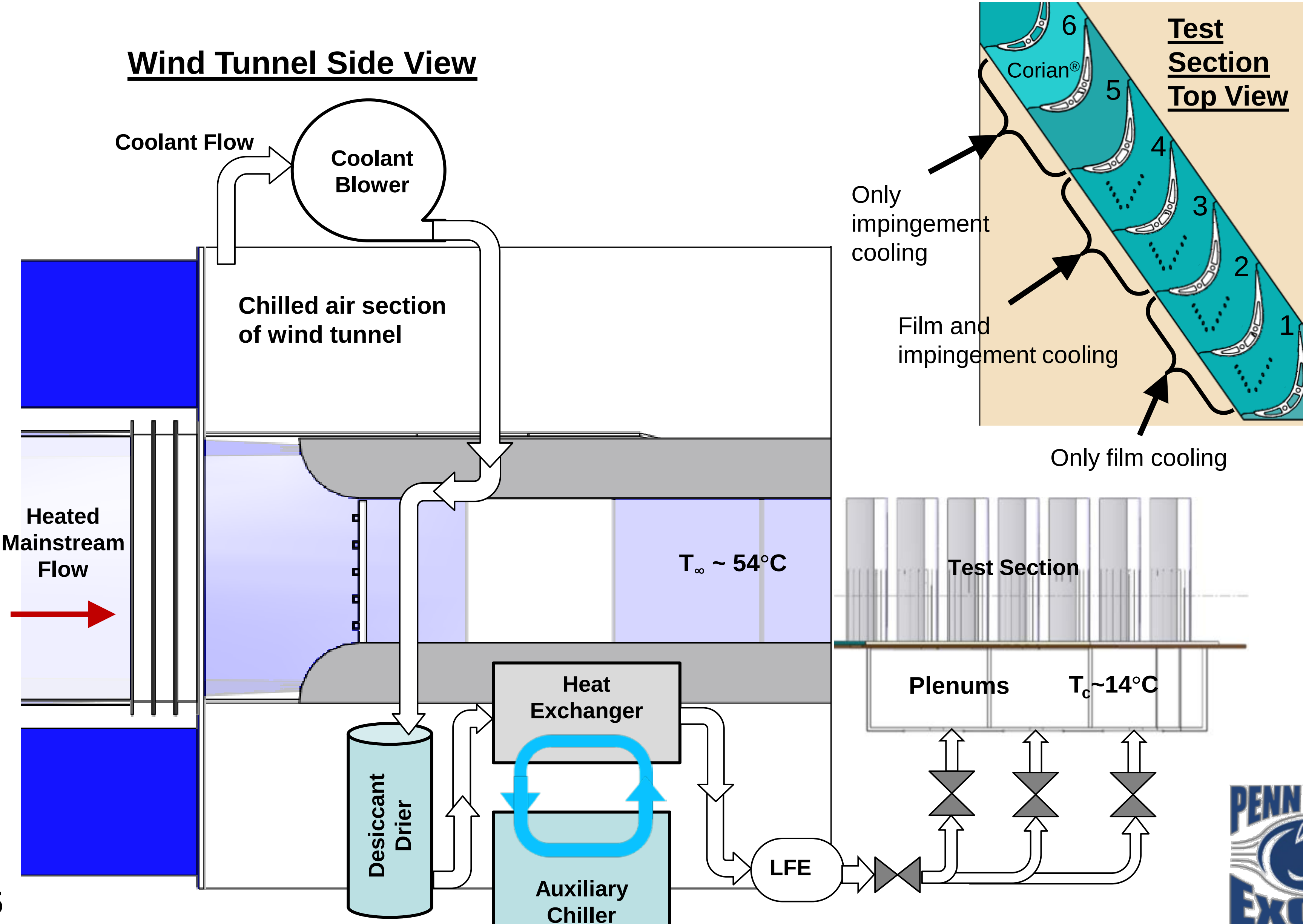
Top view of endwall within the passage



d = diameter of film and impingement holes

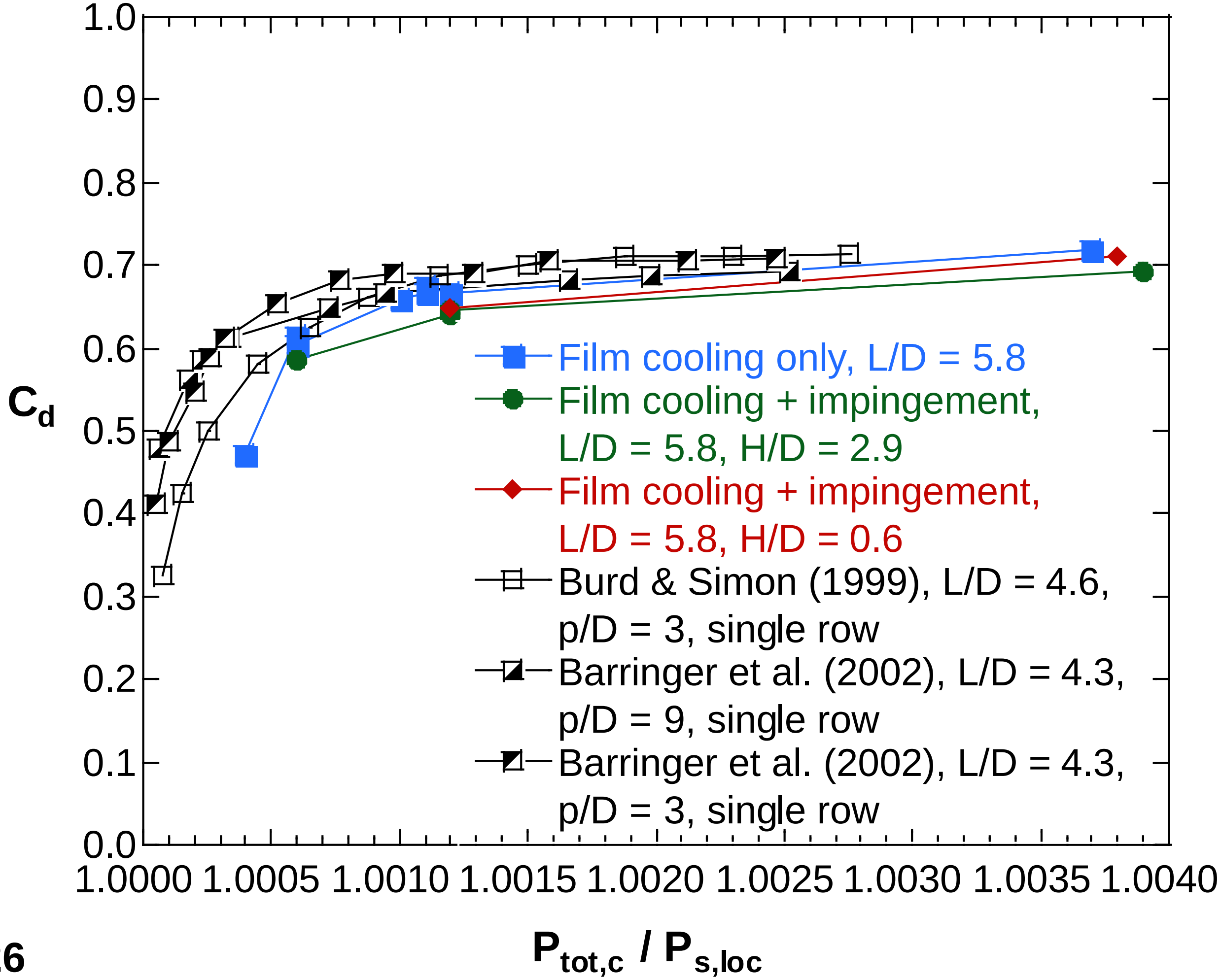
Cooling Arrangement	H/d (imping.)	M_{avg}	I_{avg}	h_∞/h_i
Film only	N/A	2.0	3.50	N/A
		1.0	0.94	
		0.6	0.32	
Impingement only	2.9	2.0	3.50	0.4-1.0
		1.0	0.94	0.6-1.4
		0.6	0.32	1.0-2.2
Film + Impingement	2.9	2.0	3.50	0.5-1.0
		1.0	0.94	0.7-1.6
		0.6	0.32	1.1-2.5
Impingement only	0.6	2.0	3.50	0.4-0.8
		1.0	0.94	0.5-1.2
Film + Impingement	0.6	2.0	3.50	0.5-1.1
		1.0	0.94	0.7-1.6

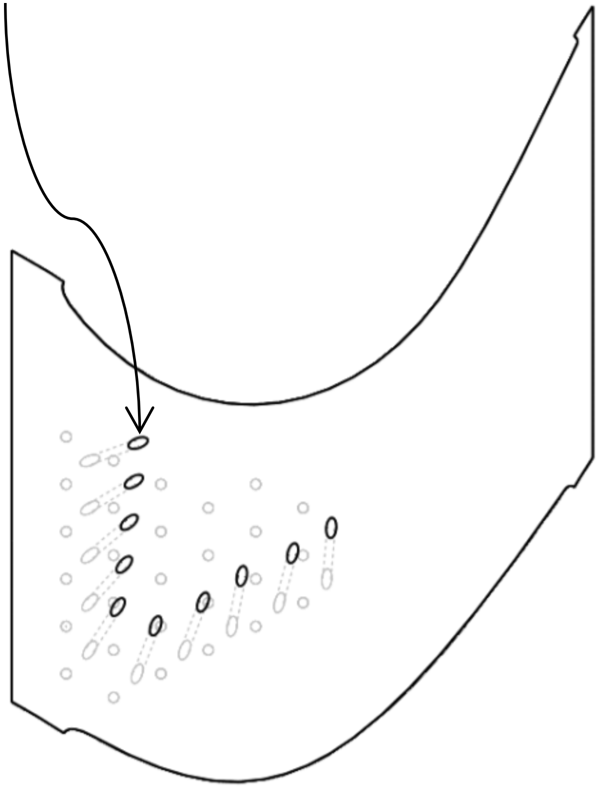
An additional chiller, heat exchanger and drier were installed to increase driving $\Delta T = T_\infty - T_c$



Discharge coefficients agreed well with literature, and C_d was lower with the impingement plate

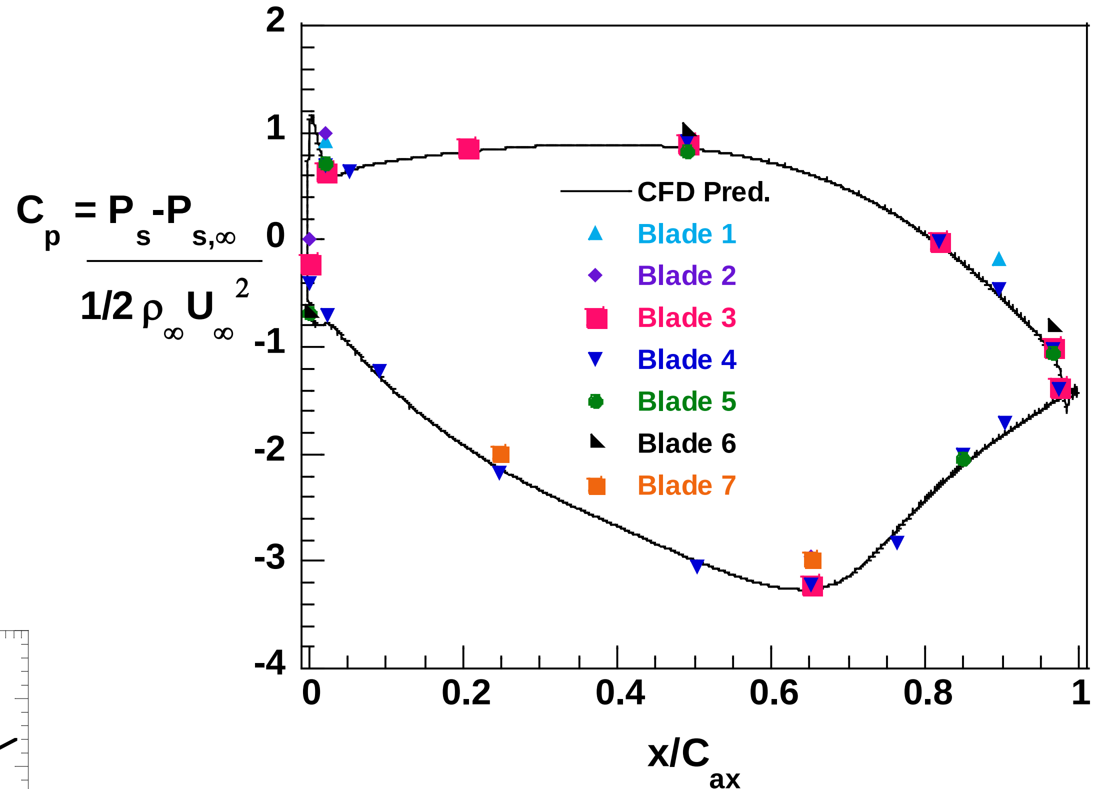
$$C_d = \frac{u_{actual}}{u_{ideal}} = \frac{\dot{m}_{actual}}{\rho_c u_{ideal} A_{hole}} = \frac{\dot{m}_{actual}}{\sqrt{2\rho_c \left(P_{tot,c} - P_{s,hole}\right)} A_{hole}}$$

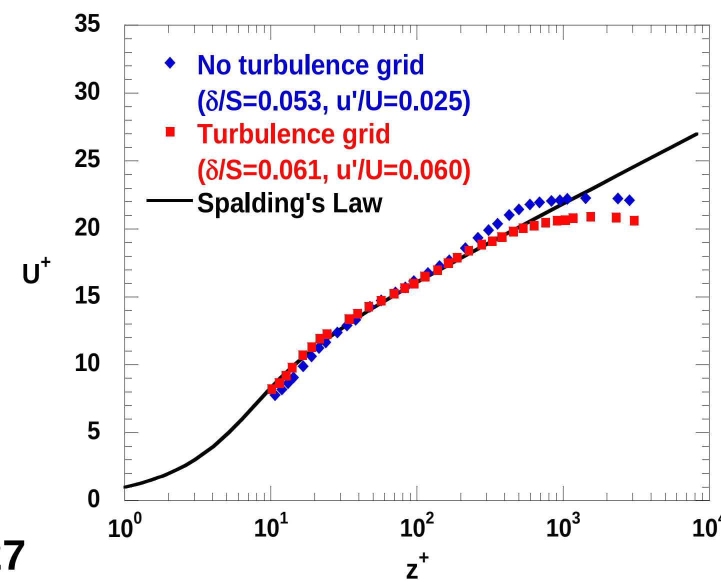




PENN STATE ExCCL

The blade midspan pressure distribution was benchmarked to a previous CFD prediction

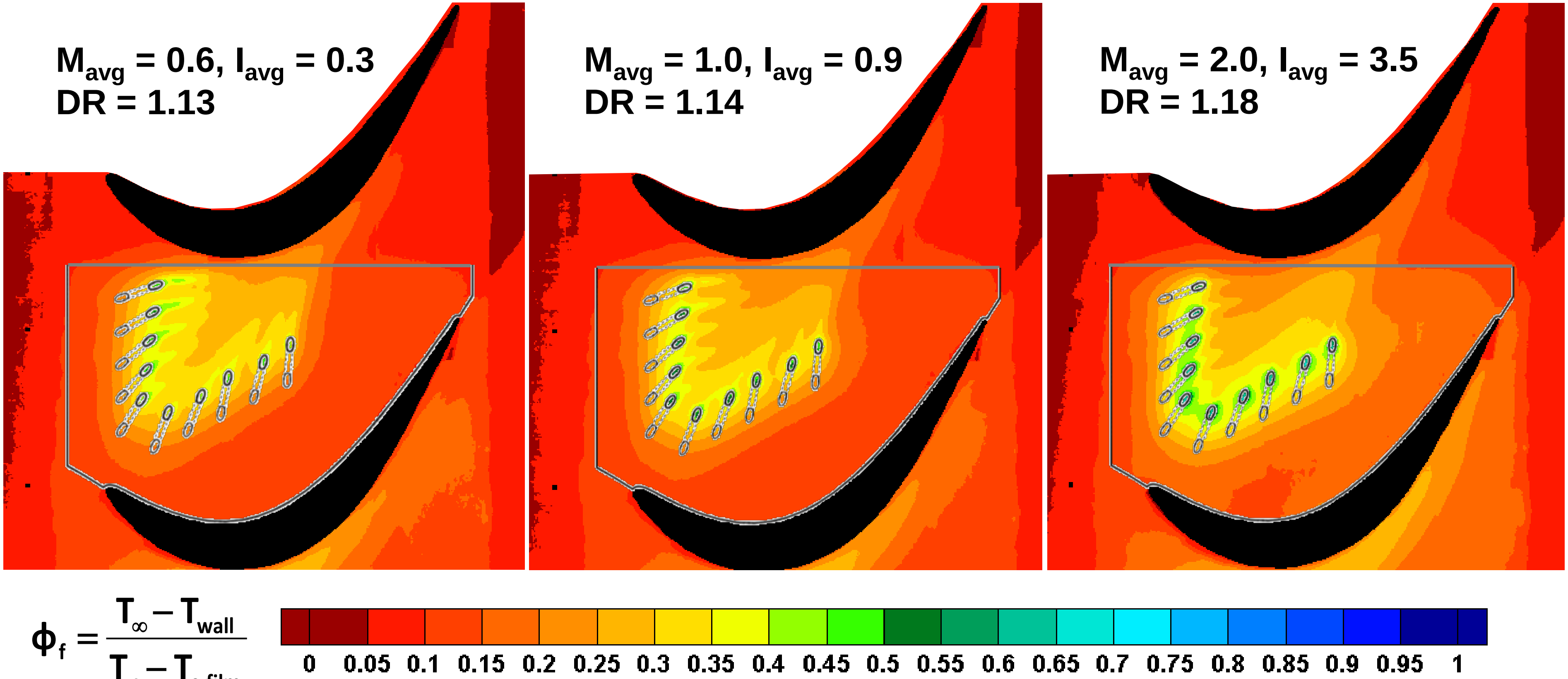




Previously measured inlet boundary layer
Lynch et al., 2011, *J. Turbomach*., **133**, p. 011019

With film cooling, ϕ was similar with blowing ratio, except for an increase directly around the hole exits

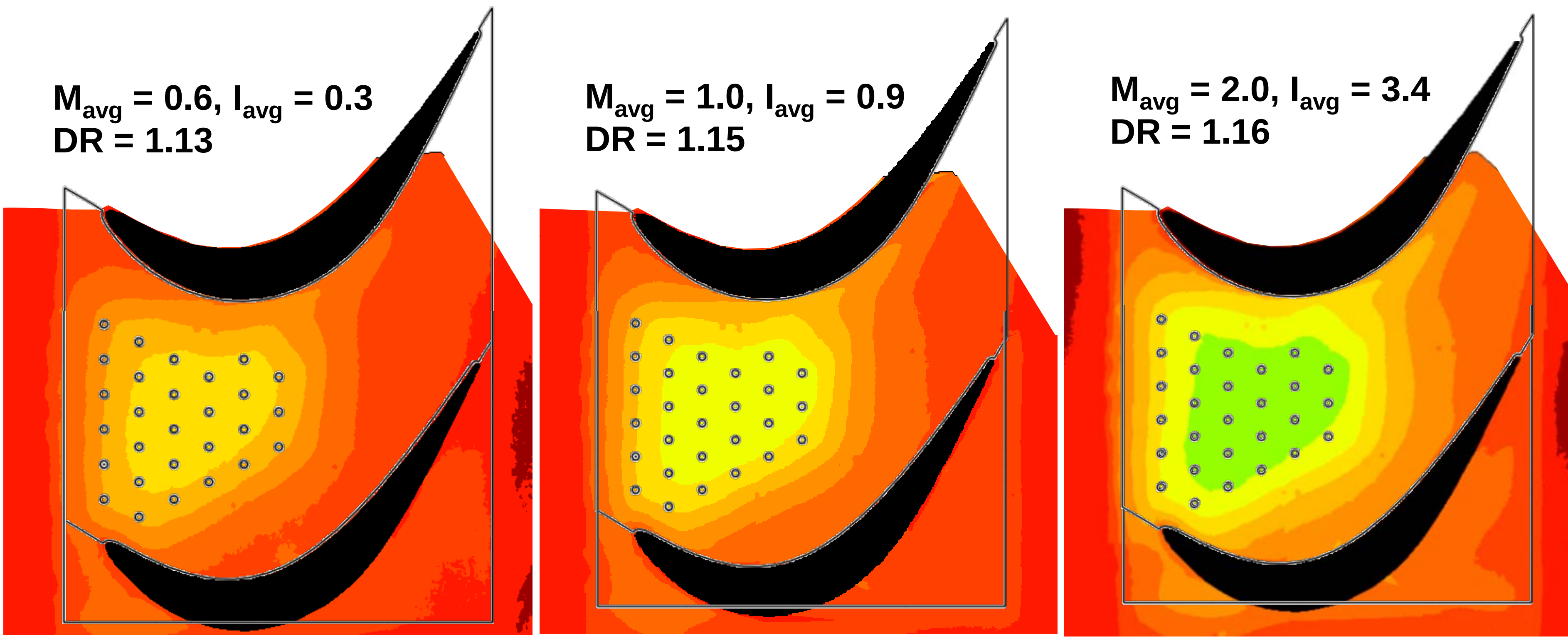
Film Cooling Only



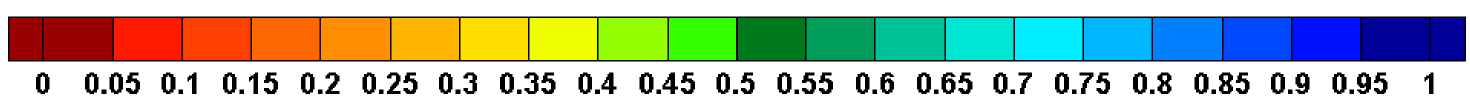
$$\phi_f = \frac{T_\infty - T_{wall}}{T_\infty - T_{c,film}}$$

For impingement cooling, ϕ was more uniformly distributed, and increased with blowing ratio

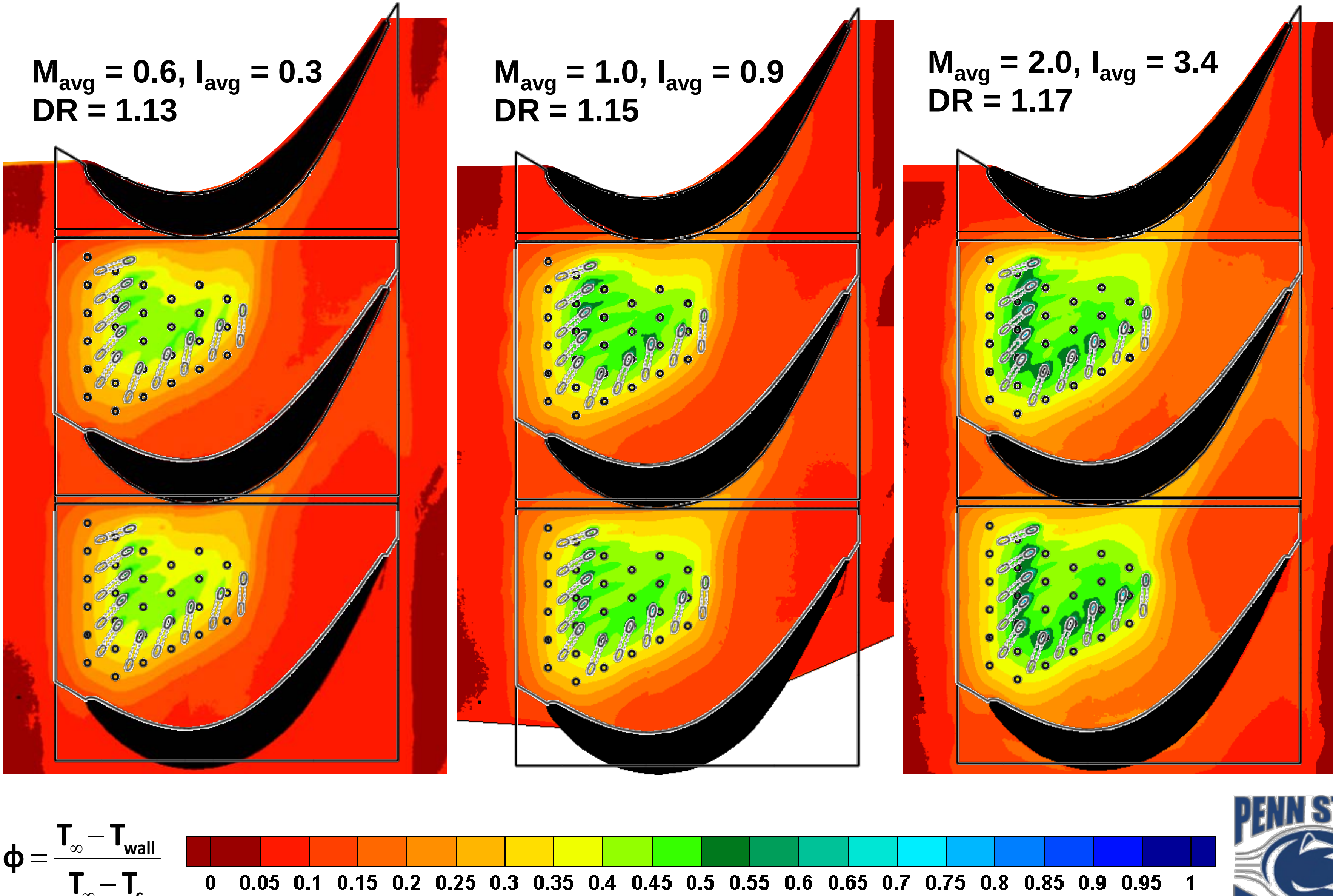
Impingement Cooling Only



$$\phi_o = \frac{T_\infty - T_{wall}}{T_\infty - T_c}$$



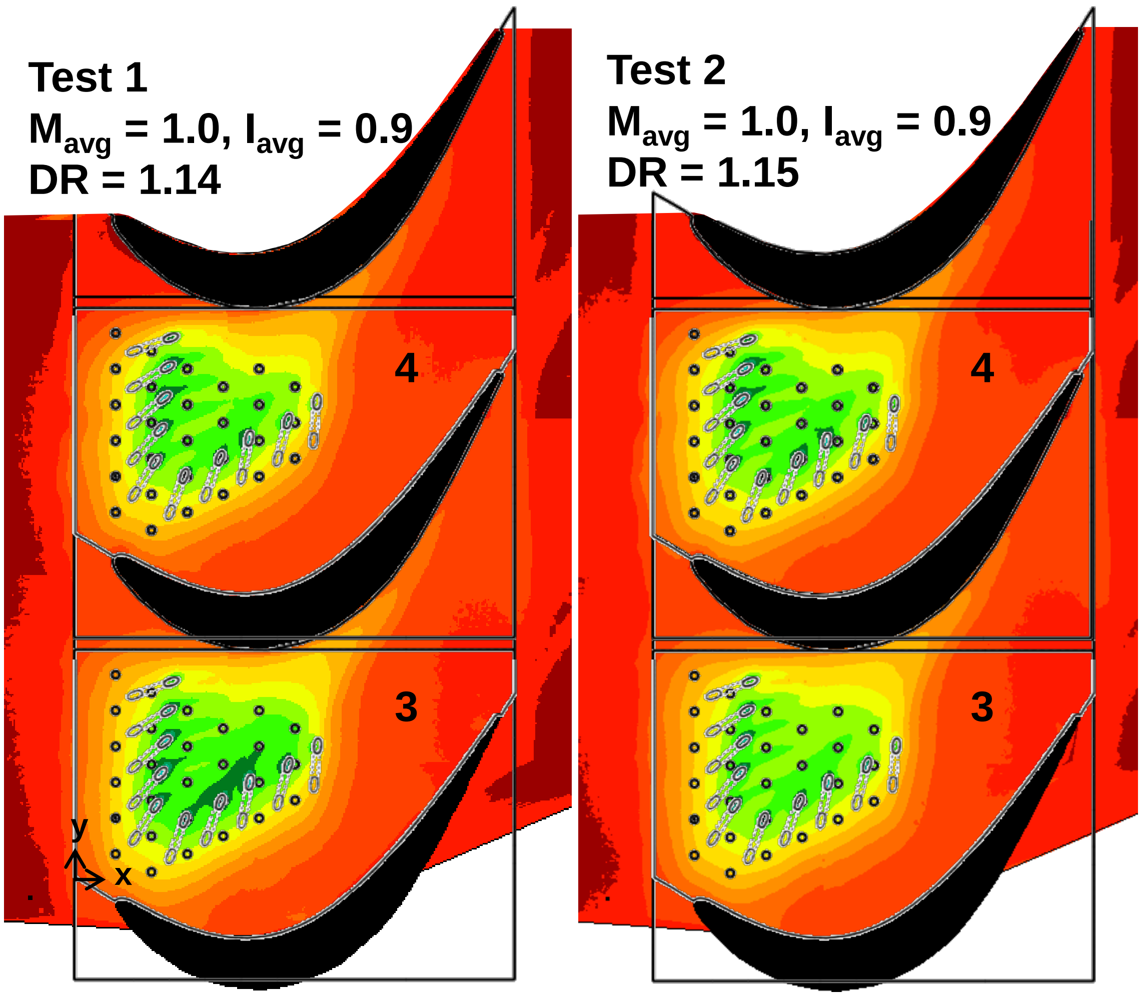
Film and impingement cooling had high ϕ around the film cooling holes, and increased with blowing ratio

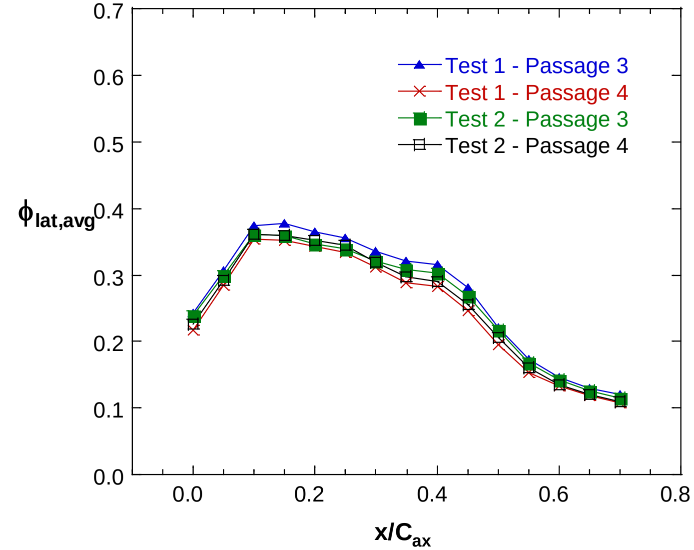


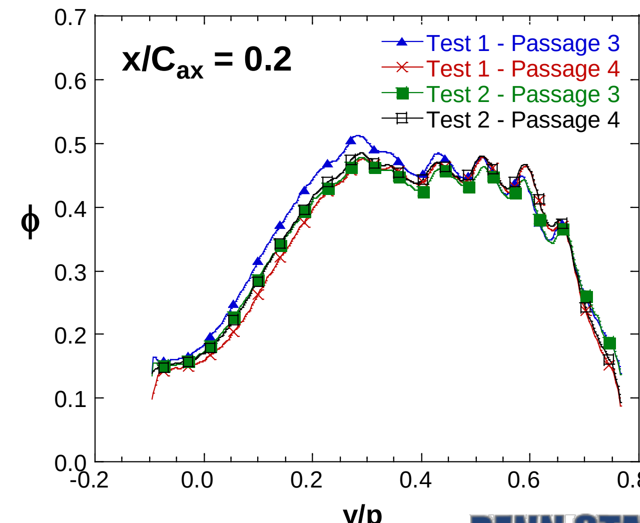
$$\phi = \frac{T_\infty - T_{wall}}{T_\infty - T_c}$$

PENN STATE ExCCL

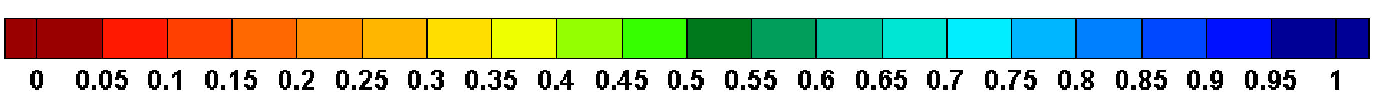
A repeated test showed good reproducibility of ϕ between passages and separate experiments



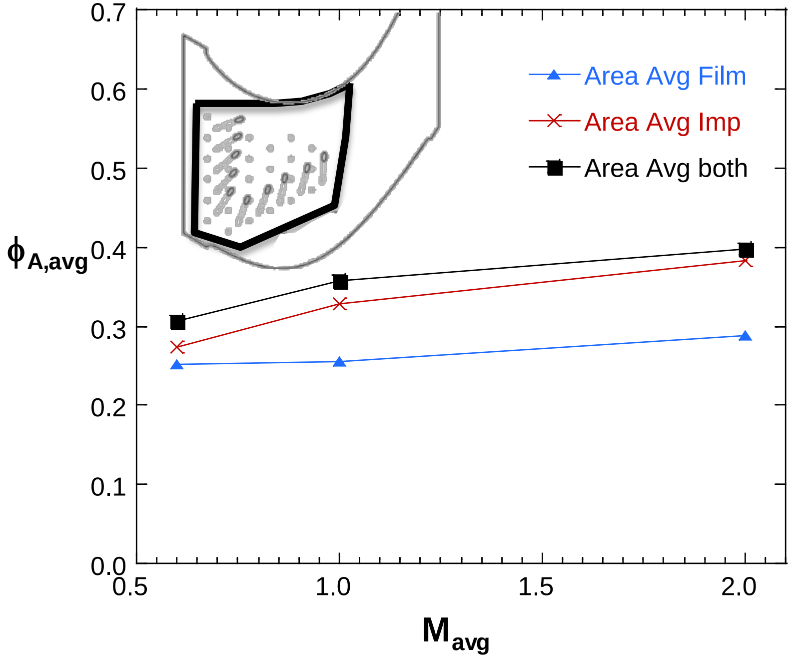


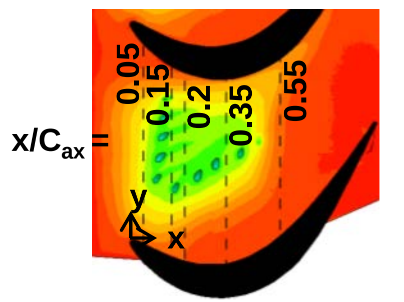


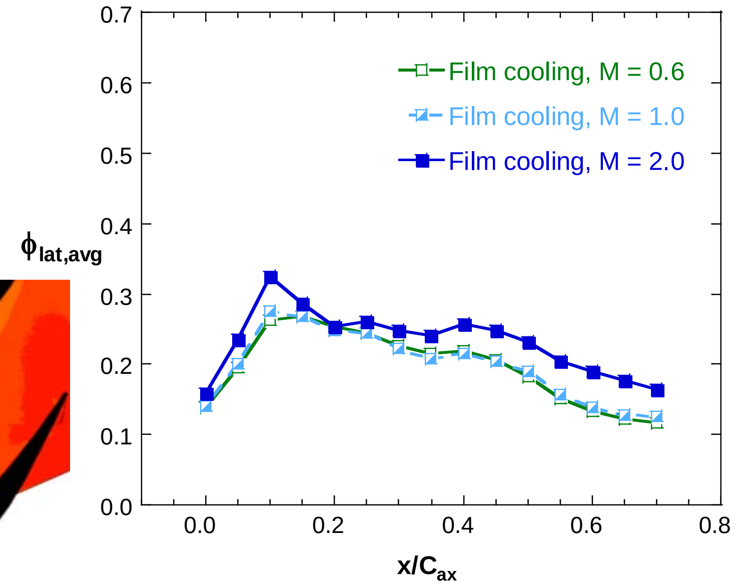
$$\phi = \frac{T_\infty - T_{wall}}{T_\infty - T_c}$$

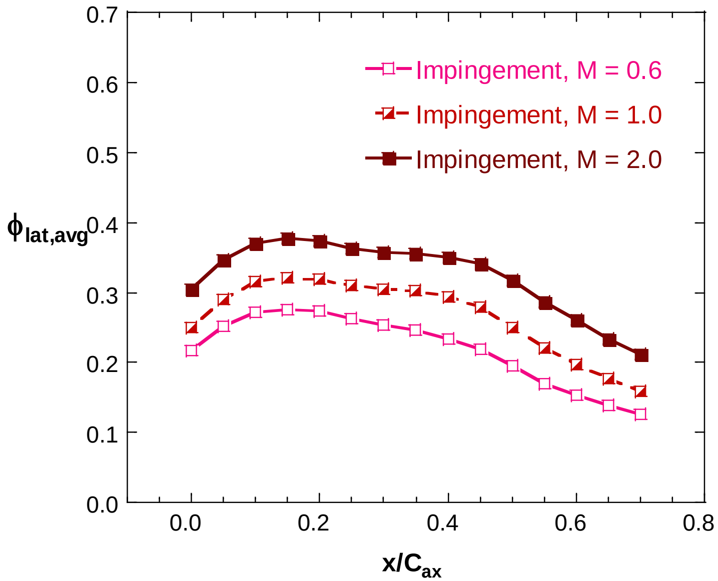


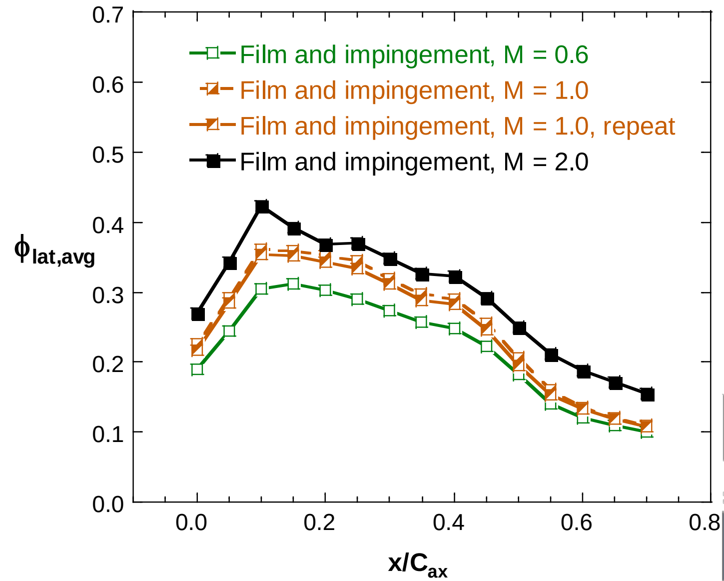
Increasing blowing ratio improved the average ϕ of impingement cooling more than film cooling



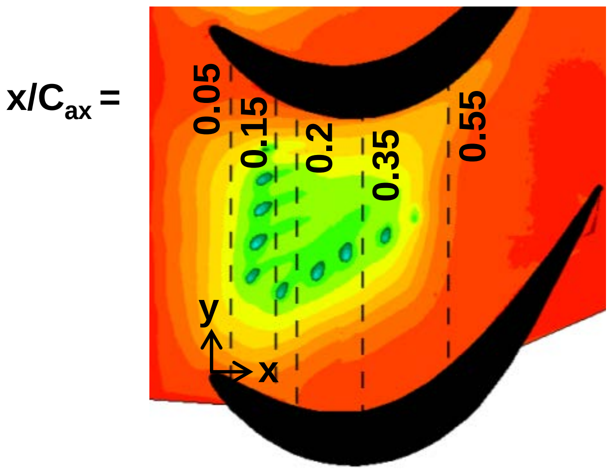


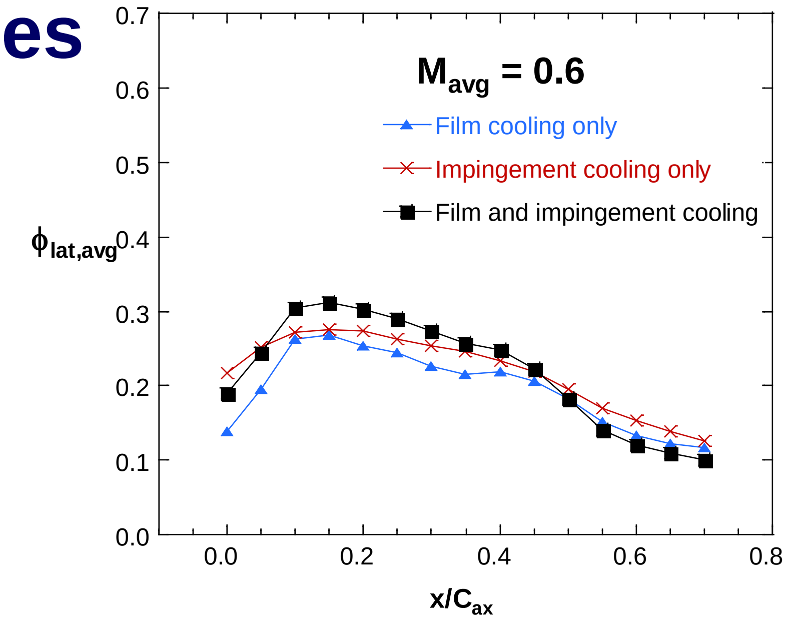


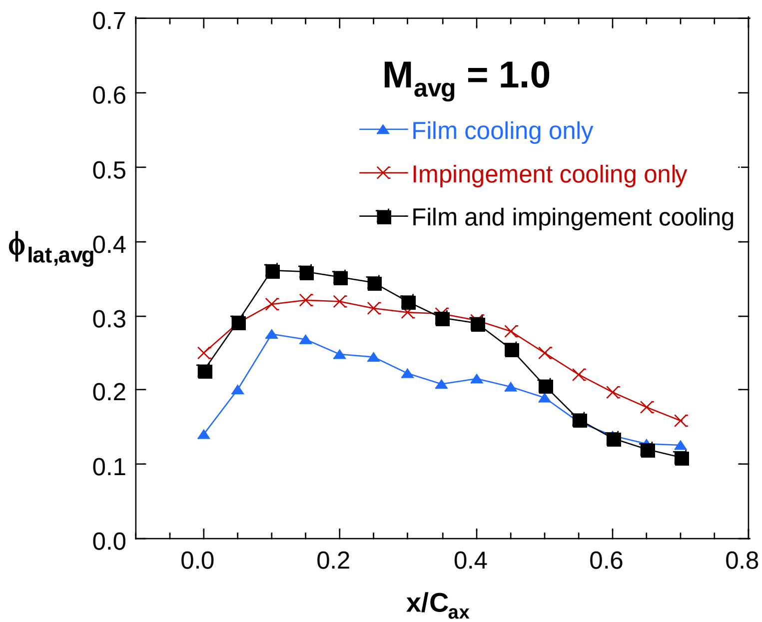


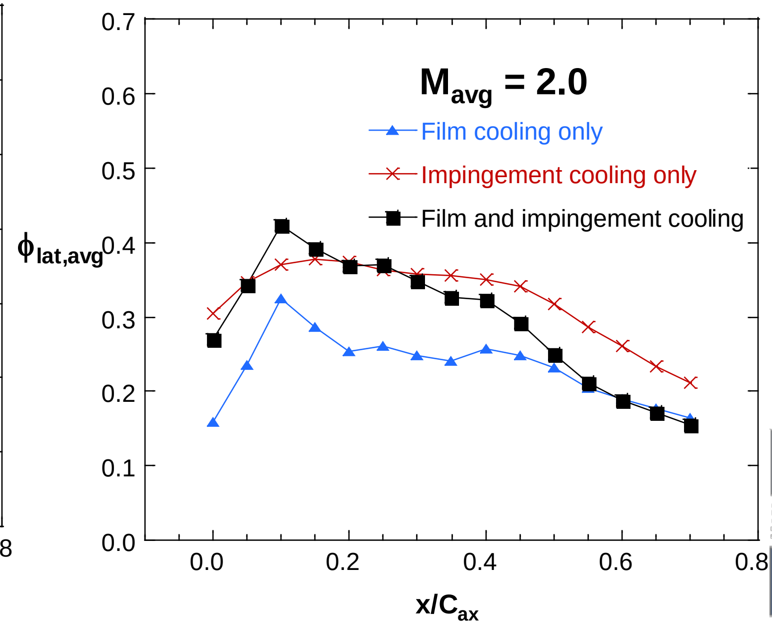


After x/C_{ax} ~ 0.5, ϕ is higher for impingement only than for both because the impingement jets are free to spread with no film holes

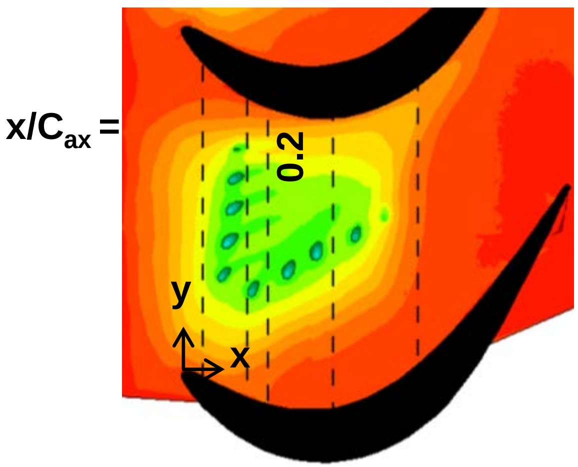


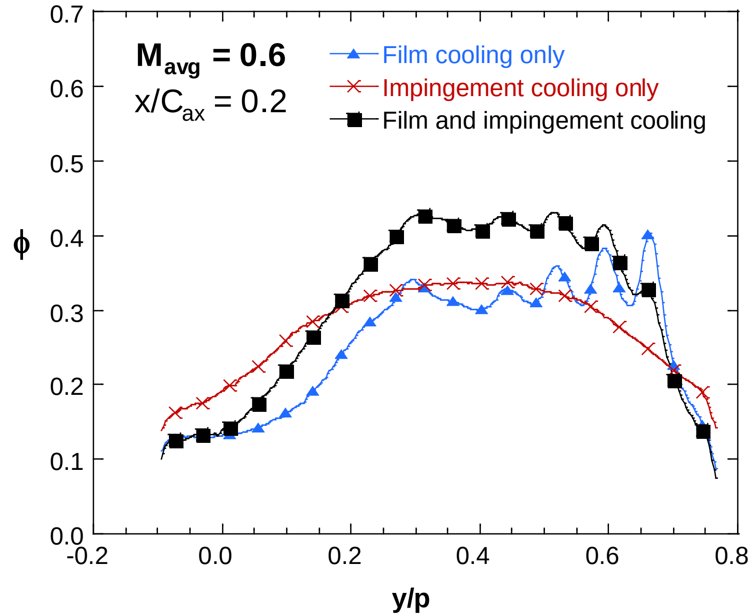


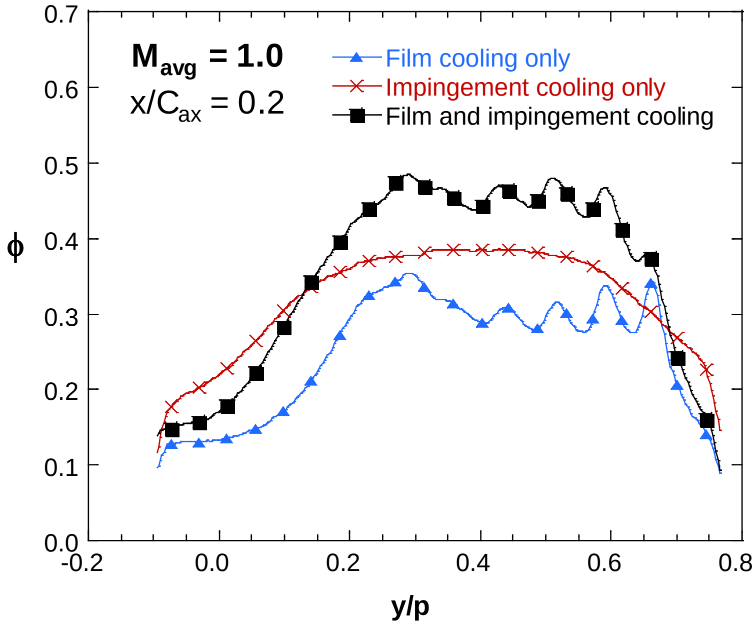


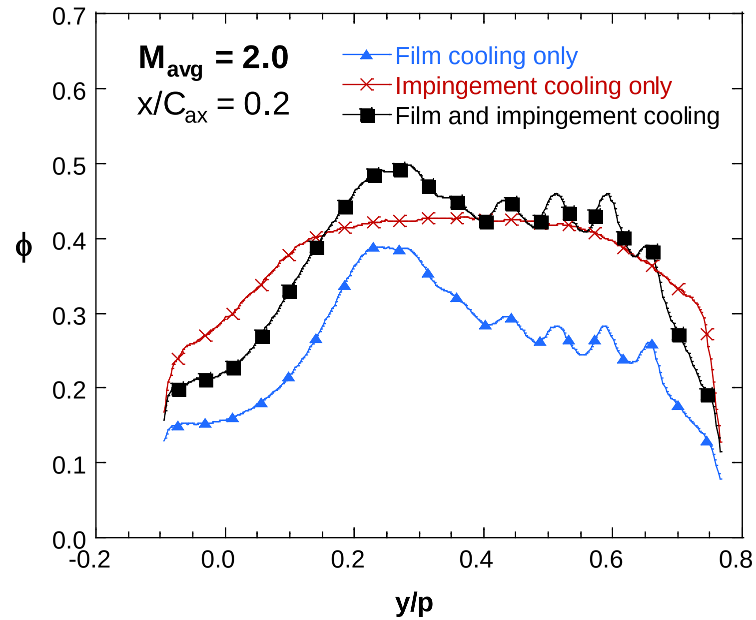


At x/C_{ax} = 0.2, the film cooling jets can be seen as well as the increase in ϕ from the impingement

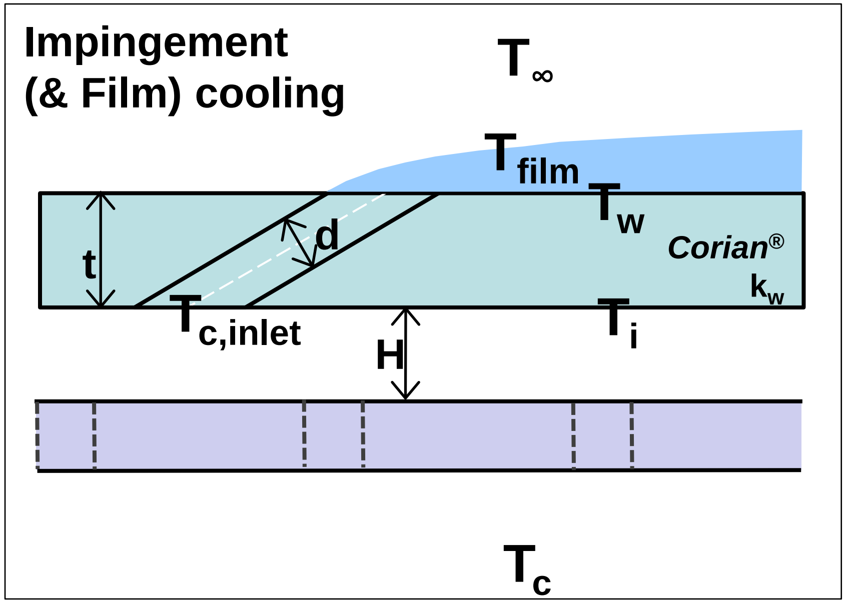


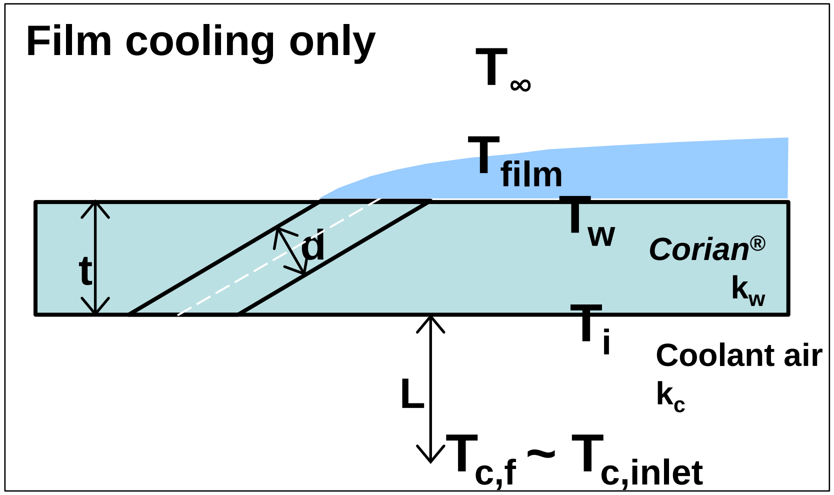






Applying a 1-D analysis, we estimated ϕ from the results from impingement and film alone





Impingement & film cooling , ϕ_{calc} :

$$q=\frac{T_{film}-T_i}{1/h_\infty+t/k_w}=h_\infty\left(T_{film}-T_w\right)$$

Impingement cooling only, ϕ_o:

$$q_o=\frac{k_w}{t}\left(T_w-T_i\right)=h_\infty\left(T_\infty-T_w\right)$$

$$Q_o=\sum q_o\,A=\dot{m}_c c_p\left(T_{c,inlet}-T_c\right)$$

Film cooling only, ϕ_f:

$$q_f=\frac{T_{film}-T_{c,f}}{1/h_\infty+t/k_w+L/k_c}=h_\infty\left(T_{film}-T_w\right)$$

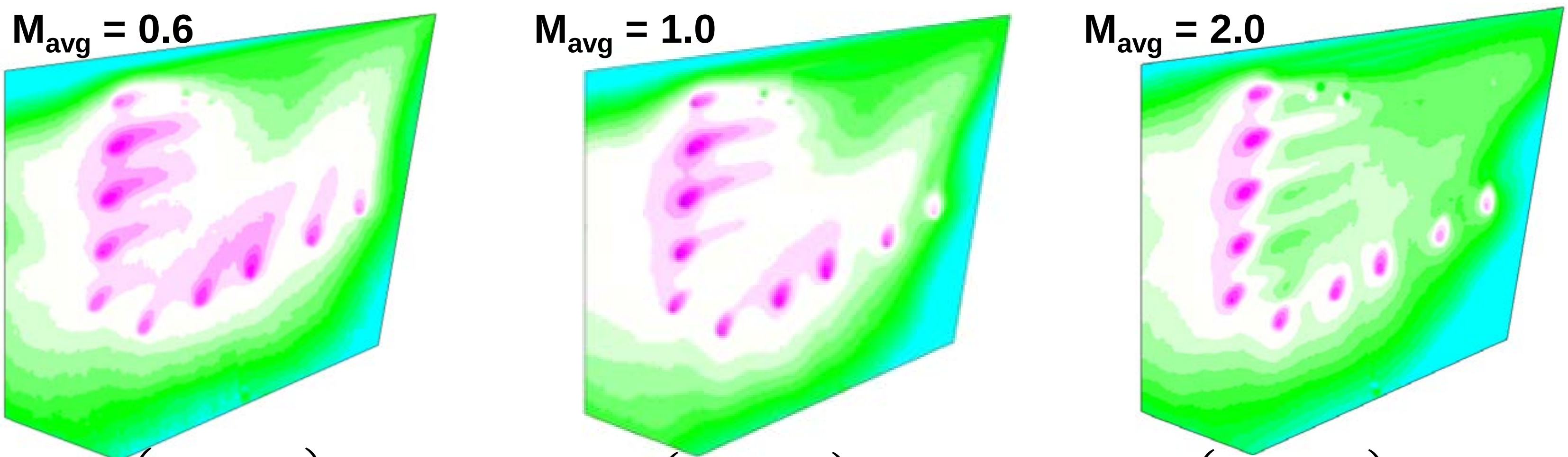
$$\phi_{calc}=\phi_o+\left(\frac{Bi}{1+Bi}\right)\left\{1-\frac{Q_o}{\dot{m}_c c_p\left(T_\infty-T_c\right)}\right\}\left[\phi_f-\left(\frac{\left(1-\phi_f\right)k_w k_c}{h_\infty\left(k_w L-k_c t\right)}\right)\right]$$

$$\phi_f=\frac{T_\infty-T_w}{T_\infty-T_{c,f}}$$

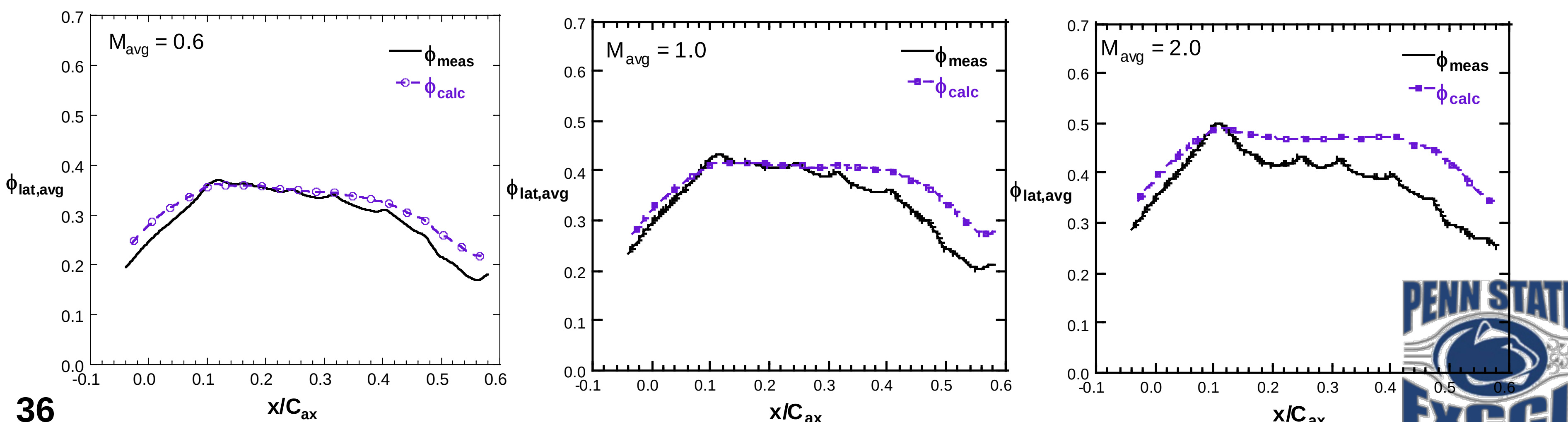
$$\phi_o=\frac{T_\infty-T_w}{T_\infty-T_c}$$

The relative difference between predicted and measured ϕ is < 5% for a large portion of M = 0.6, 1.0

Contours of relative difference $\dfrac{\phi_{meas} - \phi_{calc}}{\phi_{meas}}$

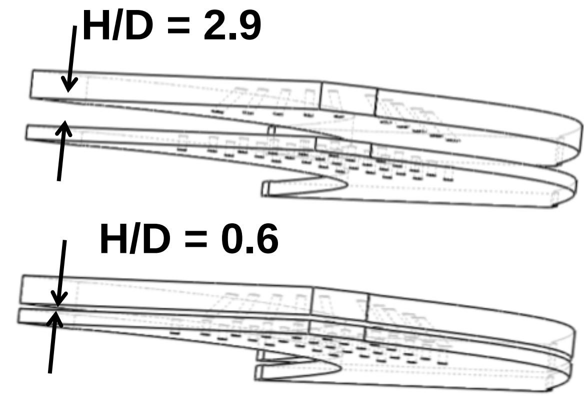


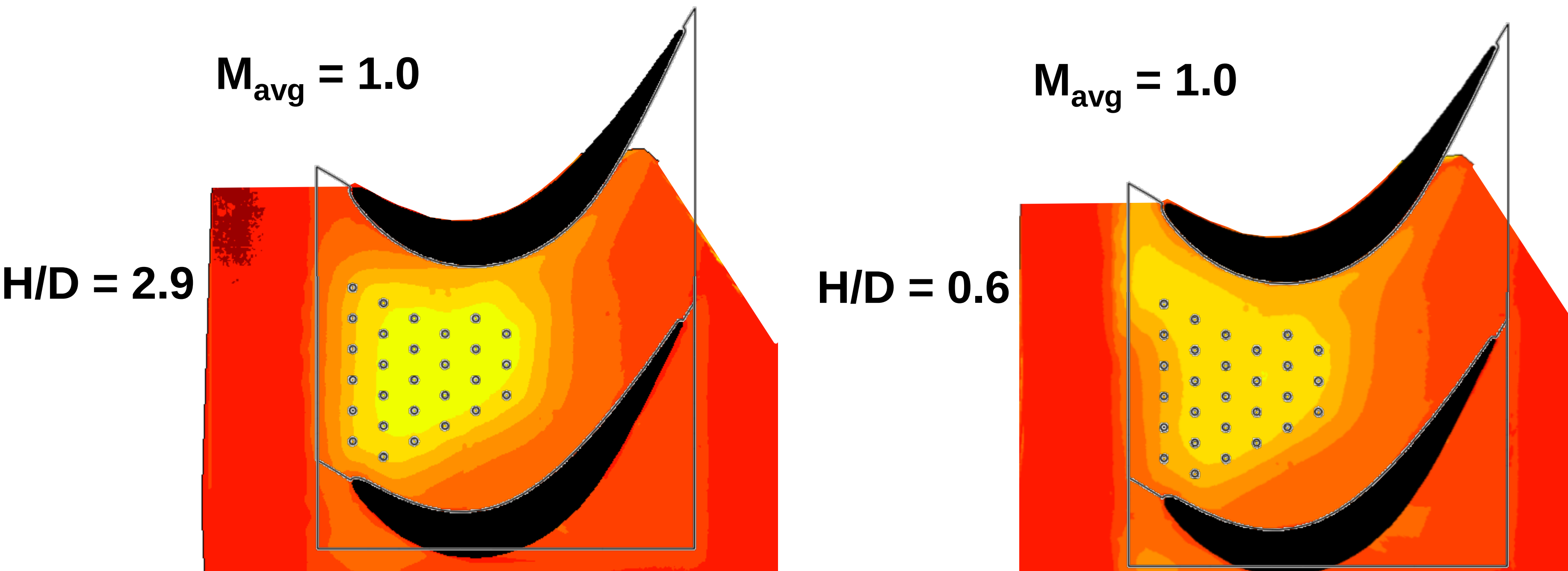
$$\left(\frac{T_\infty - T_{c,inlet}}{T_\infty - T_c}\right) = 0.75 \qquad \left(\frac{T_\infty - T_{c,inlet}}{T_\infty - T_c}\right) = 0.82 \qquad \left(\frac{T_\infty - T_{c,inlet}}{T_\infty - T_c}\right) = 0.88$$



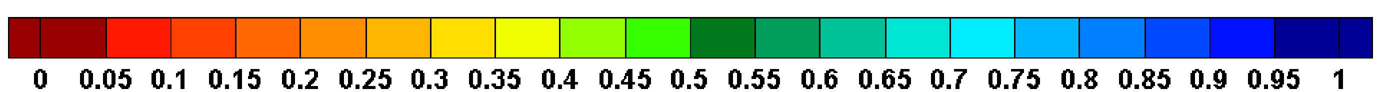
Reducing H/D to 0.6 changed ϕ shape for impingement only contours, due to higher average h_i

Plenum	H/D (imping.)	M_{avg}	I_{avg}	h_∞/h_i
film + impingement	0.6	2.0	3.50	0.5-1.1
		1.0	0.94	0.7-1.6
impingement	0.6	2.0	3.50	0.4-0.8
		1.0	0.94	0.5-1.2

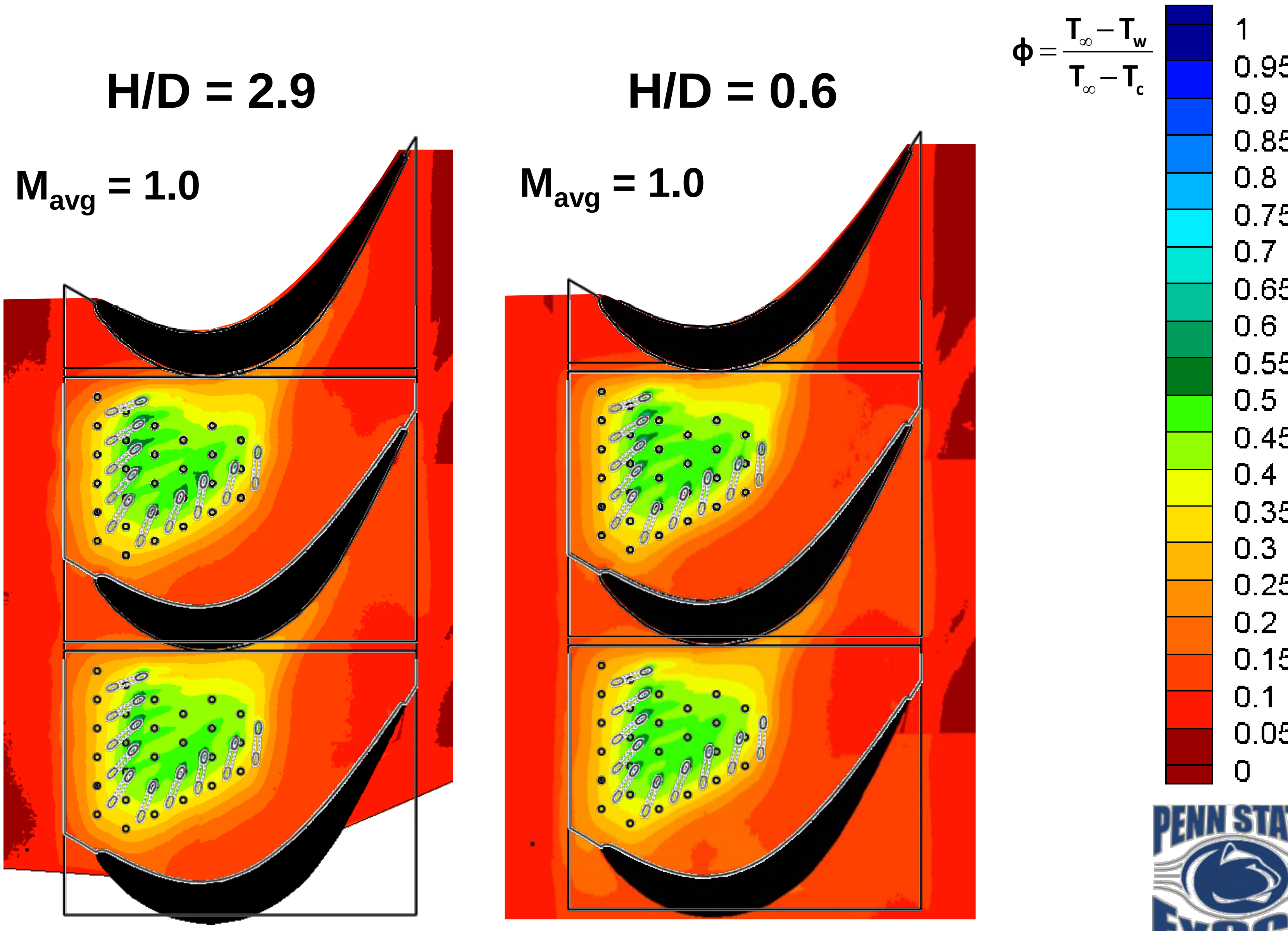




$$\phi = \frac{T_\infty - T_{wall}}{T_\infty - T_c}$$

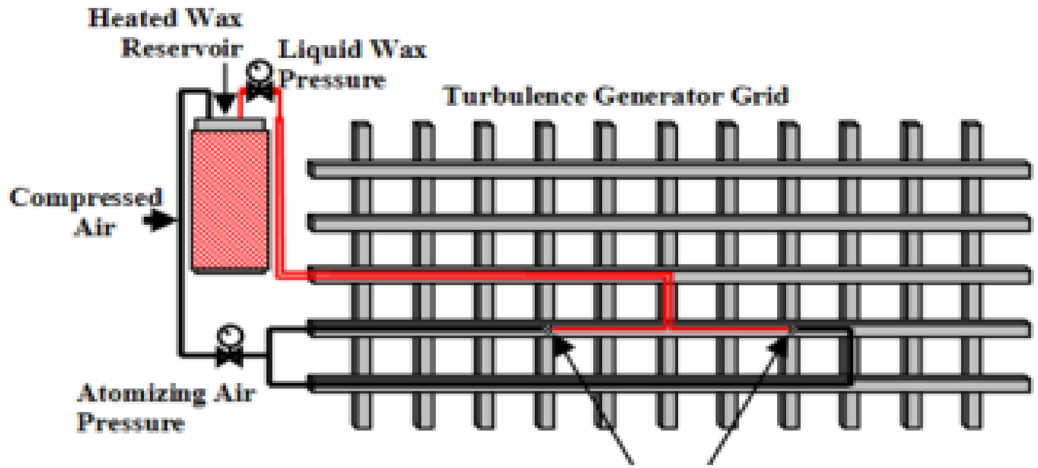


But with film and impingement, there was not much effect on ϕ when H/D was changed to 0.6

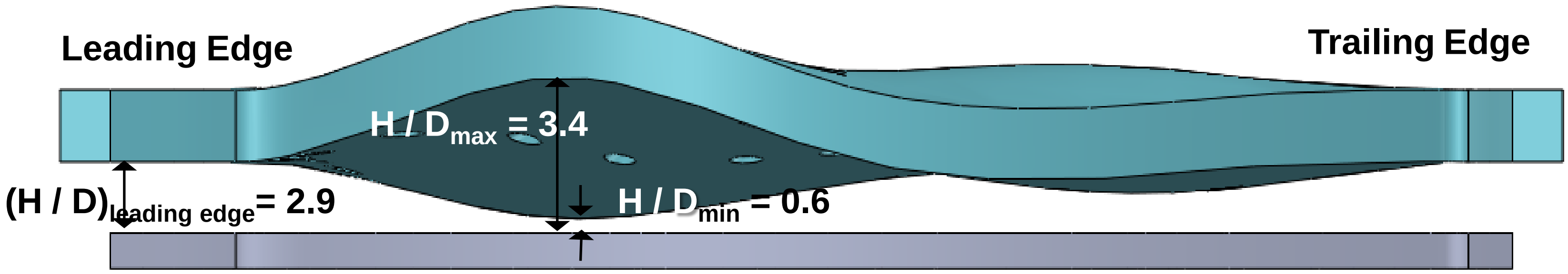


Next we will simulate deposition with wax, and test a contoured conducing endwall

Measure overall effectiveness with wax deposition



Examine effectiveness and deposition results with endwall contouring



PENN STATE ExCCL

Conclusions

Overall cooling effectiveness when using combined TBC and film cooling is dominated by the cooling effects of TBC

Deposits of contaminants significantly degraded the film effectiveness of realistic trenches, but had negligible effect on the overall effectiveness

Overall effectiveness measurements demonstrated the influence of conduction and convective cooling within the film cooling holes

Increasing blowing ratio resulted in a larger average increase in overall effectiveness for impingement cooling than film cooling

Questions?

Background image: [Hamed, A., Tabakoff, W., and Wenglarz, R., 2006]