The most remarkable changes occur for the spanwise flow angle distribution. For the low coolant mass flow rate the influence is limited to the near wall region, where a slight increase of the overturning is observed. This is interesting because in the solid wall case a decrease of the overturning was noticed from $M_2 = 0.1$ to 0.6 and 0.8. Important changes occur for the high injection rate. Both the underturning and overturning are greatly reduced such that the spanwise angle variation deviates very little from the mid span value. This is particularly striking for $M_2 = 0.8$ ($\Delta\beta_{max} = \pm 1.3$ deg). The secondary flow chart (Fig. 16) indicates indeed drastic changes with the secondary velocity vectors for $M_2 = 0.8$. These changes affect in turn considerably the iso-loss lines. Compared to the solid wall case, the high loss core on the wake suction side is considerably reduced. The reason is probably that due to the reduced secondary velocities, the transport of low energy material from the end wall to the suction surface has significantly decreased and hence also the interference losses between this cross flow and the main flow over the blade suction side. In addition to this effect there is of course the direct effect of the injected high energy coolant flow.

Contrary to the high coolant mass flow rate ($\dot{m}/\dot{m}_{tot} = 0.045$), the iso-loss lines are little modified by the small coolant mass flow rate ($\dot{m}/\dot{m}_{tot} = 0.02$). In the spanwise loss distribution, the losses remain practically unchanged for $M_2 = 0.8$ while at $M_2 = 0.6$ there is even a local increase of losses around the wall distance $2y/h = 0.1$.

Conclusion

The secondary flow losses are a form of mixing and skin friction losses which predictably increase with Mach number. The difficulty with secondary flow losses is that the phenomenon increasing the skin friction losses and enhances the mixing losses is Mach number dependent and is at present not predictable.

The test data also clearly indicate a significant influence of end wall cooling air injection upon the losses and exit air angle distribution. The data suggest that the influence of cooling air should be considered in the selection of the optimum blading angle as well as in the prediction of turbine efficiency. Three cooling air parameters should be considered as significant: the coolant to main stream total pressure ratio, the coolant mass flow ratio and the angle between coolant flow, main flow and end wall boundary layer.

The nonlinear effect of the coolant on both the losses and the outlet of flow angle suggests that we are in presence of an injection pressure ratio effect rather than a mass flow effect. Similar observations have been made for trailing edge coolant ejection.

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DISCUSSION

G. J. Hanus[1]

The authors suggest that a possible benefit of end-wall cooling, in addition to thermal protection, is the dynamic interaction effect of the coolant jets and the end-wall core flow. This effect is manifest through a shifting of the streamlines of the core flow near the end-wall due to a coolant-jet, core-flow streamline deflection phenomenon. Results are shown for both spanwise loss as well as trailing edge flow angle distribution for two end-wall cooling fractional flow rates. These results tend to support the possible hydrodynamic benefit of strategically placed end-wall cooling configurations. A note of caution, however, must be included on how the coolant test condition relates to the reported physics of the primary flow. Results of discharge flow angle and loss distribution for the test cases, in which $T_{01} = T_{0c}$, may not be representative of the distributions one would find in an actual engine application in which $T_{01} \cong 2T_{0c}$.

Thermodynamic and hydrodynamic modeling of high-temperature turbomachinery component flow at reduced flow conditions has been successfully applied for many years [1, 2]. By carefully choosing P_{01}, T_{01} and T_{0c}, proper geometric scaling can produce thermodynamic and hydrodynamic similarity between actual engine conditions and reduced-scale laboratory tests. Although the authors have maintained the proper conditions for local Reynolds number simulation over the blade surfaces (assuming the local Mach number distributions were matched with the modeled NASA cascade), they have failed to simulate realistic end-wall coolant jet/end-wall boundary layer interaction by allowing the total temperature of the coolant and mainstream to be identical. The reported over and underturning of the discharge angle as well as the loss distribution for the two fractional coolant flows would not be representative of what would occur in an actual engine environment at these coolant flow ratios. For the same fractional coolant flow conditions under high-temperature turbine operation, the physics of the coolant jet/end-wall boundary layer flow, though qualitatively similar may be quantitatively dramatically different.

To simulate the proper coolant-jet/end-wall boundary layer interaction, the following constraints must be met.

1 $$\left(\frac{\rho_c V_c}{\rho_\infty V_\infty}\right)^{\text{engine}(E)} = \left(\frac{\rho_c V_c}{\rho_\infty V_\infty}\right)^{\text{simulation }(S)}$$

2 $$\left(\frac{\rho_c V_c^2}{\rho_\infty V_\infty^2}\right)^{E} = \left(\frac{\rho_c V_c^2}{\rho_\infty V_\infty^2}\right)^{S}$$

3 $$\left(\frac{\theta}{d}\right)^{E}_{\text{inj}} = \left(\frac{\theta}{d}\right)^{S}_{\text{inj}}$$

Where θ = momentum thickness, d = coolant hole diameter, ∞ = free stream, inj = at injection site, and c = coolant conditions.

The authors have chosen item 3 as the proper ratio as might be expected in an engine environment. However, combining 1 and 2, and assuming the same medium is used to simulate the engine environment, the following condition must be met to provide the proper dynamic simulation of the end-wall cooling phenomenon,

$$(T_{0c})^S = \frac{(T_{0\infty})^S}{(T_{0\infty})^E}(T_{0c})^E$$

The authors have chosen $(T_{0c})^S = (T_{0\infty})^S$ which clearly implies that $(T_{0c})^E = (T_{0\infty})^E$, a condition which is unrealistic.

Assuming that the ratio of total coolant hole area to core flow area is representative of an engine configuration and that the injection momentum flux ratio is the dominate parameter in jet/core flow interaction, the results of Fig. 15 for the ejection mass flow ratios of 0.02 and 0.045 are actually more representative of what one might expect for an engine environment under a cooling condition of $(\dot{m}/\dot{m}_{TOT})^S/\sqrt{\alpha}$, where $\alpha \equiv \rho_c/\rho_\infty$. Since the jet trajectories (and hence jet/end-wall core flow streamline interaction) are greatly influenced by the momentum flux ratio, caution should be exercised in trying to incorporate the results of Fig. 15 into any design process.

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Authors' Closure

We agree with Dr. Hanus that it would be desirable to properly simulate the momentum flux ratio. However, it is difficult to perform as detailed measurements as described in this paper, with main stream temperatures which might be three times as high as the cooling flow temperatures. This difficulty might eventually be overcome by using mixtures of gases of different densities. Such tests are at present under way at VKI. The results will be published in due time.