Installation Effects on Heat Transfer Measurements for a Turbine Vane

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Experiments have been conducted for a modern high pressure turbine vane in the USAF Turbine Research Facility (TRF). This turbine vane was instrumented with two types of heat flux gauges. The first was a thin film Upilex gauge design wrapped over the full airfoil surface, while the second consisted of Pyrex insert type gauges. Initial measurements showed significant discrepancies in the resultant Nusselt number from these two types of gauges. As the inlet Reynolds number was decreased, these discrepancies collapsed to single laminar level Nusselt number. This suggested a potential change in the flowfield for one of the gauge types. Interrogation of the vane surface metrology showed depressions in the epoxy filler between the Pyrex insert and the metal airfoil. After filling these depressions, the vane was reinstalled and identical tests repeated. This second test revealed a better comparison for two of the Pyrex insert gauges while minimal change resulted in two other gauges. Similar results were achieved for the range of Reynolds numbers tested in laminar, transitional, and turbulent regions for the airfoil. These results suggest that further installation effects may still be present in the Pyrex gauges. Such alterations in the flow field can lead to misleading results and costly errors when applying experimental rig data to turbine design.

NOMENCLATURE

h	Heat transfer coefficient
k_g	Conductivity of the gas
k	Roughness height
Nu	Nusselt number
Pr	Prandtl number
q''	Heat flux (per unit area)
Re	Reynolds number
T	Temperature
U_∞	Velocity
v^*	Friction velocity
Greek	
δ	Boundary layer thickness
ρ	Density
μ	Absolute Viscosity
ν	Kinematic Viscosity
Subscripts	
in	Inlet conditions
tf	Thin film heat flux gauge
p	Pyrex heat flux gauge
s	Surface length
t	Total
w	Wall

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INTRODUCTION

Surface mounted instrumentation has been the primary means of acquiring aerothermal data in turbomachinery for many years. Technologies such as pressure gauges, strain gauges and heat flux gauges have been mounted directly to compressor and turbine vanes and blades in order to acquire high accuracy, high frequency response data. Typically the installation procedure has involved either mounting the gauge on top of the airfoil surface or embedding the gauge into the surface. Both of these techniques involve risks that the installation itself has not affected the measurement. With the small size of the airfoil surfaces, boundary layer thicknesses are quite small and any local surface discrepancy can easily result in a local tripping of the flow. With the gauge just downstream of this trip, the resultant measurement can be affected. Therefore extreme care must be taken to ensure that the installation is flush with the airfoil surface.

There have been a few experiments in the literature that have shown the impact of a discontinuity in the surface at the metal to gauge interface. Peabody and Diller (1998) used a single heat flux gauge mounted on a block that could be traversed perpendicular to the metal surface. With this set up they could change the step size at the interface and interrogate the resultant heat flux. They compared the heat transfer over a gauge with step heights (protrusion or recession) of 0.25 mm. This was significantly larger than the approach boundary layer thickness of 0.10 mm. Both the protrusion and the recession of the gauge significantly increased the heat transfer as would be expected.

What was also shown was that the 'flush' gauge also resulted in an increase in heat transfer over the directly deposited gauge. While this was not discussed in the paper, it is proposed to be a result that the gauge was still not truly flush to the surface. With the small boundary layer thickness, even a nonuniformity of 0.01 mm could be enough to change the flow pattern. This conclusion is drawn from the work of Liess (1975) who stated that a trip had to be greater than 1/10 δ in order to start transition for a boundary layer. Therefore a small protrusion or even a gap could cause a local trip to turbulent flow and a higher resultant heat flux.

Criteria for the flushness of gauge installation can also be correlated to roughness effects. Schlichting (1955) provides for a critical roughness size for transition of a laminar boundary layer of

$$Re_k = v^* k_{crit} / \nu = 15 \tag{1}$$

Roughness Reynolds numbers (Re_k) above this value will result in transition. In the case of an individual gauge, a true rough surface is not present but Scherbarth (1942) showed that behind single obstacles wedge like turbulence disturbances occurred with spreading angles of 14 to 18 degrees. Experiments have also shown that a trip wire can transition a laminar boundary layer to turbulent early. The relative size of the trip height determines how quickly the boundary layer will transition. For a trip to be fully effective at tripping a laminar boundary layer (i.e. transition immediately) Gibbings (1959) provides the relationship of

$$k_{trip} U_\infty / \nu = 850 \tag{2}$$

for a low Mach number flow. This critical value increases towards ~ 1200 for a Mach number of 1. Another relevant parameter is the relationship between the trip height and the displacement thickness. As the ratio k / δ^* increases toward 0.3 the point of transition moves forward from the no trip location to just behind the trip.

Measuring the correct heat transfer is essential for turbine designers. With the elevated freestream temperatures that are run in current engines, keeping the turbines thermally protected is one of the primary design concerns. Film cooling flows are tailored to maintain metal temperatures below melting temperature. This is primarily accomplished by adjusting the amount of coolant to cover the worse heat loads on the airfoil surface. This usually results in overcooling certain areas to ensure a well protected airfoil which results in a hit in overall turbine performance.

The designers ability to know the correct amount of required cooling flow hinges on knowledge of the heat transfer rate to the airfoil among other things. Uncertainty in these values result in an improper design and therefore durability problems with the turbine. This ultimately reduces the life of the turbine. Figure 1 (Haubert, 1980) shows the impact of various parameters on the life of turbine hardware due to rupture. From a heat transfer standpoint the primary variables that went in to this figure are variations in h - gas side of ± 10%, h - cool side of ± 10%, hole diameter of ±0.025 mm, $T_{coolant}$ of ± 11 K, and T_{gas} of ± 55 K. These variations represent typical uncertainties in the measurement of each parameter. Obviously from the figure, gas side h is the primary driver for rupture life durability and accurate measurements of this quantity are vital. Therefore ensuring that the instrumentation is obtaining the correct data is essential.

EXPERIMENTAL FACILITY

The experimental apparatus used in this program was the Air Force Research Laboratory Turbine Research Facility (TRF) (Haldeman, et al, 1992) a transient "blow down" device capable of operating for periods of several seconds at the correctly scaled Mach and Reynolds numbers of the turbines of actual engines. (The reference cited above describes the TRF by its former name, the Advanced Turbine Aerothermal Research Rig.) The TRF, shown schematically in Fig. 2, operates as follows: before the start of a test the entire facility is evacuated and the fast acting valve between the supply tank and test section is closed. Next, the supply tank can be filled with a mixture of N_2 and CO_2 whose composition is set to simulate the same specific heat ratio (γ) as that of vitiated air at the engine operating conditions. For these tests only nitrogen was used resulting in $\gamma = 1.4$ and a $Pr = 0.71$. This is reasonably close to the desired Pr of high temperature air, about 0.68.

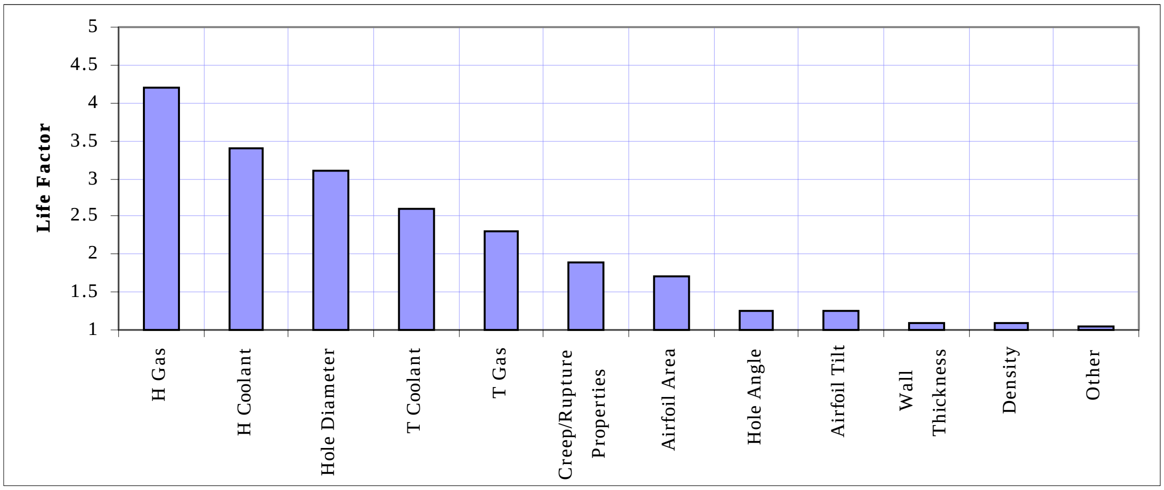

Figure 1 Rupture Life Factors

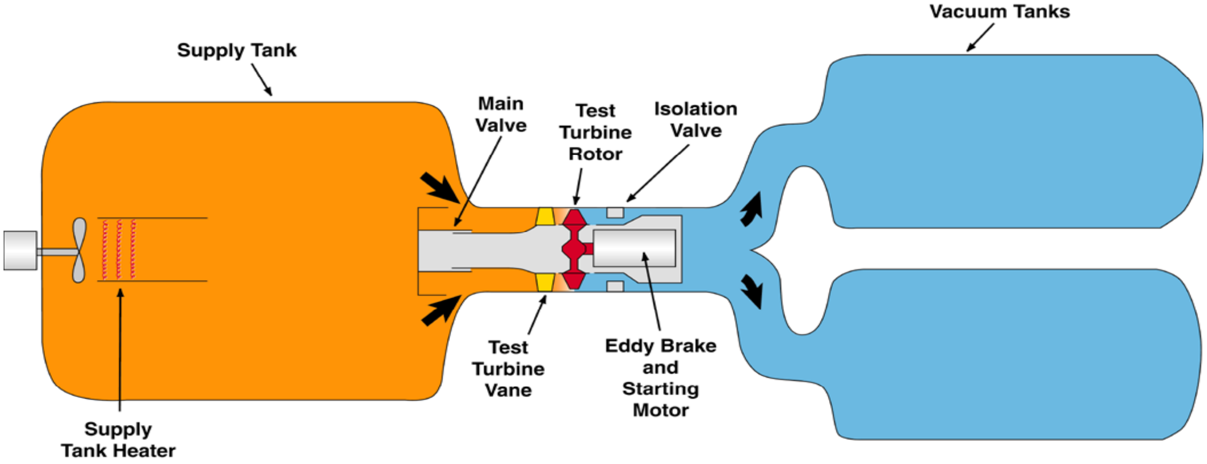

Figure 2 Schematic of Turbine Research Facility

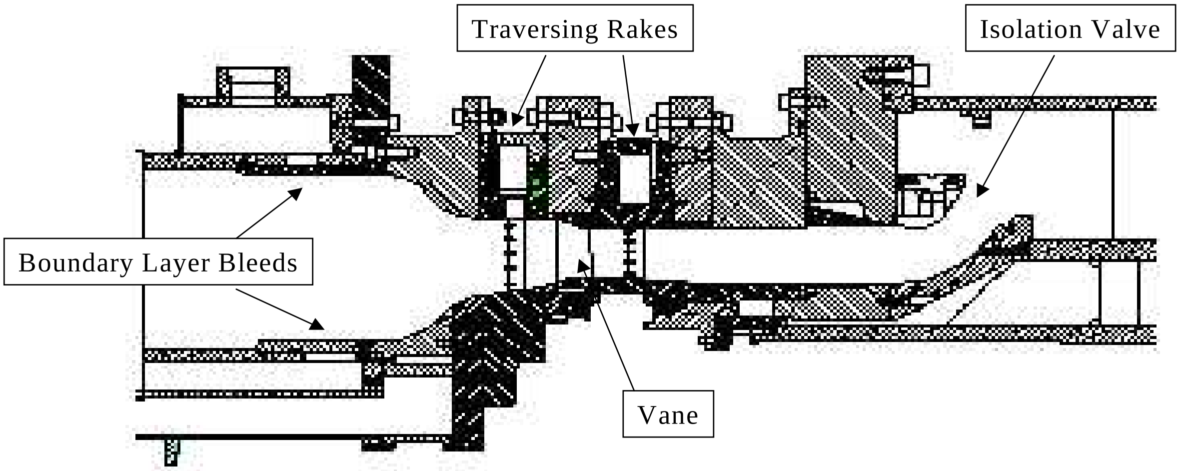

Figure 3 Schematic of Test Section

The mixture is then heated with an internal electric heater until the gas temperature to room-temperature metal ratio, representative of the engine ratio is reached. The pressure in the tank is then adjusted by adding or removing gas until the density is such that the turbine Reynolds (based on some convenient length scale such as blade chord) again matches the engine value. Also before the start of the run a choke valve downstream of the turbine is set to the proper throat area to control the pressure ratio across the vane when the gas is released from the tank.

The test is initiated by opening the fast acting supply tank valve. This enables the flow to proceed through the turbine vane for up to five seconds at the correctly scaled conditions, depending on the flow size of the turbine. The only one of the five controlling non-dimensional parameters (Re, M, γ, T_t/T_m, Pr) that changes significantly, about 20% over a typical 2 second run is the Reynolds Number because of the decreasing density of the flow. Fortunately the data is acquired with a high speed data system that can take thousands to hundreds of thousands of samples per second resulting in significant data over a finite range of Reynolds number. These small windows are then averaged to create nearly 'steady state' data. The data to be discussed later was all averaged over a window from 0.40 to 0.43 seconds after the main valve started to open. This time is necessary to allow the valve to fully open and for start up transients (including the compression wave) to pass through the system.

A cross-section of the test section holding the turbine is shown in Figure 3. Air flowing from the heated supply tank undergoes a substantial contraction before entering the vane set. Total temperatures and total pressures are each measured with two rakes of five element probes in this inlet plane. Behind the vane a similar set of two five probe rakes that measure the exit temperatures and pressures. These rakes may be traversed 90 degrees in the circumferential direction during the test to sample the flow about 1.6 chord length behind the vane.

INSTRUMENTATION

The current investigation was a result of discrepancies found between two types of heat flux gauges. A vane ring had been instrumented with both Pyrex insert gauges and thin film heat flux gauges. The insert gauges consist of 0.01 micron thick platinum hand painted onto a Pyrex 7740 insulating substrate. These pieces of Pyrex are shaped to the airfoil profile and embedded into the metal airfoil at 50% radial span. A more complete description along with the data analysis of these types of gauges can be found in Dunn et. al., 1984. One vane passage was instrumented with four suction side, a stagnation, and two pressure side Pyrex insert gauges. The Pyrex gauges actually consisted of three glass strips. One strip contained gauges at $s/s_t = 0.0$ and -0.471 along the pressure side. A second strip started at $s/s_t = -0.776$ and contained a gauge at -0.878 along the pressure side ($s_t = 5.65$ cm). The other strip started at $s/s_t = 0.251$ and contained gauges at 0.284, 0.441, 0.671, 0.880 along the suction side ($s_t = 7.32$ cm). Figure 4 shows this suction side strip of gauges. These strips were about 2.75 mm wide and were recessed into the airfoil surface at least 2.25 mm. The heat flux is determined from these gauges based on the Pyrex being semi infinite and as such assumes that the thermal wave does not penetrate the thickness of the Pyrex insert. The 2.25 mm ensures that the Pyrex is semi infinite for at least 1.5 s of testing.

Figure 4 also displays three surface mounted pressure transducers on a neighboring vane. These transducers are high frequency response gauges that are flush mounted with the airfoil surface and epoxied in place. These gauges are located at essentially the same surface location as the heat flux gauges. This provides knowledge of the local static pressure which enables calculation of the local static temperature via a Mach number calculation and the assumption of constant total properties through the vane passage. The local static temperature is needed to calculate the gas properties which are needed to calculate the local Nusselt number.

A separate vane was wrapped with a 50 micron thick film sheet of Upilex (Figure 5). This sheet was wrapped from trailing edge to trailing edge of a single vane creating no surface discontinuities along the airfoil surface. Thirteen 0.04 micron thick platinum sensing elements were sputtered on top of the Upilex prior to installation on the vane. The leadwork consisted of vapor deposited copper. This sheet contained gauges at nearly identical locations as the Pyrex gauges as well as additional gauges nominally in-between each of the original seven gauges. The heat flux is determined from reducing the measured temperature traces using a layered analysis in the Laplacian domain consisting of the finite thickness of the Upilex and the semi-infinite metal substrate. More detail of this type of heat flux gauge can be found in Jones, 1995.

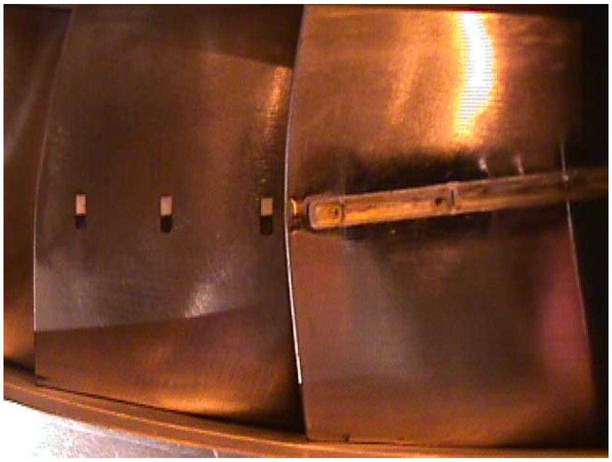

Figure 4 Vane Suction Side Pyrex Heat Flux Gauges and Pressure Transducers

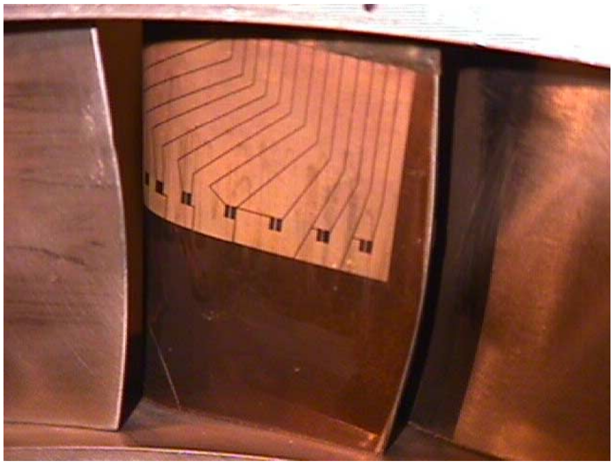

Figure 5 Vane Suction Side Thin Film Heat Flux Gauges

DATA ANALYSIS AND UNCERTAINTY

The heat flux data from both types of gauges was determined by reducing the resistance change with time to a temperature trace via a calibration for resistance to temperature. The heat flux was then calculated based on the time rate of change of the surface temperature coupled with the thermal properties of the Pyrex or Upilex and metal substrates. The details of the reduction scheme for the Pyrex gauges is provided in Weaver et al., 1994 with a similar discussion provided by Joe, 1995 for the thin film gauges. Once the heat flux is known, the heat transfer coefficient is calculated via the following equation:

$$h = q'' / (T_t - T_w) \tag{3}$$

where T_t is the total freestream temperature and T_w is the surface temperature at the gauge. With h calculated, the Nusselt number is calculated based on the local surface length, s, as:

$$Nu_s = h * s / k_g \tag{4}$$

except for the stagnation gauge which utilizes the diameter of the leading edge (12.2 mm) as the length scale. The conductivity and all gas property data is determined based on the mean temperature between the metal temperature and the local static temperature. The local static temperature is calculated via pressure instrumentation mounted at the same location as the heat transfer gauges, albeit on different airfoils as mentioned above. The pressure distribution is shown in Figure 6 along with a prediction.

More detail of the pressure measurements can be found in Joe, et al, 1998. From these pressure transducers the freestream velocity is known enabling the local Reynolds number to be calculated as:

$$Re_s = \rho * U_\infty * s / \mu \qquad (5)$$

Based on the uncertainties in the measurement of the surface resistance and the uncertainty in calibration from resistance to temperature the surface temperature uncertainty was calculated to be about 0.8 K for the Pyrex gauges and 1.7 K (95% confidence) for the thin film gauges via the propagation of errors method (Kline and McClintock, 1953). A jitter analysis was then performed (Moffat, 1982) through the reduction scheme to obtain the uncertainty in the surface heat flux. This resulted in uncertainties in q between 2740 and 6230 W/m^{2}K depending on the gauge and location. With q and Tw, the heat transfer coefficient uncertainty was determined via a propagation of errors to be about 50 W/mK for the Pyrex gauges and 100 W/mK for the thin film gauges based on an uncertainty in T_∞ of 0.7 K. This resulted in Nusselt number uncertainties between 5 and 9% for both types of gauges depending on location.

RESULTS

An initial test sequence was performed that varied the Reynolds number nearly an order of magnitude. A test window of 0.40 to 0.43 seconds after the main valve fired was used for calculating the ‘steady state’ average data for these runs. The Reynolds number variation was accomplished by varying the inlet pressure to the vane ring. For all these runs the inlet temperature was about 402 K over the averaging window and were performed at a total to static pressure ratio of 1.59. Only the lowest Reynolds number run had a different pressure ratio of 2.2. Figure 7 shows the results of this Reynolds number excursion for both the Pyrex and the Thin Film heat flux gauges.

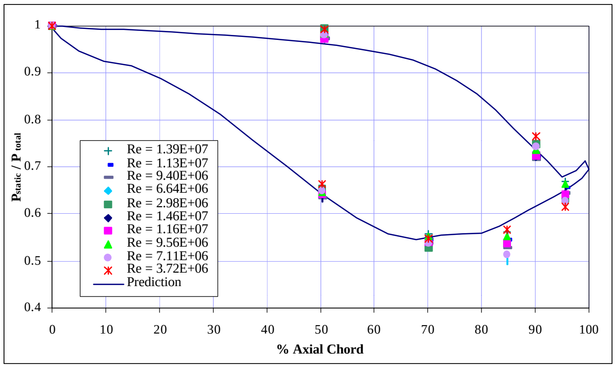

Figure 6 Pressure Distribution: Data and Prediction

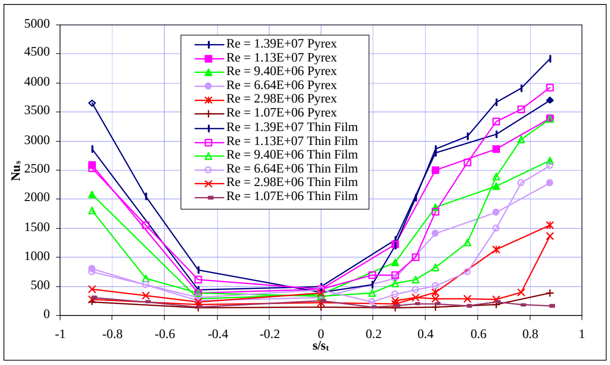

Figure 7 Initial Nusselt Number Distribution

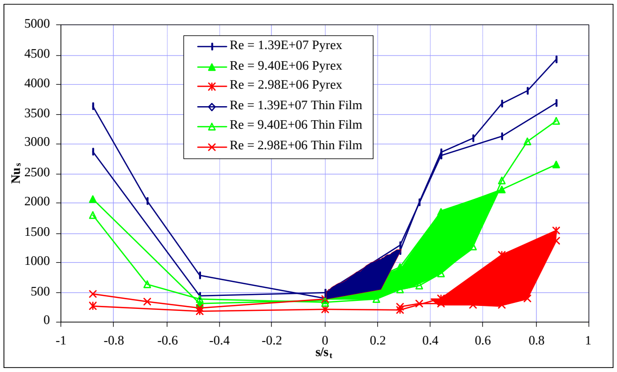

Figure 8 Transitional Region of SS Heat Transfer

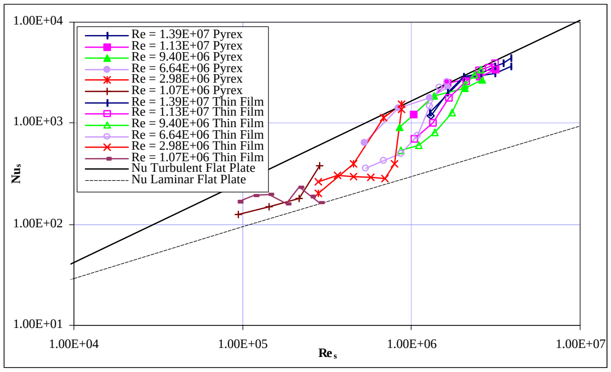

Figure 9 Local Nu vs. Local Re Comparison

As the Reynolds number increased, both types of gauges show steadily increasing Nusselt numbers as expected on both surfaces of the airfoil. For the pressure surface, these increases occur consistently and the two types of gauges show similar results. On the suction side, both types of heat flux gauges result in nearly the same Nusselt number at low inlet Reynolds number. Likewise, both the thin film and the Pyrex gauges indicate the onset of transition which continues to move forward along the vane surface as the Reynolds number increases. In dispute is the actual location of the transition point.

The Pyrex gauges typically have a faster increase in Nusselt number, suggesting that transition occurs sooner than is measured with the thin film gauges. Figure 8 highlights this early transition by comparing the data for only three of the runs on the suction side. It is apparent in this figure that the Pyrex gauges have transitioned earlier than the thin film gauges (the cross hatched region). To understand the transitional nature of the suction side flow, the data was compared to laminar and turbulent flat plate correlations as given:

$$Nu_{laminar} = 0.332 * Re_s^{0.5} * Pr^{0.33} \tag{6}$$

$$Nu_{turbulent} = 0.0296 * Re_s^{0.8} * Pr^{0.33} \tag{7}$$

Plotting the Nusselt number data versus the Reynolds number as in Figure 9 suggests that the Pyrex gauges may actually skip transition completely and become fully turbulent. A possible explanation for this result is that the boundary layer was locally tripped at the Pyrex to metal interface resulting in a turbulent boundary layer for Pyrex gauges.

Computational Investigation

To investigate why the discrepancy between the two types of gauges existed several items were investigated. Both an Euler Navier-Stokes CFD code (Ni, 1982) and STAN5 (Crawford, 1976) were run to predict the flow pattern over the vane. Two cases were run with the CFD code: $Re_{in}/l = 9.4e6\ m^{-1}$ and $Re_{in}/l = 1.1e6\ m^{-1}$. All cases were run with STAN5. The CFD code revealed streaklines that pass over the interface between the Pyrex and the metal. This is showed schematically in Figure 10. With this crosstream, the possibility exists that the flow could be tripped at the interface. Tripping is a function of the relationship of the step height to the boundary layer thickness as discussed earlier by Liess (1975). Therefore, both of these codes were used to calculate the local boundary layer thickness. Interrogation showed a nominal boundary layer height of 0.05 mm for the higher Reynolds number and a height of 0.10 mm for the smaller Re for both codes. This corresponds to displacement thicknesses of 6 μm and 12 μm for the higher and lower Reynolds number respectively.

Measurements of the Surface

When instrumenting these vanes, the metal was machined away and a piece of Pyrex was shaped to fit the contour of the airfoil. This piece was epoxied in place. The epoxy filled under the Pyrex and in-between the Pyrex and the airfoil. Afterwards the metal was polished to smooth out the interface with the epoxy. Extensive surface metrology measurements were taken of the interface between the Pyrex insert and the metal airfoil. These were acquired with the Cyber Optics Cyberscan Cobra laser metrology system. The head has a 17 mm focal length with a 0.5 mm depth measuring range with a resolution of 0.125 um. Figure 11a shows a typical trace across the Pyrex insert from one side of the metal to the other. As is shown, a significant trench occurred within the epoxy layer. It is presumed that the epoxy layer sunk, probably during curing. This was probably due to the width of the channel that was machined out for the insert. This channel was about 4.25 mm with the insert width being 2.75 mm. This extra 1.5 mm is about two times wider on the current vane than on other similar installations. Therefore, more epoxy was needed to fill the gap which shrunk upon hardening. Also apparent in the figure is that the actual gauge is lower than the metal surface. This is believed to be due to the same shrinking phenomena. As the epoxy cured, the Pyrex submerged into the cavity.

Figure 12a shows the measurable portion of the suction surface of the vane. This figure has a reference point of zero at the trailing edge of Pyrex insert. The last three suction side gauges are also shown for reference. The surface contours depict the depth of the surface below the metal (set equal to zero). This figure highlights the depression of the Pyrex into the metal and furthermore accents the trench that occurred all along the epoxy interface both on the sides as well as at the trailing edge. While no measurements were taken at the leading edge of the Pyrex strip it is assumed that a similar trench would have occurred there as well.

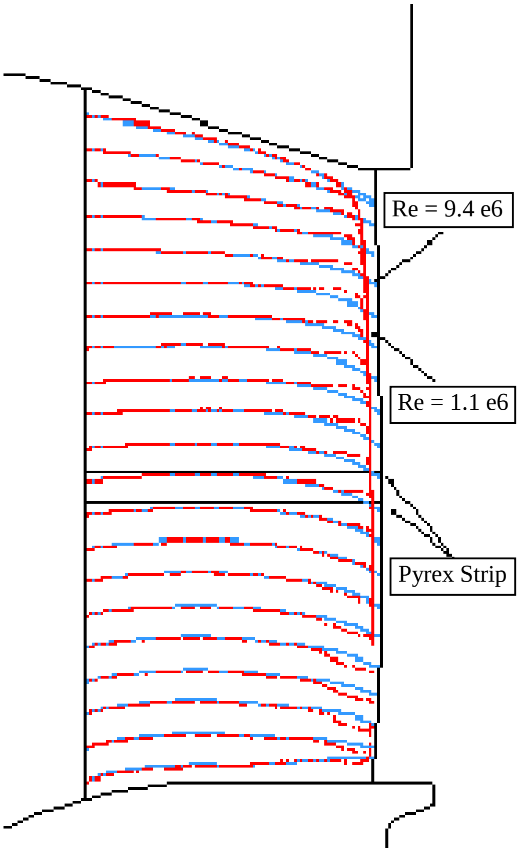

Figure 10 Streaklines over Pyrex Strip

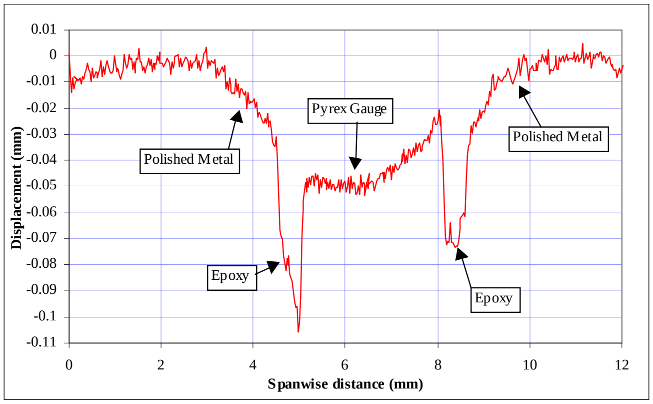

Figure 11a Typical Surface Trace Across Pyrex Insert

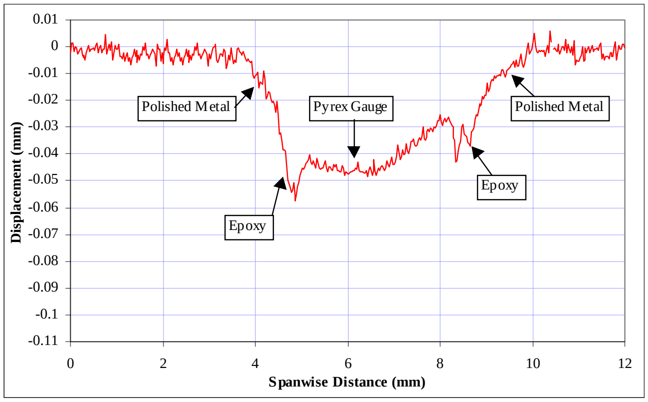

Figure 11b Pyrex Surface after Filling in Epoxy Gap

What is important is that a surface discontinuity occurred at the interface. The polishing of the metal surface resulted in a tapering of the surface down to reach the Pyrex insert. At the interface, the epoxy layer was significantly lower than the metal surface. This depression was anywhere from 0.05 to 0.12 mm. After the epoxy layer, the Pyrex gauge itself was typically below the metal surface. The Pyrex ranged from 0.02 to 0.05 mm below the metal surface on the opposite side of the epoxy. Again that portion of the metal was already 0.02 mm or so below the rest of the metal along the span due to the polishing. On the far side of the gauge, the same phenomena occurred with the trenched epoxy rising back up to the polished metal. Comparing these depressions with the nominal 0.05 mm boundary layer thickness for the higher Reynolds number shows significant penetration of the boundary layer. This could result in a localized tripping of the boundary layer to turbulent values at the interface as described by Liess (1975). In fact the epoxy gap is actually deeper than the boundary layer thickness. This coupled with the streakline pattern could actually cause the boundary layer to separate at the epoxy interface. As the Reynolds number is decreased, the boundary layer thickness increased, which would result in less of an impact on the boundary layer and any type of separation effect. Therefore the data for the two types of gauges could be expected to collapse.

While this epoxy interface certainly does not constitute a uniform surface roughness it does represent an obstacle to the main flow field. This interface can not be directly correlated to a standard trip wire either as the actual magnitude of the trip is questionable. For instance should the trip be quantified based on the total gap (about 0.1 mm) or based on the distance below the Pyrex surface (about 0.05 mm)? Also what is the contribution of the 0.05 mm depression of the entire Pyrex strip? Regardless of the appropriate value for k, the critical k values can be calculated. From equation 1 above kcrit equals approximately 0.040 mm for the majority of the suction surface of the airfoil. Again this represents the roughness height that will cause transition. The fully effective trip height calculated from equation 2 is about 0.048 mm assuming a low M# flow. Over most of the suction surface the Mach number is close to 1.0 which would result in a slightly higher value of 0.066 mm. These values are certainly of the same order as the surface discontinuities measured at the Pyrex to metal interface. Lastly, comparing these k values with the displacement height given above yields ratios well beyond the $k/\delta^* = 0.3$ which results in the trip occuring right at the interface. While it may not be possible to predict the exact nature of the flow over the epoxy interface, it is reasonable to expect that the flow has been altered from its laminar state and would exhibit more of a turbulent nature. This would correlate directly to higher heat transfer rate more closely resembling fully turbulent behavior.

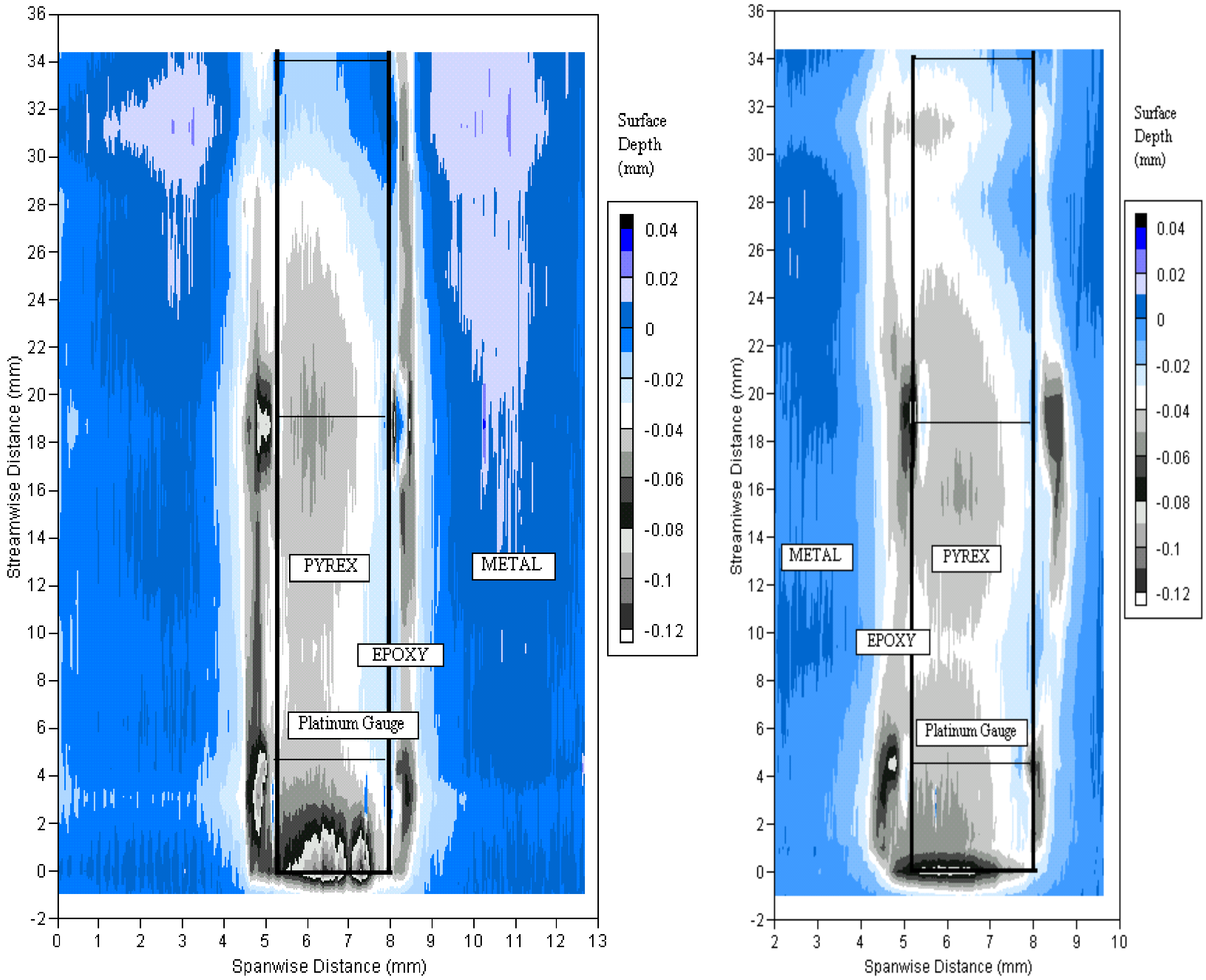

Figure 12 Suction Side Surface Contours a) Initially and b) After Filling Epoxy Gap

Repair of Installation

With this knowledge and the Cyber Optics measurements of the airfoil surface, an attempt was made to smooth out the epoxy interface. Thin layers of paint (DH 1606 High Heat Ceramic Based Paint from Dupli-Color Products) were applied on the sides of the epoxy trough to allow the paint to flow into the deepest portion of the trough. Measurements were taken after a couple of paint layers were applied. These measurements showed that the paint created a smooth transition between the metal and the Pyrex in some regions and that more paint was needed in others. After about four paint layers on both the pressure side and suction side, full surface measurements were acquired. As shown in Figure 11b and 12b, the paint filled the gap between the metal and the Pyrex. Also prevalent in these figures is that no work was performed on the Pyrex gauge itself. It is still submerged below the metal surface. A further note is that it was difficult to measure the region near the front of the suction side Pyrex strip. As shown in Figure 12, measurements were only taken from the second suction side gauge on back. No measurements were acquired at the leading edge portion of this strip. It was also difficult to apply the paint into this region due to the confined space. Because of this minimal paint was added at the front of this strip. Therefore it is uncertain how smooth this portion of the airfoil was upon completion of the repair.

Retesting of 'Repaired' Vane

The vane ring was subsequently reinstalled in the facility and a similar test sequence was performed. The Reynolds number was again varied over the same range with the goal of matching the same values as initially run. For these runs the inlet temperature was again about 403 K and the pressure ratio for all the runs was 1.59. Figure 13 shows the Nusselt number variation for the Pyrex and thin film heat flux gauges for the 'repaired' vane. Again the PS values compare quite well for both gauge types. The SS gauges show the same types of trends as initially with low laminar values at low Reynolds numbers, increasing to fully turbulent at the higher Reynolds numbers.

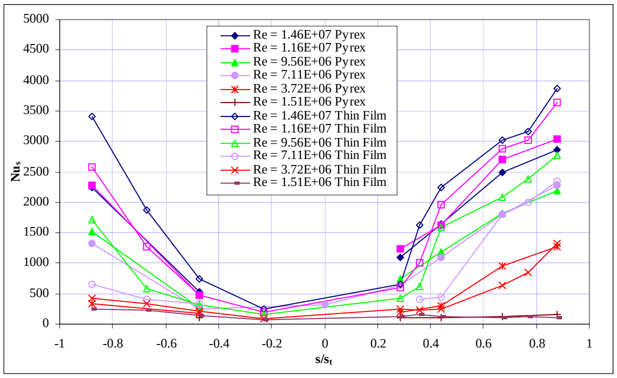

Figure 13 Repaired Nusselt Number Distribution

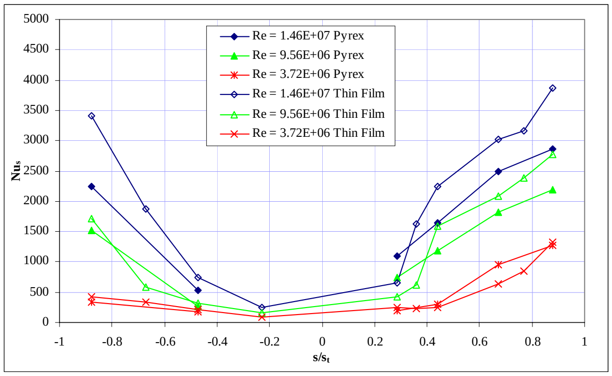

Figure 14 Decrease in Transition in Pyrex Gauges

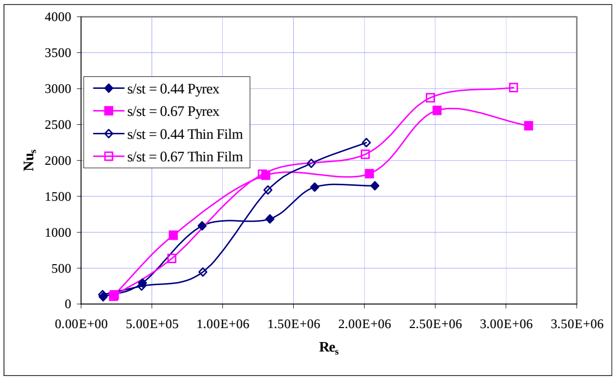

Figure 15 Transitional Behavior of s/s_t = 0.44; 0.67

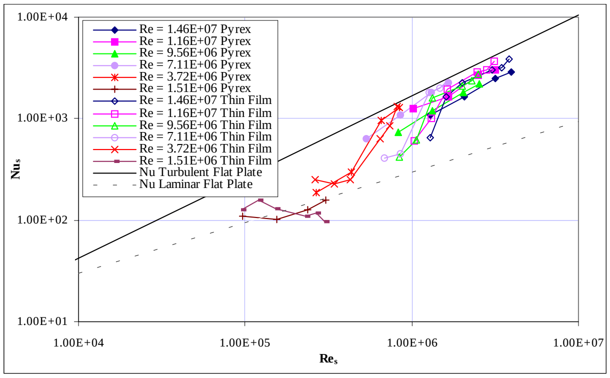

Figure 16 Repaired Local Nu vs. Local Re

Looking just at the same three Reynolds numbers as in Figure 8 shows an overall improvement in the comparison of the two types of gauge technologies (Figure 14). In fact the last two Pyrex gauges follow their thin film counterpart significantly better than the original runs. However the first two Pyrex gauges still show signs of early transition. This is shown more clearly in Figure 15 where only the second and third Pyrex gauges are shown with their thin film counterparts. This figure compares the local Nusselt number with the local Reynolds number for only these gauges. As shown, there is still a discrepancy between the Pyrex and thin film gauges at the more upstream location while the downstream location shows a much better comparison. This is also seen in comparing the local Reynolds number to the Nusselt number (Figure 16) for each test run. This suggests that there still is a problem locally on the airfoil surface. This could be attributable to not filling enough along the side of the Pyrex strip in the upstream region or an influence from the beginning surface of the strip. As mentioned previously this was a difficult region to gain access from both a surface metrology and a filling standpoint. Therefore, not much surface adjustment was made in this region. This could still be the reason for the local discrepancies.

CONCLUSIONS

Two types of heat flux gauges were tested in the Turbine Research Facility. These consisted of a Pyrex insert gauge and a wrapped on thin film gauge. Initial results showed higher Nusselt numbers on the suction surface for the Pyrex gauge for high inlet Reynolds numbers. Both types of gauges collapsed to laminar values as the Reynolds number decreased. This suggested that the Pyrex gauges were tripping to turbulent artificially. Surface metrology measurements confirmed both a dip in the surface contour at the Pyrex insert location and a trough at the interface between the airfoil metal and the Pyrex insert. These deviations were nominally ½ the boundary layer thickness suggesting the possibility that local tripping of the flow could occur. An attempt to fill the trough with a paint was successful at smoothing the interface at locations accessible to be readily painted. However, the region at the leading edge of the Pyrex insert was not accessible and therefore minimal improvements were made. Subsequent testing revealed better correlation between the two types of gauges for the downstream locations, but minimal improvement for the upstream gauges on the suction side. This suggests that either the trough was still deep enough to create a local disturbance or the overall dip in the installation was enough to influence the flow over the Pyrex insert gauges. These measurements reveal the delicacy involved in trying to instrument full scale turbine hardware. With the thin boundary layers present in these flows, very small discontinuities at the instrumentation interface can alter the flow that is being measured. This can drastically change the resultant measurement and provide misleading results to the turbine designer.

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Paper Number: 12

Name of Discusser: K.S. Chana, DERA United Kingdom

Question:
Is there a surface roughness difference between the Pyrex and the thin film Upilex gauges?

Answer:
The measurements show in Figure II are in indication of the roughness of the metal and the pyrex gauge – both about 0,005 to 0,01 mm. These values are obviously less than the overal step as well as critical roughness of 0,04 mm.
Similar measurements were taken for the thin film gauge with lower indicated roughness, but not substantial.

Name of Discusser: H. Weyer, DLR Cologne

Question:
Could you exclude inlet distortion – gay circumferential inhomogeneities in free-stream turbulence – as a reason for the discrepancies in Nu-numbers?

Answer:
We measure the inlet profile (temperature – pressure) with five radial kiel heads on 4 total rakes.
That are circumferentially traversed 90° during a run. While there is some wake shedding from struts upstream of the vane, this is confined to 4 10° sectors. The instrumented vanes are located between the struts were no influence is measured. In this region the turbulence levels have been measured at less than 1 % free stream turbulence.

Name of Discusser: T. Arts, Von Karman Institute, Rhode Saint Genese Belgium

Question:
I s there any possibility of tripping effect from the upilex layer on the flow field?

Answer:
The upilex layer was wrapped completely around the surface from near the trailing edge of the PS to near the trailing edge of the SS.
It also covered the entire span of the airfoil. Therefore the only place the 50 mm step occurred was right at the trailing edge.