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Phase and Time-Resolved Measurements of Unsteady Heat Transfer and Pressure in a Full-Stage Rotating Turbine[1]

M. G. DUNN
Calspan-UB Research Center
4455 Genesee Street
Buffalo, New York 14225

ABSTRACT

This paper presents detailed phase-resolved heat-flux data obtained on rotor blades and a comparison of simultaneously obtained time-resolved heat-flux and static pressure data obtained on the stationary shroud of a Garrett TFE 731-2 HP full-stage rotating turbine. A shock tube is used to generate a short-duration source of heated and pressurized air and platinum thin-film gages are used to obtain heat-flux measurements. Blade results are presented at several selected blade locations. Shroud surface pressure and heat-flux time histories are presented for comparable locations relative to the blade position. For these measurements, the turbine was operating at the design flow function, the design stage pressure ratio, and at 100% corrected speed.

INTRODUCTION

It is well known that turbomachinery flow fields are inherently unsteady because of the disturbances generated when rotating blades transit nozzle vane wakes and exit passages. Years of experience have illustrated that satisfactory flow field and heat-transfer predictions can be performed by assuming that the flow is quasi-steady. However, the state-of-the-art has progressed to the point where relevant unsteady-flow calculations can now be performed and supporting measurements of the unsteady flowfield become a valuable input to the development of computational codes. Significant current research is directed at determining the influence of flowfield unsteadiness on the blade heat-flux and surface-pressure distributions, on the inner blade-row gasdynamic parameters, on the state of the blade and vane surface boundary layers and on the stage efficiency. The results reported in this paper will emphasize phase-resolved[2] heat-flux data on the rotating blade. However, typical time-resolved heat-flux and static pressure data simultaneously obtained on the stationary shroud will also be presented.

Papers relevant to unsteady flow fields in turbomachinery have been present in the literature for at least the past 35 years. Early work by Kemp and Sears (1953, 1955) provided the ground work for much of the research that was to follow. Subsequently, Giesing (1968), Parker (1969), and Kerrebrock and Mikolajczak (1970) made contributions to the understanding of these problems. More recently, Dring, et al. (1980, 1981, 1982) have used a large scale rotating axial turbine stage to obtain experimental data on the nature of the unsteady flow field. Also, Hodson (1983, 1984a, 1984b) has used several different facilities to study wake-generated unsteadiness in vane exit passages and to perform measurements of boundary-layer transition and flow separation. Detailed measurements of the unsteadiness in the rotor incoming flow are presented in Hodson (1984b) which illustrate the change in incidence angle and the change in turbulence associated with the vane wakes.

Another extensive research program concerned with unsteady flow fields was reported by Lakshminarayana et

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2 The term "phase" is used here to denote the pitchwise angular displacement of a given rotor blade with respect to the stator vanes. It varies across each stator passage, in a sawtooth fashion, between the limits zero and $360^o/B$, where B is the number of stators. The term "phase resolved" denotes time-resolved data which are presented as a function of phase, rather than as a function of time. The term "phase averaged" denotes an ensemble average in which phase-resolved data at a given phase, over a succession of stator passages, are taken to be different realizations of the same event. This averaging is referred to by Adamczyk (1985) as passage to passage averaging.

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al. (1982). They reported the results of several programs designed to measure the boundary-layer and turbulence characteristics inside turbomachinery rotor passages using a large-scale, slowly rotating rig as the test device. Binder et al. (1984, 1986) reported the results of laser velocimeter measurements in the unsteady rotor flow field. These authors demonstrate very high turbulence levels associated with the vane wakes. Sharma et al. (1985) presented the results of an extensive study conducted to obtain low-speed rig data on the unsteady flow environment associated with axial flow turbines. Doorly and Oldfield (1985) used a piston-driven tunnel and a system of rotating bars to simulate the effects of shock waves and wakes shed from a nozzle on the blade. The Schlieren photographs presented by Doorly and Oldfield (1985) are helpful in interpreting the results presented herein because they illustrate the rather extensive nature of the vane wake, and the manner in which these wakes interact with the blade. More recently, Wittig et al. (1988) have reported the results of a study designed to investigate the influence of upstream generated turbulence, superimposed on the wake flow, on the blade heat-flux distribution. They used the Doorly and Oldfield rotating bar technique to generate the wake behavior, a grid to produce the turbulence, and LDV and heat transfer measurements to characterize the turbulence.

Dunn et al. (1988) presented detailed phase-resolved heat-flux data obtained on the blade of the Teledyne 702 HP full-stage rotating turbine. Results were presented along the midspan at many locations on the blade suction and pressure surfaces from the stagnation point to near the trailing edge. Guenette et al. (1988) presented time-resolved heat-flux data for the rotor of a Rolls-Royce turbine.

The purpose of this paper is to present a detailed set of phase-resolved heat-flux data obtained on the blade of the Garrett TFE 731-2 HP full-stage rotating turbine. An earlier report by Dunn (1984a) presented the time resolved heat-flux histories for several locations on the vane, the vane endwall, and the blade of this turbine. These data, however, were not phase resolved. Two more recent papers Dunn et al. (1986a), and George et al. (1987), provide a description of the analysis techniques that were previously developed at the Calspan-UB Research Center to obtain instantaneous heat-flux values from the thin-film gages at a sampling frequency consistent with the requirements of this experiment. Several changes have been made to the electronics of the data recording system since the experimental data reported in Dunn et al. (1986a) were taken. These changes were incorporated into the data collection system used for this work and they were previously described in Dunn et al. (1988). Previous papers Dunn et al. (1986a), Dunn and Hause (1982), Dunn et al. (1984a), and Dunn (1986c) have described time-resolved rotor shroud pressure and time-resolved blade heat-flux measurements for this turbine but these blade data were not phase locked as was noted above and the shroud data time resolution was not as good as that presented herein. Epstein et al, (1985) have also reported time-resolved and time-averaged heat-transfer and pressure measurements taken in the stationary shroud over a rotating turbine. The machine used by Epstein was a Rolls Royce design for which the stage pressure ratio was approximately twice the value of the Garrett TFE 731-2 turbine making quantitative comparison of the two data sets for the stationary shroud heat-transfer distributions difficult. However, the results of the two measurements are qualitatively consistent in that the trend of both is for the heat flux to be maximum near the blade leading edge and then to decrease with axial distance through the stage. This trend in shroud heat flux is anticipated because of the decreasing local enthalpy level as a result of work extraction by the turbine.

EXPERIMENTAL APPARATUS

The experimental apparatus used in this work has been described in depth in previous publications noted above and will not be repeated here. Only those portions of the apparatus important to the clarity of this paper will be included.

The thin-film heat-flux gages used in this work were constructed to have a room temperature resistance of 80 to 100 ohms. Each gage is supplied a constant current of 1 milliampere for the duration of the experiment. Prior to the experiment, the gages are calibrated in a temperature-controlled oil bath over the entire range of anticipated temperatures. The voltage versus time (or phase) histories of the individual gages are transferred to the data recording system using a slip-ring technique. The particular slip ring used here was a Poly Scientific 100-channel gold-on-gold unit that is continuously cooled and lubricated under pressure with a Freon/oil mixture. The slip ring contact noise is generally very low, having a maximum value of 25 microvolts compared to a thin-film gage output which is in the tens of millivolts range. Any one of the 100 rings with a noise level in excess of 25 microvolts was eliminated from the system.

The gage output signals were recorded on a data laboratories 2000 series recorder using a technique described in Dunn, et al. (1986a). The 2000 series recorder is a 10-bit unit with eight recording channels, a storage capability of 4K words/channel, and each channel can be sampled at a frequency in the range of 200 KHz to 2 MHz. A shaft encoder that will be described later is used to trigger the recorder. In Dunn, et al. (1986a), it was noted that some of the features of the data reported in that work could not be readily explained by the analysis and that electronic noise combined with frequency splitting due to sampling the data on the slightly accelerating turbine at constant time intervals made it difficult to easily recognize the character of the passage heat flux. Therefore, prior to obtaining the results reported here, the electronic sytem and the amplifiers used in the thin-film gage temperature recording circuits were re-designed and re-built. The amplifiers were constructed to provide capability of wide band (200 KHz), low noise (less than 3 bits out of 1024 bits), single gain data recording and were powered by D.C. voltage (batteries). Great care was taken in providing electrical shielding and adequate grounding for all of the equipment. Where possible, batteries were used as the power supply for the electronic equipment. The noise level on each individual heat-flux gage channel was measured through the entire recording system (including the slip ring) with the turbine rotating. For all channels, the pre-run system noise measured at the recording device just prior to recording the data (with the turbine at full speed) was less than 5 bits out of 1024 bits. Throughout the measurement program, an attempt was made to use as large a portion of the 1024 bits as possible for data recording.

The thin-film gage instrumentation used to perform the heat-flux measurements consisted of two contoured leading-edge inserts and flush-mounted button-type gages. The shroud pressure gages were piezoelectric-type gages. Figure 1 is a photograph of the two leading-edge inserts. Both inserts have four gages with the distribution such that each has a stagnation point gage and three additional gages spaced on the order of 1-mm (0.040-in) apart on either the suction or the pressure surface. Figure 2 is a photograph of the suction surface button-type gages. The pressure surface was instrumented in a similar manner and a photograph is shown in Dunn (1986b). Phase-resolved heat-flux data obtained on both the pressure and suction surface will be presented here. The heat-flux gage signals could be time-resolved or phase-resolved, but we have elected to phase resolve because of an interest in looking at the passage variation of heat flux and because of spatial location difficulties arising because of the slightly accelerating turbine noted in the previous paragraph. The technique for phase resolving will be described in the next paragraph.

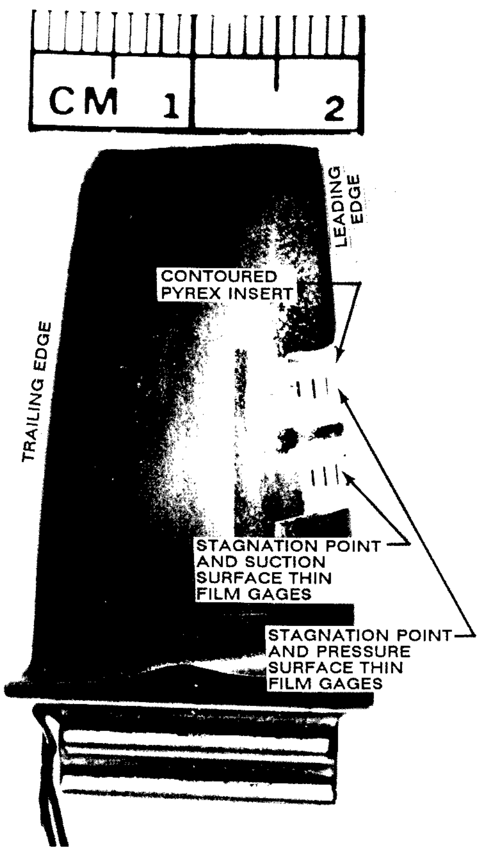

Figure 1 Photograph of contoured leading-edge inserts for rotor blade

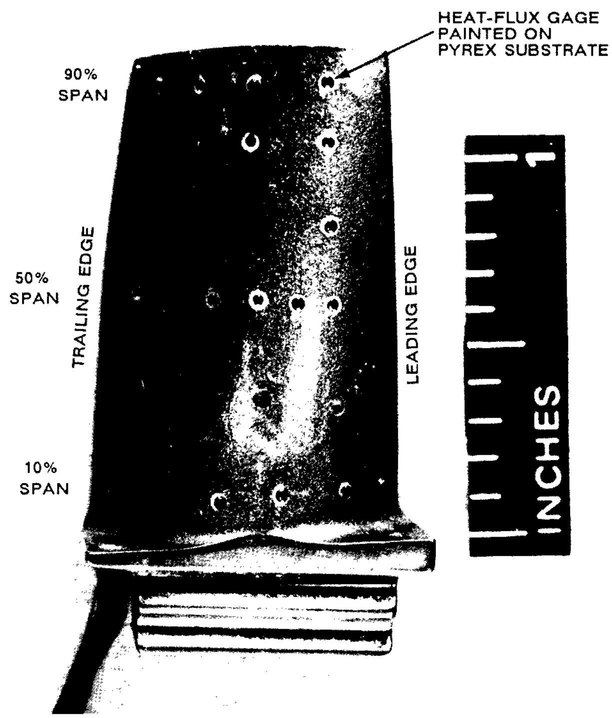

Figure 2 Photograph of blade suction surface button-type heat-flux gages

The turbine used here had 41 vanes and 78 blades. At 100% corrected speed, a blade traverses a vane exit passage in about 75μ sec. The sampling on the heat-flux gages was controlled by a shaft encoder installed on the rotor assembly which provided 720 pulses per revolution along with an index of one pulse per revolution. At the time of model assembly, the leading edge of the insert was carefully aligned with the trailing edge of a known vane. The shaft encoder was then adjusted so that the one-pulse-per-revolution index pulse always occurred at this alignment point. The vane passages are 8.78^{o} or 17.56 pulses apart. The output signal from the shaft encoder was used as an input to the recording equipment in order to sample the blade data at the same angular location within the passage from one revolution to the next, thus phase-resolving the data. This technique resulted in a sampling frequency on the order of 240 KHz, a value well above twice the highest frequency present in the data as required by the Nyquist criterion. The technique used to obtain these data and the data processing procedure are explained in detail in Dunn et al. (1986a and 1988).

EXPERIMENTAL CONDITIONS

A table giving the experimental conditions at which the measurements reported here were performed is given in Table 1. For this turbine, the turbulence intensity just upstream of the vane row was measured (Dunn et al. (1984b) Rivir, et al. (1985)) to be about 5%. The scale of the turbulence was not measured at that time, but an effort to measure the scale is on-going.

Table 1
TEST CONDITIONS AND PARAMETERS

Reflected-shock pressure	7.43×10^3kPa	(1078 psia)
Reflected-shock temperature	559.4°K	(1007°R)
NGV inlet total temperature	559.4°K	(1007°R)
Static pressure at NGV inlet	6.79×10^2kPa	(98.5 psia)
Static pressure downstream of rotor	3.56×10^2kPa	(51.7 psia)
Area on which Stanton number is based	$3.17 \times 10^{-2}m^2$	(0.34 ft^2)
Weight flow without coolant	9.31 kg/sec	(20.5 lb/sec)
Weight flow with coolant	9.45 kg/sec	(20.8 lb/sec)
Corrected rotor speed	100%	-
Turbulence intensity upstream of NGV	5.5%	-
Reflected-shock enthalpy	5.6×10^5 J/kg	(240.98 BTU/lb)
Wall enthalpy at 530°R	2.96×10^5 J/kg	(127.30 BTU/lb)
T_w/T_o	0.53	

DISCUSSION OF RESULTS

A sketch of the physical arrangement of the vane, blade, and the orientation of the blade at 0° and 8.78° phase (360°/41) is given in Figure 3. An extension of the vane mean camber line illustrates the approximate location at which the vane wake would intersect the plane of the blade row leading edge which is about 3.86°. The physical location of the vane wake with respect to the blade phase angle in the operating stage is difficult to predict for the unsteady flow environment.

Data were obtained from many more heat-flux gages on the instrumented blades and the stationary shroud than can be discussed here. A few sample results will be presented to illustrate the general characteristic of the data. The heat-transfer results are given in the form

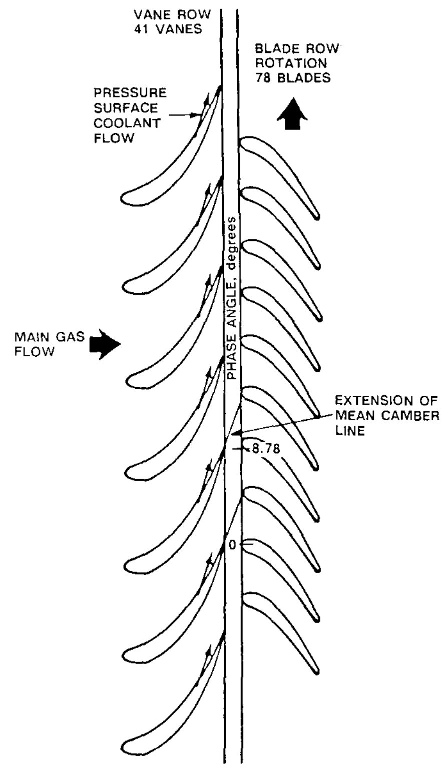

Figure 3 Sketch of stage and phase angle reference

of heat-flux (BTU/ft^2s) history as a function of phase angle instead of the previous format used to present the time averaged results which was a nondimensional Stanton number.

Figure 4 presents the passage phase-resolved heat-flux history for the geometric stagnation point. These data were obtained for the rotor at design corrected speed and without cooling air injection from the vane. Data from three separate passages during one revolution have

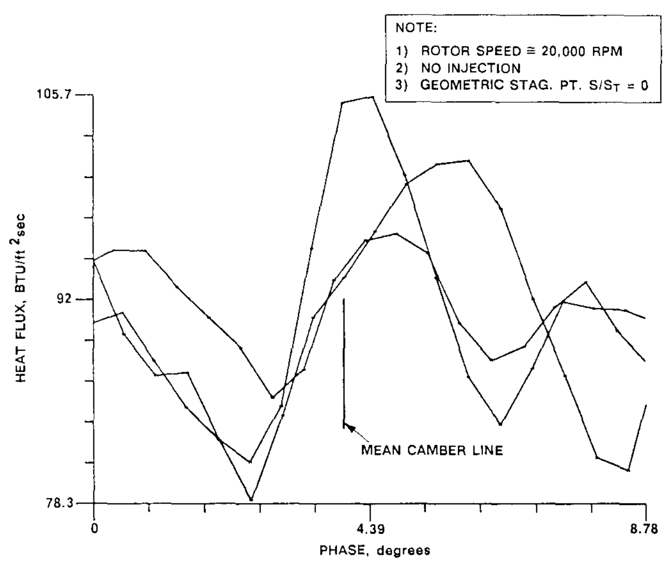

Figure 4 Phase resolved heat-flux data at geometric stagnation point on blade for three different vane passages

been reproduced on Figure 4. The individual dots are the heat-flux values calculated from the temperature data points taken from the digitally sampled data record at the particular phase angle sampled. Straight lines are drawn between successive data points. The heat-flux signal is sampled every 0.5 degrees (or about every 4 microseconds) and thus turbulent fluctuations, as one might expect to see them, are not obvious but are buried in the data record. The location of the mean camber line is also shown on Figure 4. Note that the scale of the heat-flux value shown on the ordinate has been expanded to illustrate the unsteady nature and does not begin at zero but rather at about 78. The heat-flux value is lowest in the early portion of the passage which is felt to correspond to flow outside the vane wake. As the phase angle corresponding to the location of the mean camber line intersection is approached, the heat flux increases rapidly reaching a peak in the vicinity of 4.4^{o} and then begins to fall off. The stagnation-region flow is unsteady as the blade moves from passage to passage, but the general character of the heat-flux history is repeatable. The unsteady nature of the stagnation-region heat-flux data can be attributed to several factors. Among them are; (1) changes in the flow incidence angle as the blade moves through the vane passage exit flow and then through the vane wake flow, (2) differences in the relative turbulence intensity of the passage and wake flows, and (3) differences resulting from manufacturing tolerances associated with the turbine stage. A spectral analysis of the heat-flux data was obtained using Fast Fourier Transform techniques (see typical results given in Dunn et al. (1986a) for this turbine at the present operating condition) and it was demonstrated that the dominant frequency in the data corresponded to the 41 vane passages (or wakes). It is difficult (but possible) to distinguish between passages and wakes with the Fourier analysis.

Figures 5 and 6 are phase-resolved heat-flux histories for blade locations of 9.17% and 85.5% wetted distance on the blade suction surface which can be directly compared with the stagnation-point data presented in Figure 4. The general characteristics of these downstream data are similar to those of the stagnation-region data in that the heat-flux level starts out relatively low at small phase angles and then increases rapidly to a peak at phase angles in the vicinity of the intersection of the extended mean camber line with the leading-edge plane of the rotor. In the case of the data presented by Dunn et al. (1988) for the Teledyne turbine, it was shown that the peak heat-flux values were in the vicinity of the predicted turbulent boundary-layer level and that the minimum heat-flux values were in the vicinity of the laminar boundary-layer levels. Such was not the case for the Garrett turbine results discussed herein. In general, the quasi-steady turbulent boundary-layer predictions were higher than the peak heat-flux and the flat-plate laminar boundary-layer predictions were lower than the minimum heat-flux level. The time-averaged data for this turbine presented by Dunn (1986b) and by Taulbee et al. (1988) are consistent with this observation. The phase-resolved results presented in Figures 4-6 were not ensemble averaged but multiple passages can be superimposed to get a better picture of the heat-flux signal with a result comparable to that presented in Figure 12c of Dunn, et al. (1988). An effort was initiated to perform correlations among the signals recorded for several different heat-flux gages on the same surface. This effort is still in progress and results will be reported at a later time.

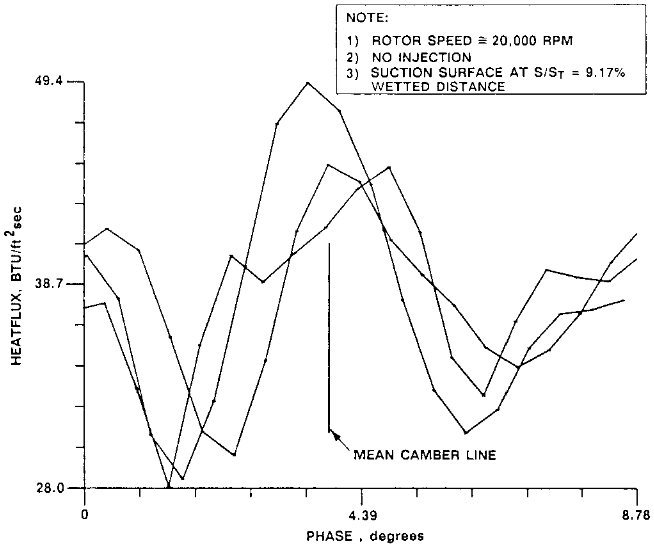

Figure 5 Phase resolved heat-flux data on blade at 9.17% wetted distance on suction surface for three different vane passages

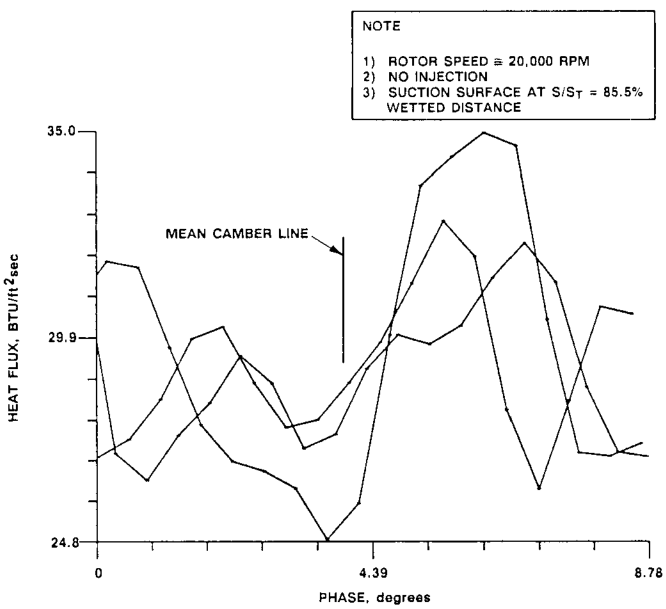

Figure 6 Phase resolved heat-flux data on blade at 85.5% wetted distance on suction surface for three different vane passages

It is of interest to discuss some of the physical factors that may contribute to the differences observed between the phase-resolved heat-flux histories of the two turbines for which detailed data have been obtained. The Teledyne stage has significantly fewer (23) guide vanes than the Garrett stage (41), and a lower solidity, resulting in angular spacing between vane wakes of 15.65^{o} versus 8.78^{o}. Further, the Garrett blade rotates slower but transits a guide vane passage in a smaller time interval (73 microseconds) than the Teledyne blade (97 microseconds). Both turbines used the same shaft encoder which provided an external trigger to the data sampling system so that the heat-flux gages were sampled every one-half degree of rotation. The vane/blade axial spacing of the two turbines was also comparable being on the order of 19% of the vane axial chord. The vane trailing edge thickness of the Garrett turbine is about 73% of the vane trailing edge value of the Teledyne turbine. It should be anticipated that the heat-flux results for the two turbines will be similar, but there is no reason to expect that they will be identical. The rate at which the Garrett blade intercepts guide vane wakes induces a disturbance on the blade boundary layer of higher frequency than was present for the Teledyne turbine. It appears that the blade boundary-layer transition is influenced by the vane wakes, and thus it is argued that the high-frequency disturbance caused by the vane wakes results in the boundary layer being in an unsteady state. However, on the basis of these data the possibility cannot be ruled out that the rapid changes in heat-flux level on the blade observed in transiting the vane wake might be due to differences in turbulence level (and/or turbulence scale) associated with the passage and wake flow regions and that these turbulence parameters would be different for the two turbines considered because of geometrical considerations. Traci and Wilcox (1975) and Lowery and Vachon (1975) have both shown that turbulence intensity can have a significant influence on stagnation-point heat transfer. Blair (1983a, 1983b) reported the results of a study designed to determine the influence of free stream turbulence on the heat transfer to a flat plate in the presence of a fully turbulent boundary layer. Later, O'Brien and Van Fossen (1985) demonstrated the influence of turbulence intensity and turbulence length scale on heat transfer in the stagnation region of a cylinder. Moffat and Maciejewski (1985) demonstrated that flat-plate heat transfer levels can be significantly increased as a result of turbulence. More recently Moffat and Maciejewski (1988) have presented a much more detailed study of the influence of high free stream turbulence on heat transfer in turbulent boundary layers. Subsequent work by O'Brien (1988) suggests that vortex shedding may be a major contributor to the magnitude of the excursions of heat flux experienced by the blade in passage through the vane wake. It should also be noted that the vane exit Mach number for the Garrett turbine was slightly less than unity, while the vane exit Mach number for the Teledyne turbine was equal to or slightly greater than unity.

Figures 7 and 8 are heat-flux histories for locations of 11.4% and 85.7% wetted distance on the blade pressure surface obtained for the same experimental conditions as those of Figures 4-6. The shape of the pressure surface heat-flux history at 11.4% wetted distance is very different from the corresponding result on the suction surface. This is not unexpected since these locations have a very different exposure to the vane exit flow. The three passages shown in Figure 7 illustrate that the peak heat flux occurs at small phase angles and then decreases reaching a minimum just beyond the intersection of the mean camber line. This general characteristic is consistent with that found at a corresponding location on the Teledyne blade (see Figure 10 of Dunn et al. (1988)). Figure 8 illustrates the heat-flux history for three passages at a location of 85.7% wetted distance on the pressure surface. These data indicate a peak heat-flux value at a phase angle just beyond mid passage. This trend is somewhat different from, but not inconsistent with that observed with the Teledyne machine. The contour plots presented in Dunn et al. (1988) (see Figure 11(b) of that paper) illustrate a sharp peak at small phase angle followed by a trend that is completely consistent with that observed here. For Figure 7 and 8, the predicted heat-flux levels

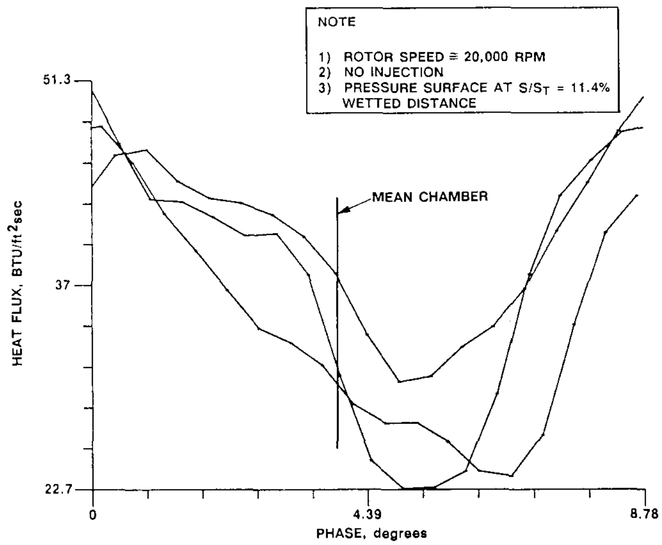

Figure 7 Phase resolved heat-flux data on blade at 11.4% wetted distance on pressure surface for three different vane passages

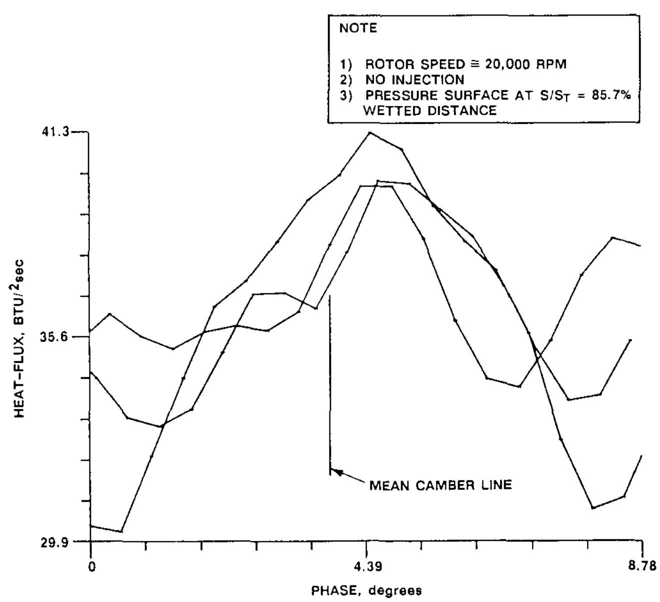

Figure 8 Phase resolved heat-flux data on blade at 85.7% wetted distance on pressure surface for three different vane passages

obtained from the turbulent flat-plate prediction and the laminar flat-plate prediction compare to the data in about the same manner as described in the preceding paragraph for the suction surface.

The Garrett TFE 731-2 HP turbine uses slot injection from the vane pressure surface to cool the trailing edge. This cooling system has been used to investigate the influence of cooling gas injection on both vane and blade time-averaged heat-flux levels (Dunn (1985) and Dunn (1986b)). This previous work indicated that vane injection results in an increase in blade heat-flux level at the geometric stagnation point and at locations on the blade suction surface up to wetted distances of about 15%. Figure 9 is a plot of the phase-resolved heat-flux history at the blade geometric stagnation point in the presence of upstream vane injection. The overall heat-flux level is higher than the corresponding no-injection result shown on Figure 4 as would have been anticipated. The time-averaged results reported in Dunn (1986b) illustrate that at the stagnation point, injection resulted in a 15 to 20% increase in heat flux. The slot injection appears to significantly disturb the flow in the stagnation region and on the early portion of the suction surface of the blade. However, the general characteristics of the heat-flux histories for the two cases are similar. There are several possible reasons for this observed increase in heat-flux level, among them being the potential for changes in the local turbulence intensity and scale as a result of the injection. As noted earlier in this paper, several authors have demonstrated the importance of turbulence intensity and/or turbulence scale on the heat transfer in the stagnation region of a cylinder and to the surface of a flat plate.

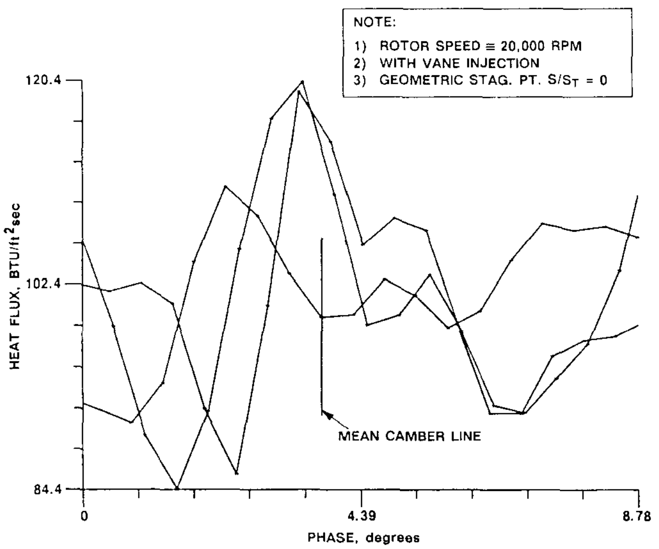

Figure 9 Phase resolved heat-flux data at geometric stagnation point on blade with vane injection for three different vane passages

In looking at the heat-flux data presented in Figures 4-9, an estimate of the uncertainty in both heat-flux value and phase angle is important. A detailed calibration of voltage output versus temperature at constant current is obtained for each heat-flux gage using an NBS-traceable thermometer and an oil bath system. After the gage is placed in the turbine component and the turbine model installed in the tunnel, a calibration signal is passed from the gage terminals through the data recording system. Immediately prior to initiation of the test-gas flow, the room temperature signal from each of the gages is sampled and recorded with the rotor at design speed. These pre-run heat-flux data are a very small percentage of the time-averaged heat-flux signal recorded during the test-gas flow. However, the pre-run values provide an estimate of the system noise level. The data obtained from the experiment consist of a heat-flux phase (or time) history with the fluctuating component due to vane passage cutting superimposed on the time-averaged signal (the heat flux averaged over several revolutions). The uncertainty associated with the time-averaged portion of the heat-flux data is estimated to be on the order of ±1%. As noted above, the heat-flux signals associated with blade transit of the vane passages represents fluctuations superimposed on a much larger signal. The magnitude of the heat-flux fluctuations can be relatively large as illustrated in Dunn, et al. (1988) and in Figures 4-9 of this paper. The magnitude of the gage temperature fluctuation is small compared to the temperature corresponding to the time-averaged heat flux, but the rate of change of temperature with respect to phase angle (or time) is very large. The combination of a small gage temperature change and the large temperature gradient increases the uncertainty in the measurement. It is estimated that the uncertainty in the instantaneous heat-flux value is on the order of ±3%. The uncertainty in the phase angle is mainly associated with turbine manufacturing tolerances and not with the shaft encoder. Once the encoder unit is secured to the shaft and aligned with a particular vane trailing edge, there is little uncertainty in where it will provide repeat signals to trigger the data recording system.

The instrumentation package in the stationary shroud for this turbine includes heat-flux gages and pressure transducers. The time-resolved data taken in the shroud region were obtained as part of a larger program intended to determine the detailed behavior of leakage flow in the tip/shroud region. The blade tip as well as the blade surfaces in the immediate vicinity of the tip (90% span) were all instrumented with heat-flux gages. The time-averaged heat-flux distributions on the respective surfaces are relatively easy to obtain and these form the basic data for the purposes of the leakage flow study mentioned above. However, the time-resolved heat-flux and surface-pressure data are helpful in understanding the behavior of the leakage flow. Figure 10 is a sketch of the shroud instrumentation relative to the blade location drawn approximately to scale. For the purposes of this discussion, attention will be confined

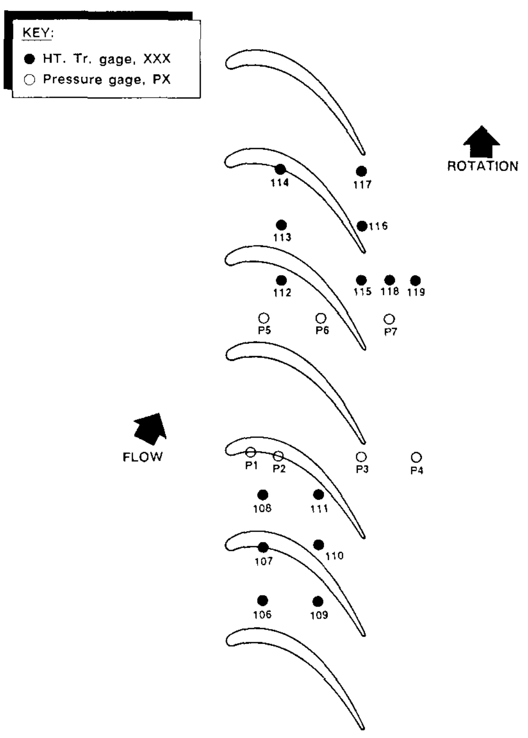

Figure 10 Sketch of shroud heat-flux and pressure instrumentation relative to blade location

to heat-flux gage position 115 and pressure gage position P3 which are both in about the same location relative to the blade pressure surface and of about the same axial chord. The time elapsed between successive blade passage at a fixed shroud location is on the order of 38.5 sec and the sampling rate is on the order of 4.2 sec/sample which results in approximately 9 data samples per blade passage. This sampling rate is sufficient to define the characteristic of the heat-flux and pressure time history as will be illustrated in Figure 11. Here, comparisons are made on the same time base of the heat-flux history at position 115 and the surface pressure history at location SP3. Approximately four blade passages are shown in Figure 11. The heat-flux data give the appearance of a slightly upward trend with time. If a larger portion of the blade revolution were presented, it would be illustrated that the heat-flux level has a slight waviness but is relatively constant. The agreement between the peaks and valleys of the respective time histories is felt to be reasonably good suggesting that the heat-flux and surface pressure are tracking each other in phase. The blade passage time is noted on the figure. Figure 12 is a plot comparable to Figure 11 but for the case of vane slot injection. From previous work with this turbine it is known that the magnitude of the time-averaged heat flux is only slightly influenced by the presence of injection. The with-injection value was found to be on the order of 46 BTU/ft^{2}sec and the corresponding value without injection being on the order of 44 BTU/ft^{2}sec. The standard deviation of the time-averaged data are such that the influence of vane injection on the shroud heat-flux vlaues is not significant.

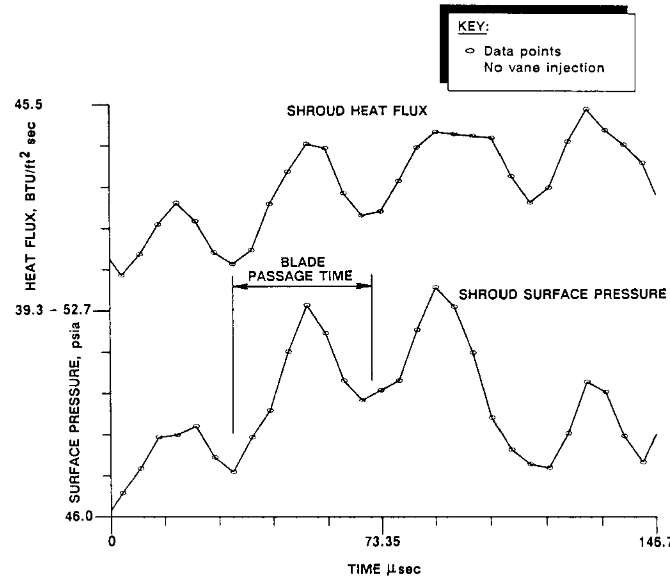

Figure 11 Time-resolved stationary shroud heat-flux and surface-pressure history

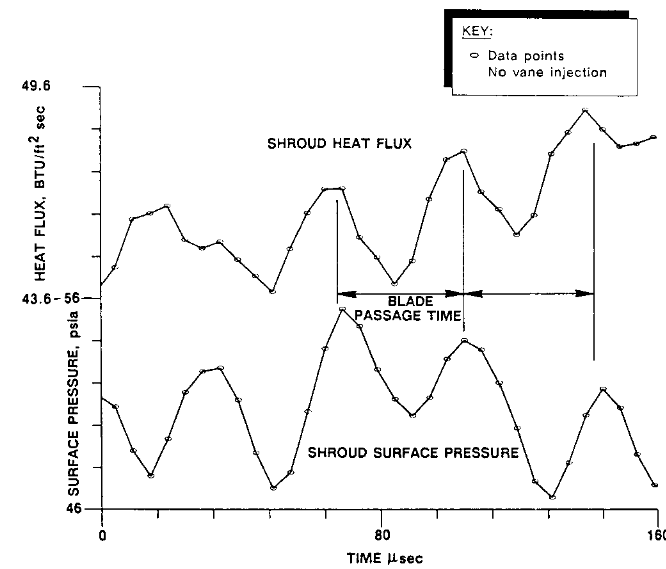

Figure 12 Time-resolved stationary shroud heat-flux and surface-pressure history

CONCLUSIONS

Phase-resolved heat-flux data have been presented for several locations on the Garrett TFE 731-2 HP turbine blade. The peak heat-flux value at the geometric stagnation point and on the blade suction surface appears to be associated with the flow coming from the vane trailing edge wake. Whether this increased heat flux is due to enhanced turbulence in the wake or whether it is due to the influence of the wake flow on the blade boundary layer cannot be determined from these measurements. The presence of vane slot injection is shown to have an insignificant influence on the level of heat flux in the stationary shroud for this particular turbine. However, at the geometric stagnation point of the blade near midspan, the influence of upstream vane injection has a significant influence on the heat-flux history. Time-resolved stationary shroud heat-flux and surface pressure histories are shown to be in phase with each other for the geometric location investigated in this work.

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