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ENTROPY GENERATION MEASUREMENT IN A LAMINAR TURBINE BLADE BOUNDARY-LAYER

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ABSTRACT

This paper demonstrates a method of calculating the entropy generation rate in an incompressible laminar turbine blade boundary-layer from measurements of surface heat transfer rate. It is shown that the entropy generated by fluid friction in an incompressible blade boundary-layer is significantly less than that generated by heat transfer at engine representative temperature ratios. The centre blade in a low-speed linear cascade is electrically heated and isolated from the airflow with a bypass valve. Upon opening the valve the blade is transiently cooled and thin film heat transfer gauges, painted on machinable glass ceramic inserts mounted into the surface of the blade, are used to record blade surface temperature and surface heat transfer rate signals; local Nusselt numbers are then calculated. Non-dimensinnal temperature distributions are derived across the boundary-layer using the blade surface heat transfer rate and a similarity condition. The equation describing the local entropy generation per unit volume is then integrated through the boundary-layer at each chordwise measurement point on the blade surface.

NOMENCLATURE

Ec	Eckert number	–
Fr	Frößling number	–
H_δ	Boundary-layer shape factor	–
M	Mach number	–
Nu	Nusselt number	–
P	Pressure	N/m^2
Pr	Prandtl number	–
R	Gas constant	J/kg K
	Correlation coefficient	–
Re	Reynolds number	–
$\dot{S}'''_{gen}$	Entropy generated per unit volume	$W/m^3 K$
$\dot{S}''_{gen}$	Normal entropy generation rate	$W/m^2 K$
T	Temperature	K
T_r	Temperature ratio	–
c	true chord	m
c_p	specific heat at constant pressure	J/kg K
h	local convection coefficient	$W/m^2 K$
k	thermal conductivity	W/m K
$\dot{m}$	mass flow rate	kg/s
m, n	power law exponeots	–
$\dot{q}$	local heat flux	W/m^2
u, v	x- and y-direction velocities	m/s
x, y	Cartesian coordinates	–
%Tu	percentage turbulence intensity	–
δ	velocity boundary-layer thickness	m
δ_S	entropy sub-layer thickness	m
δ_t	temperature boundary-layer thickness	m
γ	specific heat ratio	–
μ	dyoamic viscosity	kg/m s
ρ	density	kg/m^3

Suffices

c	true chord
ff	fluid friction
ht	heat transfer
o	total condition
s	blade surface
w	wall
∞	inlet freestream condition
*	dimensionless quantity

Dimensionless Parameters

$$Ec = \frac{(\gamma - 1)M^2}{T_r} \qquad M = \frac{u}{\sqrt{\gamma R T}}$$

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$$Nu = \frac{hc}{k} \qquad Fr = \frac{Nu_c}{\sqrt{Re_c}} \qquad Pr = \frac{\mu_\infty c_{p\infty}}{k_\infty}$$

$$Re = \frac{\rho_\infty u_\infty c}{\mu_\infty} \qquad T_r = \frac{T_w - T_\infty}{T_\infty}$$

List of Abbreviations

CFD	Computational Fluid Dynamics
UL	University of Limerick
%SSL	percentage suction surface length

INTRODUCTION

In turbomachinery the word "loss" is used to describe any process which reduces overall efficiency and its predictinn tends to be based on cnrrelations of experimental data. However, cnrrelations are limited in terms of accuracy and in the range of parameters and their values; consequently, their use may yield misleading results. The approach by Dentnn and Cumpsty (1987) and Denton (1993, 1990) towards a physical understanding of the loss phenomena that reduces efficiency can be of more benefit than a loss correlation, as it depicts reginns of high entropy. The approach taken in this study is to calculate the local entropy generatinn rate; this clearly demnnstrates bnth where the loss is generated and the proportinn of lnss attributable to viscous and heat transfer phenomena. These are two clear advantages over the calculation of local entropy. However, both methnds are based upon the principle nf lncal thermodynamic equilibrium and this has never heen justified in turbomachinery flows where large thermal and viscnus gradients imply that conditions are far from equilibrium.

The second law of thermodynamics implies that viscous friction and heat transfer result in the generation of entropy. For an uncooled gas turbine in equilibrium, a blade nr vane will establish itself at the relevant adiabatic wall temperature and nn heat transfer will nccur. Consequently, the entropy generated within the bnundary-layer will be snlely due to viscnus frictinn. For an internally cooled hlade nr vane, however, entropy will be generated throughout the velocity and temperature boundary-layers, accumlatively cnntributing tn the nverall loss in the machine. Denton (1993) assumed an adiabatic machine when considering the thermodynamics of the cycle, thereby resnlving all heat transfer induced losses as an internal cooling lnss. In contrast, the approach adopted herein cnncentrates on the generation of entropy within the velocity and temperature bnundary-layers.

The measured increase in the entropy generatinn rate is a cnnvenient parameter. Firstly, its value is independent of the frame of reference, rotating or stationary. Secondly, the additive prnperty of the thermal and viscous dissipative terms allows a sectinnal analysis of the machine's individual cnmponents contributing to the overall loss and, more importantly, a relative analysis of the heat and friction lnsses which is only feasible in this manner (Bejan, 1979). Finally, the entropy generation rate in a system is a measure of its performance and allows comparisons between systems that fulfil the requirements outlined by Davies and Wallace (1995) fnr similarity.

Fritsch and Giles (1993) compared the entropy rise through the stage of a transonic turbine for bnth a steady and unsteady viscous simulation. However, the entropy generated due to heat transfer in a turbine hlade has yet to he measured. The concept of entropy generation and its relevance to turbomachinery will be addressed, while boundary-layer entropy generation rate measurements in a linear cascade are presented later.

ENTROPY

The mnst commonly used expression for entropy in turbomachinery calculations is that given by :

$$\dot{S}_{gen} = \dot{m}\left[c_p \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right)\right] \tag{1}$$

Alternatively, losses at any arbitrary point in the flow may be defined by the local rate of entropy generation per unit volume, $\dot{S}'''_{gen}$, first derived by Bird *et al.* (1960) fnr a two-dimensional incompressible flow and its use has since been pioneered by Bejan (1979):

$$\dot{S}'''_{gen} = \frac{k}{T^2}\left[\left(\frac{\partial T}{\partial x}\right)^2 + \left(\frac{\partial T}{\partial y}\right)^2\right] + \frac{\mu}{T}\left[2\left[\left(\frac{\partial u}{\partial x}\right)^2 + \left(\frac{\partial v}{\partial y}\right)^2\right] + \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right)^2\right] \tag{2}$$

where T is the absolute temperature at the point in the flnw where $\dot{S}'''_{gen}$ is evaluated. Increasing the absolute temperature of a gas turbine stage will increase both the thermal and component efficiencies since the local entropy generation rate is inversely proportinnal to T for fluid friction and T^2 for heat transfer.

Davies and Wallace (1995) defined the nnn-dimensional local rate of entropy generatinn per unit vnlume as:

$$\dot{S}'''^{*}_{gen} = \frac{\dot{S}'''_{gen}\,\delta^2}{k_\infty} = T_r^2\left[\frac{\delta}{\delta_t}\right]^2\left[\left(\frac{\partial T^*}{\partial y^*}\right)^2 + \begin{matrix}\text{similar}\\ \text{terms}\end{matrix}\right]\left[T_rT^* + 1\right]^{n-2}$$
$$+ \Pr \mathrm{Ec}\, T_r\left[\left(\frac{\partial u^*}{\partial y^*}\right)^2 + \begin{matrix}\text{similar}\\ \text{terms}\end{matrix}\right]\left[T_rT^* + 1\right]^{m-1} \tag{3}$$

and also showed that the dnminant mode of entropy generation is heat transfer for an incompressible flow with equivalent gas turbine temperature ratins. In this study the problems with measuring the entropy generation rate due to heat transfer in an incompressible laminar boundary-layer on a turbine blade are demonstrated. The methodology is to generate thermal boundary-layer profiles from measured surface heat transfer rates.

LINEAR CASCADE HEAT TRANSFER FACILITY

The existing linear cascade at the University of Limerick is a twn-dimensional model of the turbine blade mid-span utilised by the Brite/Euram programme (Santoriello *et al.*, 1993). Transient heat transfer experiments are performed in this cascade by heating an instrumented blade to the desired temperature in order to achieve a predetermined temperature ratio between the fluid and the blade; the direction of heat transfer is immaterial for incompressible flow. Wallace (1996) presents a detailed description of the cascade, while Table 1 below outlines the typical operating parameters.

Table 1 : Typical UL Linear Cascade Operating Parameters

Inlet Reynolds number, Re_c	2.0×10^5
Inlet turbulence intensity, $\%Tu$	0.8%
Total temperature, T_o	293K
Turbine blade temperature, T_s	440K
Prandtl number, Pr	0.694
True chord, c	0.094m

The central blade passage was instrumented with 24 pressure tappings. The velocity and Mach number distributions for an inlet Reynolds number of 2.0×10^5 are shown below in figure 1; this is the maximum inlet Reynolds number for heat transfer experiments obtainable. The flow around the blade is mostly incompressible, reaching a maximum Mach number of approximately 0.25 on the suction surface.

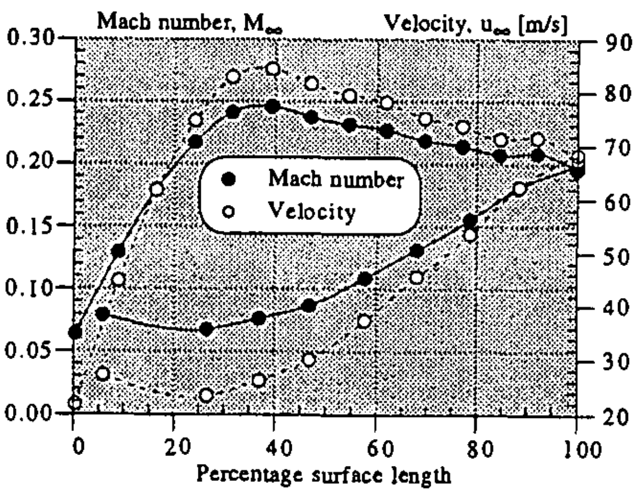

Figure 1 : Mach number and velocity distributions around the UL linear cascade turbine blade: $Re_c = 2.0 \times 10^5$

Thin film heat transfer gauges, painted on MACOR™ machinable glass ceramic inserts designed to ensure one-dimensional heat conduction, monitor the surface temperature of the blade as it cools with time and the corresponding heat flux is determined using electrical analogue techniques; each gauge is 3mm in length and their resistances at 293K varied between 60Ω and 65Ω while temperature coefficient of resistance values were in the range 2.40 - 2.51 x 10^{-3} K^{-1}.

The instrumented blade was internally heated by four cartridge heaters, each separately controlled by a Eurotherm™ 91ε controller. An Agema Thermovision 880 Infra-red scanning system deployed to determine the blade surface temperature at mid-span measured 440K ±2K.

The thin film gauge and heat transfer signals were processed using a WANG 386PC containing a Metrabyte DAS-16/F A/D board and data acquisition software. The overall A/D programmable scan rate was variable up to 50kHz depending on the number of data acquisition channels, with 12 bit resolution and an input voltage range of ±2.5V. Therefore, the maximum voltage error due to digitisation is ±0.610mV. A pitot mounted pressure transducer is used to measure the inlet turbulence intensity level of 0.8% (Wallace and Davies, 1996).

The blade suction surface heat flux measurements will be used to define typical non-dimensional temperature boundary-layer distributions and examine the chordwise variation in the non-dimensional normal entropy generation as a function of the inlet Reynolds number and a boundary-layer shape factor; this investigation will draw on the entropy analysis presented earlier.

HEAT TRANSFER MEASUREMENT

In most transient test facilities heat transfer is measured by monitoring the variations of the surface temperature and relating this to the surface heat transfer rate. Although the experiments performed in the cascade were transient, the results obtained for the distribution of the local heat transfer coefficient on the suction surface of the turbine blade refer to the steady state, because the time required to establish the flow field, including the thermal and velocity boundary-layers, is extremely short compared with the thermal time constant of the blade or the run time of the experiment (Jones *et al.*, 1993). The long run time of the cascade heat transfer experiment allows the blade surface temperature to drop appreciably with time after the onset of the flow when a bypass valve is opened. This creates essentially a quasi-steady state condition as far as the flow is concerned, with each condition corresponding to a different blade surface temperature distribution. The result for an isothermal blade is determined by extrapolating the transient measurement back to zero time when the isothermal blade is at its initial test temperature (Oldfield *et al.*, 1984). The local heat transfer coefficient, h, at any point on the blade surface is determined as:

$$h = \frac{\dot{q}_s}{T_w - T_o} \qquad (4)$$

The local blade surface heat transfer coefficients are presented in figure 2 for five different inlet Reynolds numbers, Re_c. A data sample size of ten was chosen for each steady state heat transfer measurement around the blade for five different inlet Reynolds numbers and averaged, in order to minimise the random errors. Estimates of the random error standard deviatioos were calculated to an uncertainty of 95% confideoce, and are graphically represented by limit bars in figure 2. The local heat transfer coefficients were then converted intn lncal Nusselt numbers, Nu_c, based on the blade chord, c, and the thermal conductivity, k, evaluated at the test film temperature.

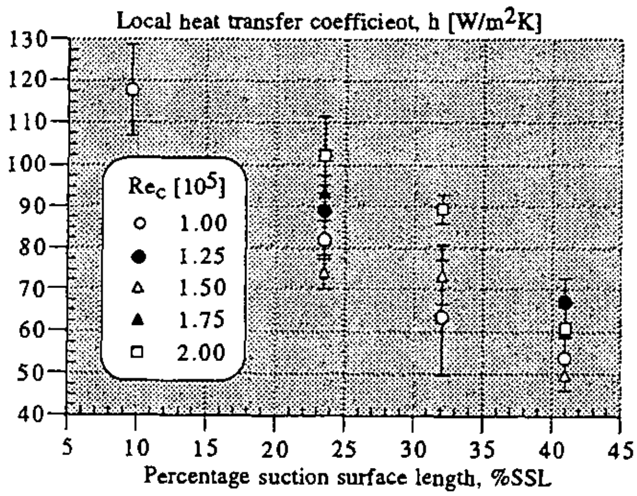

Figure 2 : Blade suction surface heat transfer coefficients

The trend of the results presented in figure 2 is consistent with those measurements recorded at similar inlet Reynolds numbers. Han *et al.* (1993) found suctioo surface Nusselt numbers to decrease monotonically with increasing streamwise distance from the blade stagnation point, and then to increase sharply due to transition in the turbulent flow at 67%SSL. Consigny and Richards (1981) reported a mostly laminar flow over the suction surface of the blade for an inlet Reynolds number of 2.34×10^5. Because these tests were performed at high subsonic and transonic turbine exit Mach numbers, values of heat transfer coefficients were of order three times greater than those reported here.

The maximum error caused by the surface temperature discontinuity of the machinable glass ceramic inserts in the measured values of the heat transfer rate and the heat transfer coefficient are ±2.1% and ±5.7% respectively as determined by Kays and Crawford (1993); these errors correspond to the first thin film gauge at 9.6%SSL and will decrease with increasing distance downstream from the leading edge. Flow visualisation over the blade surface indicated no boundary-layer tripping due to the ceramic inserts.

Frößling Number

Schlichting (1979) determined for laminar flat plate flow that heat transfer data plotted as a non-dimensional group, the Frößling number, is indepeodent of the Reynolds number. Figure 3 is a plot of the Frößling number of the turbine blade heat transfer data for the range of test Reynolds numbers; the limits presented in figure 3 refer to 95% confidence and 0%SSL corresponds to the stagnatioo point. Empirical correlations for forced convection heat transfer around cylinders were used to evaluate the Nusselt number at the stagnation point of the turbine blade. Five empirical correlations were used: Churchill and Bernstein (1977), Eckert and Drake (1972), Fand (1965), Hilpert (1933) and Whitaker (1972). All fluid properties were evaluated at the experimental test film temperature. Additionally, figure 3 shows that the Frößling number is independent of the Reynnlds number fnr incompressible flow, even in the preseoce of a pressure gradient.

ENTROPY MEASUREMENT

Local blade surface heat flux values measured in the laminar boundary-layer yield local values of the temperature gradient at the blade surface. Utilising these thermal gradients and known boundary coodiitions at the boundary-layer edge and blade surface, a family of temperature boundary-layer distributions were constructed. Local entropy generation rates were then determined, the entropy generation rate perpendicular to the surface evaluated and its distribution on the suction side of the blade plotted. The following analysis may be considered to be the first step in obtaining a quantitative value of the total entropy generation rate within a laminar temperature boundary-layer on a turbine blade. It is equivalent to calculating the entropy generated in an internally cooled blade bnundary-layer due to heat transfer. The entropy generation rate is always positive as seen from equation (2). Although the heat to the blade is mostly recovered when the cooling flow is blown out through the tip or trailing edge, the generated entropy is not lost and will appear as a system irreversibility.

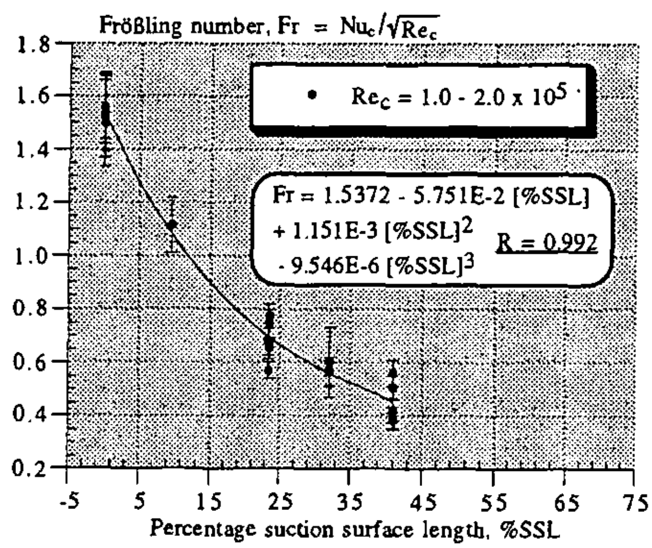

Figure 3 : UL linear cascade heat transfer measurements chordwise variation with inlet Reynolds number, Re_c, plotted as the Frößling number

Temperature Boundary Conditions

The heat flux boundary conditions in addition to well known thermal boundary conditions at the blade surface and at the boundary-layer edge are used to generate typical temperature boundary-layer profiles. The conditions are:

Blade surface	Boundary-layer edge
$y = 0 \Rightarrow y^* = \frac{y}{\delta_t} = 0$	$y = \delta_t \Rightarrow y^* = \frac{y}{\delta_t} = 1$
$T = T_w \Rightarrow T^* = \frac{T - T_\infty}{T_w - T_\infty} = 1$	$T = T_\infty \Rightarrow T^* = \frac{T - T_\infty}{T_w - T_\infty} = 0$
$\frac{\partial^2 T}{\partial y^2} = 0 \Rightarrow \frac{\partial^2 T^*}{\partial y^{*2}} = 0$	$\frac{\partial T}{\partial y} = 0 \Rightarrow \frac{\partial T^*}{\partial y^*} = 0$

A further condition at the blade surface is:

$$h\left[T_w - T_0\right] = -k\left.\frac{\partial T}{\partial y}\right|_{y=0} \tag{5}$$

Non-dimensionalising equation (5) above using the same non-dimensional ratios utilised above yields:

$$\left.\frac{\partial T^*}{\partial y^*}\right|_{y^*=0} = -Nu_c \tag{6}$$

Therefore, the measured Nusselt number is an additional temperature boundary condition that will be used to construct non-dimensional temperature boundary-layer profiles.

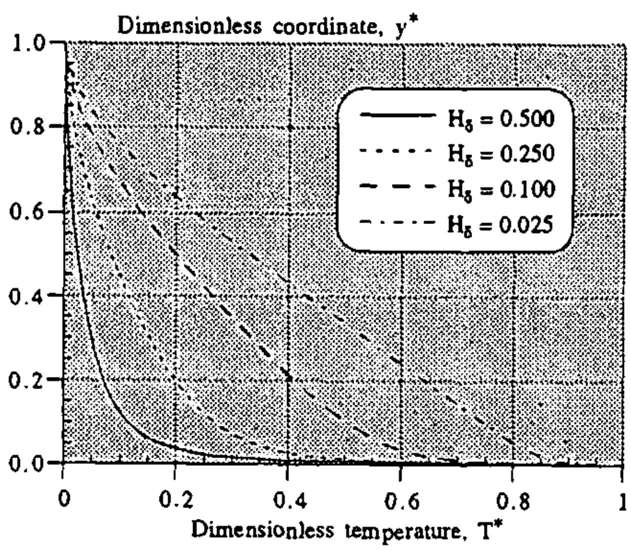

Figure 4 : Non-dimensional incompressible temperature boundary-layer distributions for values of the boundary-layer shape factor, H_δ, with $Nu_c = 400$: equation (7)

Temperature Boundary-layer Distribution

A family of profiles obtained by trial and error that satisfies the boundary conditions outlined above is:

$$T^* = \left[y^* - 1\right]^2 e^{y^*}\left[1 + \left(\frac{Nu_c - 1}{H_\delta}\right)y^*\right]^{-H_\delta} \tag{7}$$

An illustration of this is presented in figure 4 for varying values of the boundary-layer shape factor, H_δ, and a fixed measured Nusselt number.

Different values of the boundary-layer shape factor, H_δ, give rise to different profiles as shown in figure 4. The two central profiles in figure 4 are probable turbine blade temperature distributions while the severe variations in gradient of the two outer profiles, especially at the blade surface and boundary-layer edge, are unlikely. However, it is not known what the distribution of the thermal boundary-layer would be in the chordwise direction around the cascade blade. Therefore, the analysis will continue and account for this boundary-layer feature by varying the value of the boundary-layer shape factor, H_δ, throughout.

The variation of the non-dimensional temperature distribution for a range of local Nusselt numbers representative of those measured from the UL linear cascade is presented in figure 5 for a single value of the boundary-layer shape factor.

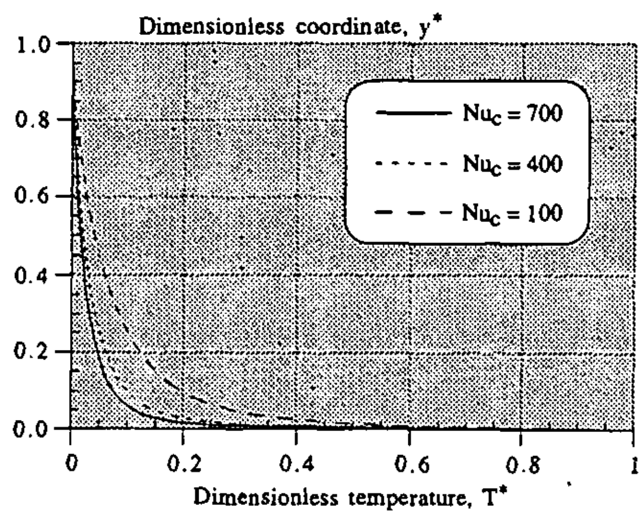

Figure 5 : Non-dimensional temperature boundary-layer distribution variation with Nu_c: $H_\delta = 0.5$

Local Entropy Generation Rate

The knowledge of blade surface heat transfer rates and the value of the boundary-layer shape factor is necessary in order to construct a temperature boundary-layer distribution at the blade surface. Since the boundary-layer shape factor, H_δ, is unknown, the non-dimensional local entropy generation rate, $\dot{S}_{gen}^{'''*}$, will be determined for a range of H_δ values. As heat transfer is the dominant mode of entropy generation, then equation (3) reduces to the expression below for an incompressible flow with constant fluid properties.

$$\dot{S}_{gen}^{'''*}(\text{Heat transfer}) = \frac{\dot{S}_{gen}^{'''}\delta_t^2}{k} = T_r^2\left[\left(\frac{\partial T^*}{\partial y^*}\right)^2\right]\left[T_r T^* + 1\right]^{-2} \tag{8}$$

The entropy generation rate within the boundary-layer is plotted in figure 6 for illustrative values of the shape factor and Nusselt number. It is clear that the majority of the entropy generated in the near wall region. This conclusion is equally true for all measured Nusselt numbers and reasonable shape factors.

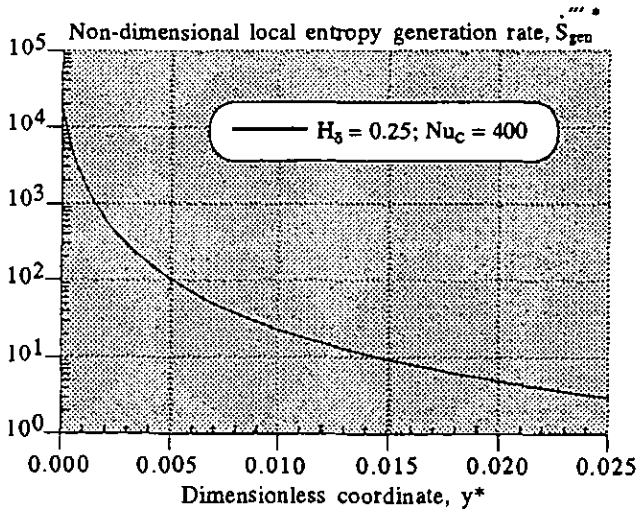

Figure 6 : Non-dimensional local entropy generation rate boundary-layer distribution for $0 \le y^* \le 0.025$

The area under the curve is the non-dimensional normal entropy generation rate perpendicular to the blade surface:

$$\dot{S}''^{*}_{gen} = \int_0^{\delta_t} \frac{\delta_t}{T^2}\left[\frac{\partial T}{\partial y}\right]^2 d\,y = T_r^2 \int_0^1 \left[\left(\frac{\partial T^*}{\partial y^*}\right)^2\right]\left[T_r T^* + 1\right]^{-2} d\,y^* \qquad (9)$$

The non-dimensional normal entropy generation rate variation with the Nusselt number has been determined for a range of boundary-layer shape factor values and is presented in figure 7; this is perfectly general and does not require blade surface heat flux measurements. All the non-dimensional entropy generation rate distributions were numerically integrated using *Maple V* (Char *et al.*, 1991).

It is evident that the variation of the non-dimensional entropy generation rate perpendicular to the surface with Nusselt number is linear, and this is true for $0.025 \le H_\delta \le 0.50$; however, it is difficult to comment on whether this is a universal characteristic and not related to the form of the mathematical function chosen to represent the non-dimensional temperature boundary-layer distribution. The linear regression equatioos of the data in figure 7 each have a linear correlation coefficient, R, equal to one indicating a perfect correlation. Moreover, the different non-dimensional temperature profiles, characterised by a variation in the boundary-layer shape factor, gives rise to different entropy generatinn rates; the fuller the profile, the greater the temperature gradients and the larger the normal entropy generation rate. Therefore, the boundary-layer temperature distribution and its shape factor must be well defined in order to determine accurately the normal entropy generatinn rate.

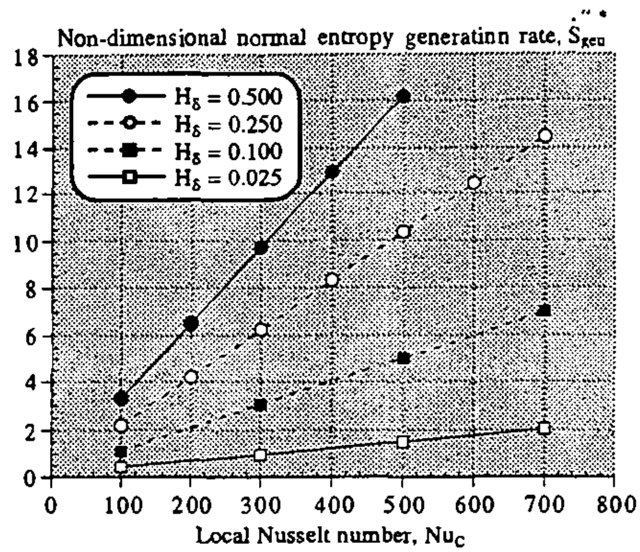

Figure 7 : Non-dimensional normal entropy generation rate variation with the local Nusselt number, Nu_C, and the boundary-layer shape factor, H_δ

Recalling the heat transfer measurements on the suction surface of the UL linear cascade that were presented in figure 3, it is now possible to plot the chordwise variation of the normal entropy generation rate assuming a similarity condition such that the temperature boundary-layer distribution does not vary in the chordwise direction; that is, the boundary-layer shape factor, H_δ is constant over the measurement region of the suction surface.

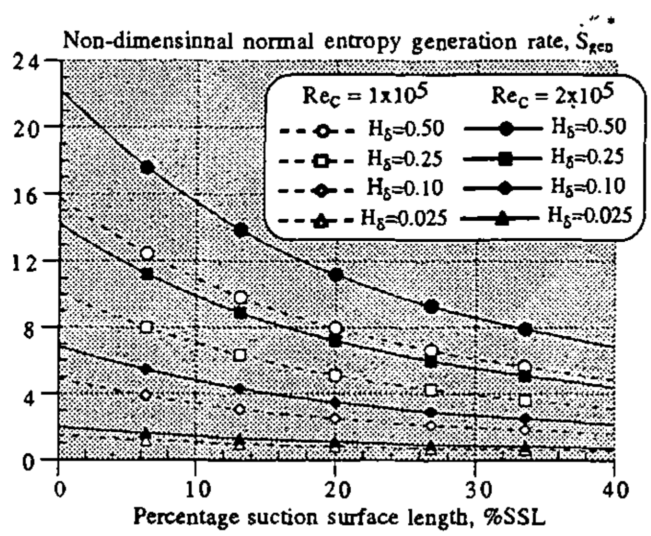

Figure 8 : Chordwise distribution of the non-dimensional entropy generation rate perpendicular to the surface and variation with the inlet Reynolds number, Re_C, and the boundary-layer shape factor, H_δ

This is plotted in figure 8 above for the maximum and minimum experimental inlet Reynolds numbers, Re_c, and varying values of the boundary-layer shape factor. The important aspect for a designer is to design a blade profile that produces the required work for the minimum amount of loss and the required reliability; in order to do this it is necessary to know where the loss has been generated. It is possible that the method described will show this without prior knowledge of the boundary-layer shape factor.

ENTROPY SUB-LAYER

The entropy sub-layer is defined as the near wall region where 99% of the entropy is generated. In calculating turbomachinery blade profile loss and heat transfer irreversibilities it is this region that must be accurately predicted. Taking this a step further, the entropy sub-layer due to heat transfer and fluid friction will be denoted by $\delta_{S(ht)}$ and $\delta_{S(ff)}$ respectively; only in special cases such as this analysis where the dominant mode of entropy generation is heat transfer will $\delta_S \equiv \delta_{S(ht)}$. The heat transfer entropy sub-layer is non-dimensionalised by the temperature boundary-layer thickness, δ_t. Denton and Cumpsty (1987) defined an entropy thickness within the boundary-layer which reduced to the form of the energy deficit thickness assuming the static pressure and stagnation temperature are constant within the boundary-layer.

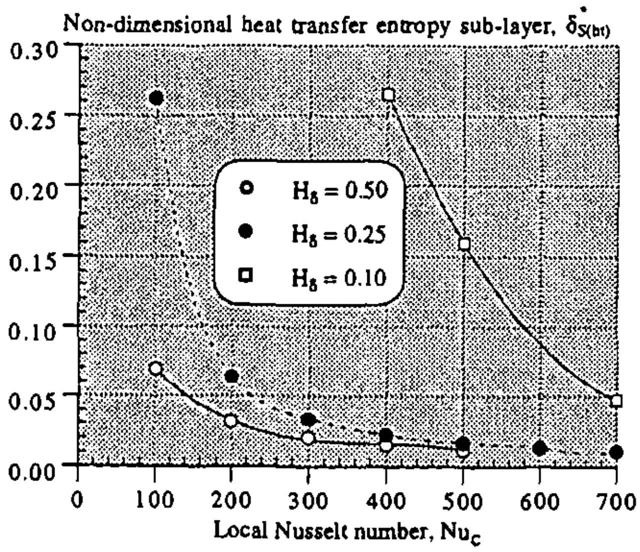

Figure 9 : Non-dimensional entropy sub-layer variation with the local Nusselt number, Nu_c, and the boundary-layer shape factor, H_δ; equation (7)

The entropy sub-layer has been determined for the non-dimensional temperature boundary-layer distribution and shown in figure 9 as a function of the boundary-layer shape factor, H_δ; the entropy sub-layer decreases with an increasing boundary-layer shape factor as expected. The entropy sub-layer also highlights the sources of irreversibility not evident from entropy generation rate distributions. For engine Nusselt numbers, the plot asymptotically approaches a value of approximately 0.01, and the entropy sub-layer will be only 1% of the thermal boundary-layer thickness, δ_t. Therefore, experimental or numerical analysis of the entropy generated around a blade must be concentrated in a region close to the blade surface of the order of tens of microns thick. As shown in figure 4, the temperature profiles are nearly linear in this region which greatly simplifies the problem.

DISCUSSION

A method for determining the non-dimensional spanwise entropy generation rate due to heat transfer in the laminar boundary-layer on the suction surface of an isothermal internally cooled airfoil has been presented as a function of the boundary-layer shape factor and the Reynolds number; constant fluid properties were assumed and the method could easily be extended to account for the chordwise variation of these parameters. To evaluate the total entropy generation rate and determine the turbine blade efficiency for comparison with conventional experimental techniques, the boundary-layer thickness must be known. However, the present method shows how the chordwise variation of the non-dimensional entropy generation rate may be determined. Reducing and eventually minimising irreversibilities generated in critical areas by varying the blade profile maybe the first step in designing a stage for minimum loss using entropy analysis.

The temperature profile must be well defined in order to determine accurately the entropy generation rate. A linear relationship exists between the local entropy generation rate and the local Nusselt number as presented in figure 7. This relationship is valid for all incompressible flows with convective heat transfer as the dominant mode of entropy generation.

A new concept of the entropy sub-layer was proposed. This is a logical parameter when trying to determine the entropy generated by heat transfer and fluid friction within the boundary-layer. This parameter has helped to highlight the near wall region where the majority of entropy is being generated and the importance firstly, of correctly determining the temperature gradient at the blade surface and secondly, the mesh griding of CFD codes.

A similar approach to that presented could be adopted to evaluate the non-dimensional local and chordwise entropy generation rates due to fluid friction. More specifically, if the aerodynamic wall shear stress around an airfoil were measured and converted into a blade surface velocity gradient, then a local velocity gradient boundary condition could be formed and a typical velocity boundary-layer distribution constructed if its shape factor were defined. The shear stress gauge calibration of Duffy *et al.* (1995) is of significance in this respect.

CONCLUSIONS

- Thin film heat transfer gauges were successfully used to measure laminar surface heat transfer rates around the suction surface of a turbine blade.

• For well defined boundary conditions the temperature profile in the boundary-layer was shown to be a function of a shape factor.

• A new concept of the entropy sub-layer was proposed, which defines the near wall region within the boundary-layer where 99% of the entropy is generated.

• Surface heat transfer measurements alone are not adequate to determine the boundary-layer normal and chordwise entropy generation rates as the boundary-layer shape factor and its chordwise distribution must be known.

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