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FUNDAMENTAL MECHANISMS THAT INFLUENCE
THE ESTIMATE OF HEAT TRANSFER TO
GAS TURBINE BLADES

R. W Graham
Lewis Research Center
Cleveland, Ohio

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R. W. GRAHAM
Head, Fundamental Heat Transfer Section
NASA Lewis Research Center
Cleveland, Ohio
Fellow, ASME

ABSTRACT

The quest for improved efficiency has motivated
the elevation of turbine inlet temperatures in all
types of advanced aircraft gas turbines. The accom-
modation of higher gas temperatures necessitates
complex blade cooling schemes so as not to sacrifice
structural integrity and operational life in advanced
engine designs. Estimates of the heat transfer from
the gas to stationary (vanes) or rotating blades
poses a major uncertainty due to the complexity of
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these blade rows is three dimensional with complex
secondary viscous flow patterns that interact with
the endwalls and blade surfaces. In addition, up-
stream disturbances, stagnation flow, curvature ef-
fects, and flow acceleration complicate the thermal
transport mechanisms in the boundary layers. Some
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discussed. The chief purpose of this paper is to
acquaint those in the heat transfer community, not
directly involved in gas turbines, of the serious-
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INTRODUCTION

Improved fuel consumption performance of turbo-
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Thus, elevating the gas temperatures higher will re-
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heat flux loads on the blades and vanes. The gas
side heat transfer information is part of the itera-
tion process necessary to establish cooled blade or
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Although the general heat transfer problems are
familiar to those involved in thermal design of tur-
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fundamentals may not be obvious to those doing basic
research in unrelated areas. The main purpose of
this paper is to acquaint those in the heat transfer
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the prediction of turbine blade or vane gas-side heat
transfer is to be improved. There are equally impor-
tant research questions regarding the coolant-side
heat transfer, but they are beyond the scope of this
paper.

In fulfilling this purpose, the following items
will be discussed:

(1) A general description of the hot gases that
enter the turbine blade row as to their thermal state
and flow characteristics.

(2) A brief description of the analytical meth-
ods employed in calculating the gas-side heat trans-
fer to turbine blades or vanes. An evaluation of
their current accuracy in predicting heat transfer
will be presented.

(3) An enumeration will be made of basic flow
phenomena such as stagnation, curvature effects, ac-
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The quest for improved efficiency has motivated the elevation of turbine inlet temperatures in all types of advanced aircraft gas turbines. The accommodation of higher gas temperatures necessitates complex blade cooling schemes so as not to sacrifice structural integrity and operational life in advanced engine designs. Estimates of the heat transfer from the gas to stationary (vanes) or rotating blades poses a major uncertainty due to the complexity of the heat transfer processes. The gas flow through these blade rows is three dimensional with complex secondary viscous flow patterns that interact with the endwalls and blade surfaces. In addition, upstream disturbances, stagnation flow, curvature effects, and flow acceleration complicate the thermal transport mechanisms in the boundary layers. Some of these fundamental heat transfer effects will be discussed. The chief purpose of this paper is to acquaint those in the heat transfer community, not directly involved in gas turbines, of the seriousness of the problem and to recommend some basic research that would improve the capability for predicting gas-side heat transfer on turbine blades and vanes.

INTRODUCTION

Improved fuel consumption performance of turbojet engines necessitates elevating the cycle pressure ratio and the turbine gas inlet temperature so as to realize improved efficiency. The gas temperatures in operational aircraft engines have already reached levels where some turbine cooling is required. Thus, elevating the gas temperatures higher will require more sophisticated cooling schemes for the same turbine blade material. Vital to the heat transfer design is an accurate estimate of the local gas-side heat flux loads on the blades and vanes. The gas side heat transfer information is part of the iteration process necessary to establish cooled blade or stator designs.

Although the general heat transfer problems are familiar to those involved in thermal design of turbines and comprehensive survey papers exist (see ref 1), the relevance of these problems to heat transfer fundamentals may not be obvious to those doing basic research in unrelated areas. The main purpose of this paper is to acquaint those in the heat transfer community, not affiliated with the aircraft gas turbine industry, with some of the important basic heat transfer problems that need further investigation if the prediction of turbine blade or vane gas-side heat transfer is to be improved. There are equally important research questions regarding the coolant-side heat transfer, but they are beyond the scope of this paper.

In fulfilling this purpose, the following items will be discussed:

(1) A general description of the hot gases that enter the turbine blade row as to their thermal state and flow characteristics.

(2) A brief description of the analytical methods employed in calculating the gas-side heat transfer to turbine blades or vanes. An evaluation of their current accuracy in predicting heat transfer will be presented.

(3) An enumeration will be made of basic flow phenomena such as stagnation, curvature effects, acceleration, secondary flows and transition that influence local heat transfer rates. The status of basic research in these areas will be discussed.

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tions of basic heat transfer research that should be pursued in order to improve the certainty in predicting the gas-side heat transfer to turbine blades.

CONDITIONS OF THE HOT GAS ENTERING THE TURBINE

For most commercial transport aircraft flying today, the turbine inlet gas temperature is approximately 2000°F (1094°C) and the pressure is about 20 to 25 atmospheres. More advanced engines for military missions and new generations of commercial transports will operate with turbine inlet temperatures of 2500°F (1371°C) and above, and at pressure levels of 25 to 30 atmospheres. Futuristic plans call for advancing the turbine inlet temperature to near 2800°F (1538°C) and the pressure to 40 atmospheres. These temperatures are average values but the hot gases leaving the combustors have a temperature profile, so the first stages of the turbine see a non-uniform gas temperature. By the time the gases reach the turbine, they have been turbulently-excited in the blade row of the compressor and in the combustion process. Thus, the turbine sees turbulent, nonsteady hot gases. Further complicating the situation are the radiation properties of the gases. The first row of turbine vanes will probably see the luminescence of the flames in the combustor. The radiation component of gas-side heat transfer can be an appreciable fraction of the total heat transfer to the first stage. However, the analytical method(s) for computing the radiation component will not be discussed in this paper. It is apparent that the turbine is subjected to a rather hostile thermal environment that is not steady. Such an environment is difficult to depict analytically.

GAS-SIDE HEAT TRANSFER PREDICTION

Current design practice for estimates of the convective heat transfer for turbine blades or stators generally involves the use of a two dimensional boundary layer analysis program. Two examples of these kinds of programs are references 2 and 3. Reference 2 is a numerical method for computing either laminar or turbulent boundary layers by solving the integral momentum equation. In implementing the program, the user must make use of some sort of transition criteria or at least specify whether the boundary layer is laminar or turbulent.

A comparison of experimentally measured Nusselt number in a turbine cascade with the predicted values from the computational method of ref. 2 are shown in figure 1, taken from reference 4. Considering the fact that the analysis is a two dimensional flat plate analysis without curvature, etc. built into it, the computer program does a fair job of estimating the local heat transfer coefficient distribution. However, on the suction side of the cascade blades, an error of approximately 35% is encountered near the mid-chord. Since the theory miscalculates the heat transfer coefficient, the actual wall temperature would turn out to be in error by at least 100°F (55.5°C) for a constant gas temperature of 2500°F (1371°C). From information received through a personal communication with Robert L. Dreshfield of Lewis Research Center, a 100°F increase in the temperature of a nickel alloy at turbine blade temperatures would effect an order of magnitude decrease in the estimated life of the blade. Such a decrement in life is intolerable and emphasizes the necessity for greater precision in estimating gas-side wall temperatures.

The computational program in reference 3 (STAN5) is a differential-type which has been used specifically for estimating the gas-side heat transfer to turbine blades. It is based on the Spalding-Patankar method. Reference 5 is such an application where the program is used to estimate the gas-side heat transfer of blades that incorporate a coolant insert.

STAN5 is designed to solve the two dimensional boundary layer equations for both laminar and turbulent conditions. There is an automatic provision in the program to transition from laminar to turbulent flow according to a momentum thickness Reynolds number criterion. Along with the boundary layer momentum equation, the program has the capability of handling an indefinite number of diffusion equations and solving the momentum and diffusion equations simultaneously.

The question of transition and the effect of the free stream turbulence level on the blade boundary layer is a major uncertainty in any prediction scheme. Figure 2, taken from reference 6, shows the variation in measured local heat transfer coefficients in a cascade when the upstream turbulence was varied and shows comparisons between data and predictions. Note on the figure the predicted heat transfer coefficients: flat plate correlations and the Spalding-Patankar method on which STAN5 is based. It is evident that the Spalding-Patankar method predicted well at one turbulence level (5.9%) on the pressure surface but was not accurate on the suction surface for the same turbulence level. More will be said about turbulence level effects and upstream flow disturbances in a later section.

The point to be made in this section of the paper is that current analytical methods which are based on two-dimensional flat plate models need further refinements to account for the real flow physics on blades or vanes. Some of the real flow phenomena will be described in the next section.

PHENOMENA AFFECTING HEAT TRANSFER

In turbines, the flow is being accelerated and turned by the blading and in the rotor rows is simultaneously being acted upon by centrifugal and coriolis forces. The hub surface and the outer casing wall also play a major role in establishing the flow patterns developed in the flow passages of turbomachinery. Consequently, the gas flow through an axial flow turbine is highly complex and very difficult to model and analyze.

Figure 3 depicts a blade passage for either a rotating or stationary blade row in a turbine. In this two dimensional picture, the blade sections are at some mid-span position. The added complexities of the hub or tip section flow patterns are not presented in the sketch. As the flow enters the blade row, some of the streamlines impact on the leading edge of the blade to form a stagnation region. The boundary layer that develops in the vicinity of the stagnation zone appears to be laminar at first, then undergoes a transition process. Depending on the local Reynolds number and pressure gradient, and external flow disturbances, the boundary layer remains in a transition state or becomes turbulent around the leading edge of the blade. The streamlines that follow the suction surface continue to experience convex curvature. On the opposite surface of the blade, the streamlines encounter concave curvature. On both surfaces, local regions of flow acceleration and deceleration are encountered which could contribute to transition or retransition processes occurring in the boundary layer. At the trailing edge of the blade row, separation is apt to

develop which will influence the wake. Any internal coolant flow that is dumped out from the trailing edge of the blade or vane will also influence the wake. Note that secondary cross flows are indicated in the blade passage which vary in intensity and direction along the span of the blade passage. These cross flows will be most significant across the end wall and hub wall.

It is clear from considering the schematic of a turbine blade row that several flow-phenomena are operating which influence the heat transfer. Among the more obvious are:

    (1) impingement or stagnation flow
    (2) curvature effects
    (3) flow unsteadiness and turbulence
    (4) accelerating and decelerating flow
    (5) secondary flows
    (6) body force effects
    (7) transition and detransition processes

Fundamental forced convection heat transfer experiments have been performed and reported that deal with each of the above topics. In this paper, most (but not all) of these fundamental heat transfer effects will be discussed and an assessment rendered as to their relative contribution to the uncertainty in estimating the gas-side heat transfer to the turbine blade. The potential interaction of several of these effects will be discussed.

Impingement or Stagnation Flow

The impingement of a jet or a free stream flow on a flat or cylindrical surface has been studied quite extensively as a fluid mechanics and heat transfer problem. Although crossflow impingement on a cylinder is a truer representation of a turbine blade leading edge condition, jet impingement on a flat plate is also a useful phenomena to study. Cross-flow studies have been the source of correlations or analytical models applicable to impingement flows and the associated heat transfer at the leading edges of turbine vanes or blades.

Some early heat transfer research performed by Schmidt and Wenner of Germany (ref 7) on stagnation heat transfer to a cylinder in crossflow is relevant information. Their study incompassed flow conditions from laminar to turbulent Reynolds numbers and they made measurements of the local heat transfer rates over the entire surface of the cylinder. The measured profiles of Nusselt number around the cylinder from reference 8 are shown in figure 4. These Nusselt number profiles are useful information for estimating the leading edge heat transfer of turbine blades or vanes. It turns out that the stagnation region Nusselt Number is directly proportional to the square root of the Reynold's Number. The constant of proportionality is very close to unity but is dependent on the turbulence level of the upstream flow. In reference 9, a similar stagnation relationship was observed for impingement of a jet on a flat plate. For that case, the constant of proportionality is also sensitive to the upstream level of turbulence.

For the region just beyond the stagnation locus on the leading edge of the blade, an estimate of the local heat transfer coefficient distribution has to be made. For cross flow on a cylinder, one approximate approach is to modify the stagnation correlation with an angular correction. The correct term is

$(1 - \left|\frac{\phi}{90°}\right|^3)$ where ϕ is measured in degrees from the stagnation point to the locus in question on the front surface of the cylinder (ref 10). The fact

that the Reynolds Number is raised to the one half power implies the initiation of a laminar boundary layer at the stagnation region and the geometric factor corrects for boundary layer growth around the edge of the cylinder.

Turbulence and Flow Unsteadiness

One of the most important influences on the local heat transfer distribution on the leading edge and on the pressure and suction surfaces of a turbine vane or blade is the time-unsteady condition of the flow as it enters the blade or vane row. In this time-unsteady category, two principal unsteady contributions have been lumped together, namely turbulence level and the pulsating or unsteady characteristics caused by the interaction of a rotating blade row with a stationary vane row. Frequently, these two effects are not considered separately. Furthermore, in an actual turbojet engine application, the unsteadiness of the flow coming into the turbine has been affected by the compressor and, more significantly, by the combustion process. Obviously, the flow reaching the turbine has been subjected to a number of influences that make the flow anything but idealized steady flow with a known turbulent structure. In fact, a purist may object to the term "turbulence" being used to denote the structure of the flow as it applies to turbines. The argument being that the usual sources of the turbulent production involved in a turbulent flow experiment may be relatively unimportant and noninfluencial in a turbine application. Thus, when a turbine designer refers to the turbulent structure of the flow entering the turbine, he is not generally talking about the classical turbulence level definition of the aerodynamist. In the turbine application, the unsteady flow intensity is defined as the unsteady flow component normalized with respect to an average flow velocity at some station in the system. From such a definition, rather large values of intensity can result as compared to ordinary turbulence. Some results of intensity measurements at the exit of combustors show values exceeding 50%.

Concerning this issue of how to define a disturbance intensity, the author of reference 11 suggested three different definitions of disturbance. Referring to a typical velocity hot wire record shown in figure 5, the following definitions were proposed.
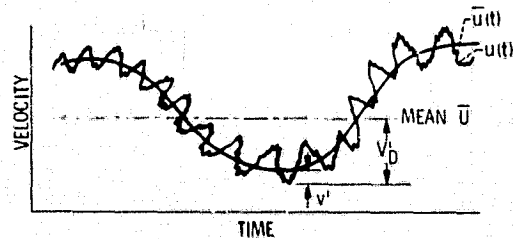

Figure 5. - Hot wire trace of unsteady flow.

(a) Overall Disturbance Level

$$Tu_D = \frac{\sqrt{(v'_D)^2}}{\bar{U}}$$

(b) Free-stream Turbulence Level

$$\hat{Tu} = \frac{\sqrt{(v')^2}}{\bar{U}}$$

(c) Unsteadiness Level

$$\tilde{Tu} = \frac{\sqrt{[u(t)-\bar{u}(t)]^2}}{\bar{U}}$$

These three definitions are interrelated by the following equation

$$Tu_D^2 = \hat{Tu}^2 + \tilde{Tu}^2 \qquad (1)$$

This manner of distinguishing the various types of disturbances has some merit and is an answer to the objections raised about intensity definitions that do not discriminate between disturbance intensity and turbulence intensity.

More important than the definitions are the actual effects of these manifestations of flow unsteadiness on the local heat transfer rates. The designer must have some idea of an "influence coefficient" that will establish a level of perturbation beyond the steady-state assumption normally used in either correlations or more elaborate numerical methods.

Although the flow in a turbine is unsteady, measurements pertaining to the nature of the boundary layer have revealed that laminar-like as well as turbulent boundary layers are possible. Consequently, laminar unsteady flow is a relevant topic to consider.

Not a great deal of research has been done on unsteady flow--especially experimental verification of analytical models. In ref 12, Lighthill derived expressions for the friction coefficient as a function of the Strouhal number. (The Strouhal number ($\tilde{\omega}$) is a dimensionless frequency normalized with respect to a reference free-stream velocity.) His prediction is prescribed to apply to two domains. For example, the high Strouhal number correlation for C_f is

$$\frac{C_f}{2} \sqrt{Re_x} = 0.332 + B\tilde{\omega}\cos(\omega t + \frac{\pi}{4}) \qquad (2)$$

where B is a coefficient in the idealized expression for the unsteady flow.

Analytical studies of unsteady laminar boundary layers on rotating blades have been reported in reference 13. Using a perturbation solution of the energy equation, the variance in the heat transfer in response to a sinusoidal disturbance was estimated for various locations over the blade. Large perturbations in the level of heat transfer (approaching an order of magnitude) are predicted in comparison to the flat plate values calculated from the Blasius equation.

Turbulence Effects on Stagnation Flow

Professor Joseph Kestin of Brown University and his colleagues have performed a number of investigations on the effect of upstream turbulence on stagnation heat transfer. In fact, their early work is generally referenced by those conducting more re-

search on the effect of upstream disturbances. In reference 14, the authors pointed out that a decided trend in enhancement was observed over a range of turbulence intensities. The biggest changes in the heat transfer tended to occur at the low turbulence levels and to gradually flatten out as the turbulence intensity reached higher values. In 1970, Kestin and Wood (ref 15) published a paper which elucidated the mechanism considered to be responsible for the enhancement in the stagnation region. They pointed out that the flow field in a laminar boundary layer outside a stagnation line is not two dimensional. It has a vortical structure which results from an instability different from the classic Gortler-type. The authors developed a mathematical model of this instability. Physically, the vortices align themselves on parallel axes perpendicular to the flow vector and are uniformly spaced along the upstream portion of the stagnation body. Their spacing is a function of $(Re)^{-1/2}$ and the level of the upstream turbulence. Increasing the turbulence reduced the spacing and enhanced the stagnation heat transfer.

In 1977 Sadek et al (ref 16) did a visual study of these vortices associated with stagnation flow across a cylinder. From high speed cinematography and still pictures they gathered evidence about the scale and distortion of these vortices as they negotiated around a cross-flow cylinder. As the vortices first approached the cylinder they tended to stretch into longer vortices with higher rotational velocities and smaller scale. They also tilted such that the axes of the vortices coincided closely with the flow streamlines around the cylinder. When the vortices were close enough to the cylinder to interact with the laminar boundary layer, the vortex scale increased appreciably. This interaction process was thought to sponsor transition and, consequently, higher heat transfer rates. Their visual images of the vortices showed them to have a coherent vortex structure.

In reference 17, the author carried out an analytical estimation of the augmentation effected by these vortex tubes. The estimate of 8% augmentation fell considerably below the experimental value of 19%. It appears as though it will be difficult to estimate the augmentation by analytical means only.

Although jet impingement on a flat plate does not represent the flow field at the leading edge of a turbine blade, information about the turbulent structure near the stagnation zone may aid our general understanding of the stagnation mechanism where impingement flow is involved.

In reference 18, turbulence intensities of an impinging jet on a flat plate were measured for three jet velocities. The spacial distribution of intensity for one of these jet velocities (138 m/s) is shown in figure 6. Note that the intensity varies from .05 to 0.16 and a zone of high intensity occurs close to the plate. It may seem incongruous that high intensity turbulence appears so close to the stagnation region where a laminar-type heat transfer correlation is recommended involving the square root of the Reynolds number. Stagnation on a flat plate and cylinder appear to be similar mechanisms. For both, the free stream turbulence was amplified near the stagnation zone. Such amplification augments the local heat transfer between the fluid and the wall. If the boundary layer initiates at the stagnation locus on a cylinder as a laminar boundary layer, it must be a considerably different boundary layer development than the classic laminar boundary layer that begins at the sharp edge of a flat plate in a two-dimensional flow field. For stagnation flow, the aforementioned tur-

bulent amplification must encroach on the stability of the laminar boundary layer.

Enhanced Turbulence on a Flat Plate

In the past six years, several studies have been devoted to the study of the effect of free stream turbulence level on the transport processes in a flat plate turbulent boundary layer. In reference 19, sandpaper grit was mounted at the leading edge of the flat plate and turbulence generating rods of various diameters and spacings were mounted upstream of the test section. The nature of the boundary layer on the flat plate was detected by static pressure taps, crossed hot wire anemometers and pressure/temperature probes. It was concluded that an augmented free stream turbulence level did thicken the flat plate boundary layer and dramatically influenced the wake region of the layer. It was also noted that the properties of the inner portion of the boundary layer were not greatly influenced by the free stream turbulence. In general, however, the boundary layer takes on a non-equilibrium quality because of the advection of free stream turbulent energy and the non-equilization between production and dissipation of turbulent energy within the boundary layer.

In reference 20, it was reported that the turbulent shear stress was increased across the entire thickness of the layer when the free stream turbulence was augmented. As would be expected from the Reynolds analogy between heat and friction, it was observed that the heat transfer increased appreciably also. If the flat plate boundary layer was laminar, 50% increases in the heat transfer rates were observed when the free stream turbulence was boosted from essentially zero to 9%. Substantially smaller increases in heat transfer were observed when the boundary layer was already turbulent.

In reference 21, rectangular grids were employed to augment the turbulence upstream of an electrically heated flat plate. It was observed that the free stream turbulence had little effect on heat transfer when its intensity was less than that of the boundary layer. But when the free stream turbulence exceeded the intensity of the boundary layer, definite augmentation in heat transfer was observed. The trend in the heat transfer and friction data due to augmentation of free stream turbulence is showed in figure 7.

The reader is referred again to figure 2 which shows the changes in levels of heat transfer with free stream turbulence measured in a two dimensional cascade. From four references cited herein about flat plate heat transfer, it is apparent that free stream turbulence is influencial in controlling the magnitude of the local boundary layer heat transfer.

Wake Disturbances

The nature of the wake flow region in turbomachinery has been investigated by Professor Lakshminarayana and colleagues of Penn State University. Hot wire measurements were made in three directions simultaneously. Through a system of data storage and computerized data processing, the perturbations of flow in the axial, tangential and radial flow directions were determined. In reference 22, the wake region behind a free vortex compressor rotor was examined at eight radial locations and five axial stations downstream of the rotor. The wake defect ratio of the axial velocity decayed in an exponential manner. The influence of a wake on the heat transfer of a downstream blade row is not well understood and warrants further research.

Curvature Effects

There is an extensive literature about curvature effects on turbulent flow in channels or pipes. No attempt will be made herein to cite a comprehensive bibliography on the subject. A very thorough review of the literature appears in ref 23 by Bradshaw. For the purposes of this report, a few important conclusions from this literature will be shared.

Turbine vane or blade geometry involves concave curvature on the pressure surface and convex curvature on the suction side. As would be expected intuitively, the two types of curvature produce different effects. The flow channel between two adjacent blades (or vanes) is a segment of a curved channel and this geometry has been popular in curvature experiments. In a curved channel, even with an appreciable aspect ratio, a considerable amount of secondary flow exists. Consequently, a pure two-dimensional flow without any secondary flows is not encountered in the experimental research. However, this simulates the type of secondary flow pattern observed in a vane flow passage caused by the endwalls. Visualization of the secondary flow on the endwalls of a 6:1 aspect ratio channel reported in ref 24 showed evidence of appreciable crossflow from the concave wall across to the convex wall.

Generally, the turbulence intensity on the convex wall is somewhat lower than the comparable flat plate. The convex curvature tends to inhibit turbulence production. As a consequence, both the convective heat transfer rates and wall friction are lower than the flat plate values. The local values depend on the radius and the arc length of the curvature, but decrements of 20% in friction and heat transfer as compared to the flat plate, are possible.

On the concave surface, the turbulence intensity is augmented by Taylor-Gortler type vortices which align themselves in the direction of the flow. The enhanced turbulence on the concave wall elevates the heat transfer rates and the friction. For curvatures in which the boundary layer thickness to radius of curvature ratio is approximately 0.015, the enhanced heat transfer and friction may be anywhere from 26% to 40% above the flat plate value depending on the arc length (ref 24).

There have been several analytical or empirical methods proposed to explain the effect of curvature on turbulent flow. Based upon Prandtl's treatment of centrifugal body forces, Bradshaw has suggested that the Prandtl mixing length can be made a function of the Richardson number, which depicts the relative influence of buoyancy to shear production. The gradient Richardson number, Ri, is defined as

$$Ri = \frac{-g}{\rho} \frac{\partial \rho}{\partial y} / \left(\frac{\partial U}{\partial y} \right)^2 \tag{3}$$

where g = gravity
ρ = density
y = dimension normal to the surface
U = free stream velocity

In Prandtl's original treatment of the effect of body forces on the mixing length, he showed that a modified mixing length would be a function of a parameter equivalent to one half of the gradient Richardson number. It turns out that correction to the flat plate mixing length, ℓ_o can be written as

$$\frac{\ell}{\ell_o} = 1 - \alpha U/r / \frac{\partial U}{\partial y} \tag{4}$$

where r is radius of curvature and α is an empirical constant with a representative value of 10.

In Bradshaw's report (ref 23) the ratio ℓ/ℓ_0 is referred to as "F". Depending upon the direction of the curvature (concave or convex), the magnitude of F varies from less than unity to something greater than unity. A suggested range might be from 0.5 to 1.5. The above equation must be interpreted as a simplification and is very empirical even though it involves modification to the mixing length parameter used in complex analyses of the turbulent boundary layer.

In an analytical boundary layer method presented in ref 25, the effects of streamline curvature re accounted for by a proposed eddy viscosity function that is a product of the flat plate eddy viscosity and a function of a curvature parameter. The turbulence energy equations are simplified so that all terms can be expressed as double velocity correlations. Further simplifications result from the assumption that the turbulent energy production balances the turbulent dissipation. In the turbulent energy equations, length scales (see page 121, reference 25) are introduced as proportionality constants to relate the energy terms of the equations. It turns out that only two of the length scales are independent and these can be determined from the law of the wall and the plane flow turbulence energy equation.

In reference 25, some of the experimental work is compared to the analytical prediction of several of the boundary layer integral parameters. For instance, the growth of the experimental form factor H (ratio of displacement thickness to momentum thickness) on the convex wall agreed with the analytical prediction. Also, the analytical model of the boundary layer proved to be reliable for predicting the coefficient of friction. As would be expected, the friction could not be predicted when separation was imminent.

Corroboration of the concave surface boundary layer characteristics was not achieved because the presence of the longitudinal Taylor-Gortler vortices made averaging of the boundary layer flow data very difficult.

Eide and Johnston (ref 26) modified a flat plate turbulent boundary layer code developed at Stanford University (STAN5) for the effects of curvature. The mixing length was corrected in the same fashion as Bradshaw; viz by a factor involving the Richardson number.

Patel (ref 27) formulated a two dimensional integral boundary layer equation for curved channel flow utilizing a polar coordinate system. Included in his formulation were the static pressure variations across the boundary layer. As has been observed by a number of investigators, the pressure gradient terms ignored in flat plate boundary layer theory cannot be ignored for curved channel geometries. The selection of polar coordinates did simplify the derivation of the integral momentum equation, but still it turned out to be a much more complicated expression than the familiar flat plate information. Also, Patel redefined the integral momentum and displacement thicknesses for the polar coordinate system. He carried out some experimental work with a 90° bend channel and compared the experimental results with his analytical model. The experimental boundary layer shape factor agreed with the analytical prediction.

Most of the experimental investigations of curvature effects have involved fluid mechanics studies and only a few have dealt directly with curvature

effects on heat transfer. There is not much comparable information about the mechanism of heat transport within the turbulent thermal boundary layer. However, in reference 24, comparison of the boundary layer profiles for the temperature and velocity layers showed similarity. Without more detailed information, one would have to assume that the Reynolds analogy applies. Therefore, the mixing length correction factors for the thermal boundary layer would probably be the same as the comparable viscous boundary layers.

Acceleration and Deceleration Effects on Turbulent Boundary Layers

The difficult heat transfer problems in rocket nozzles encountered in the late 50's and 60's motivated research investigations of accelerating and decelerating turbulent boundary layers. Experiments were carried out in two-dimensional and axisymmetric flow channels. As would be expected from physical intuition, acceleration and deceleration effects alter the structure of turbulent eddies. For instance, acceleration causes stretching of turbulent eddies which results in reduced turbulence intensity. In fact, rapid acceleration rates could bring about a phenomenon known as "relaminarization" in which a boundary layer that was originally turbulent would tend to behave more like a laminar boundary layer. Figure 8, taken from ref 28, illustrates the almost-discontinuous drop in heat transfer that accompanies the relaminarization phenomenon in a nozzle. Note from the figure that relaminarization occurred in a Reynolds number range of one to two million. The onset of relaminarization is not very predictable, just as the transition from laminar to turbulent flow is not predictable either. A parameter has been devised that has proved to be a valuable indicator of relaminarization susceptability. Such a parameter reflects the degree of acceleration and is generally referred to as the "acceleration parameter K". It is defined as:

$$K = \frac{\mu_e}{\rho_e U_e^2} \frac{du_e}{dx} \tag{5}$$

The subscrip e denotes the edge of the boundary layer; μ is viscosity, ρ is density, and u is velocity.

Note from figure 8 that there is a definite trend toward lower heat transfer rates than the flat plate turbulent flow levels. As the Reynolds number is reduced, the measured Stanton number departs from the turbulent level and establishes a new curve below the flat plate turbulent value. Thus, it can be concluded that acceleration tends to diminish heat transfer rates in turbulent flow below the zero-pressure gradient flat plate values.

Deceleration brings about just the opposite effects from acceleration. In an axisymmetric facility with the same inlet configuration as reported in reference 28, it was demonstrated that sizeable enhancements in heat transfer and turbulence intensity would result from decelerated flow. As shown in figure 9 from reference 29, the local Stanton number rose to ratios of 3.25 times the zero pressure gradient value after deceleration. Turbulence intensity escalated to 40% in the center of the diffusing section. As is indicated on figure 9, separation is a definite possibility during a severe deceleration.

Secondary Flow

The flow in turbines is complicated by the presence of three dimensional viscous flows that are sub-

ject to pressure gradients inherent to a particular type of turbomachine. Substantial research efforts are underway to delineate these secondary flows experimentally and to create computer programs capable of describing the complex flow patterns. Ideally one would like to have the three dimensional, compressible flow Navier-Stokes equations in a useable analytical form.

The categories of secondary flow to be found in turbomachines were rather completely discussed in a series of NACA reports. Their contents are summarized in ref 30. Smoke, ink and paint streaks, and other visual techniques were employed to demonstrate the physical existence of secondary flow patterns. The visual techniques employed did not allow complete mapping of the complex secondary flow patterns inferred from pressure surveys at the trailing edge of stators or blades. However, the researchers were able to discern several important secondary flow patterns with their techniques. Among them were the following types of secondary flows:

(1) A "Passage Vortex" in the endwall of a cascade. A portion of the boundary layer in the endwall of a cascade experiences an overturning due to static pressure gradients acting on it. As this low momentum fluid negotiates the blade passage it experiences an overturning caused by the pressure gradient between the pressure and suction surfaces of the cascade. It eventually hugs the suction surface in the endwall corner and forms a vortex stream on exiting from the cascade. It was demonstrated that this passage vortex continued this behavior when it entered a downstream blade row. I: continued to be influenced by the pressure-to-suction pressure gradient and thus, did not experience the same turning as the main flow.

More recent investigations of the behavior of the passage vortex in ref 31 have revealed that overturning of the boundary layer on an endwall can result in the separation of that boundary layer near the leading edge of a blade row. Subsequently, the separated boundary layer becomes a passage vortex. A thin, laminar boundary layer replaces the separated one.

(2) "Radial Secondary Flow" was observed in a stator row of an annular cascade in reference 30. From shear streaks apparent on nondrying paint and from pressure surveys made at the exit of the stator row, it was concluded that most of the low energy boundary layer accumulated at the hub of the cascade. Some secondary flow was evident at the tip and in the wake region, but the majority was associated with the hub region. A radically different distribution of secondary fluid would occur in a rotating blade row. The imposition of a centrifugal force field promotes secondary flow migration toward the tip region of the rotor where the major portion of the low energy boundary layer will accumulate. Thus in a turbine, the secondary flow will build up alternately at the hub and tip regions of the machine. With multistaging, these buildups could lead to sizeable accumulations of secondary fluids in the final stage or stages of axial turbomachinery. Obviously, the presence of large accumulations of secondary flow will influence aerodynamic performance and heat transfer in the regions affected.

In reference 30, the authors were able to visualize vortex formation in the tip clearance region of a cascade. This vortex was completely separate from the "passage vortex" described earlier. Both vortices were observed simultaneously and exhibited opposite rotations and did not show any tendency to mix when near to one another. Despite the apparent independence of these vortices, the tip clearance secondary flow was a significant contributor to the overall secondary flow effects observed in a cascade. There has to be an interaction between the "passage" and "tip" vortices in establishing the overall secondary flow. The local convective heat transfer will be coupled to the secondary flow pattern.

Transition

Characterizing the boundary layer as laminar or turbulent is a major question in prediction of the heat transfer from the hot gas to the blade or stator surface of a turbine stage. Therefore, it would be highly desirable if a general analytical method would exist for predicting the nature of the boundary layer. In the predictive method, STAN5 transition is determined by a momentum thickness criteria. However, what might be termed a general method for predicting transition conditions does not exist. Some success has been enjoyed in particular cases wherein experimental information provided reference data that could be used in linear stability theory to predict the change in laminar or turbulent conditions in the boundary layer.

A number of linear stability theories have been developed as mentioned in ref 32. For some conditions, linear stability seems to work well but it is also possible to find cases where it fails. In transition theory, it is the inherent stability of the boundary layer flow that is being analyzed. In flow applications such as turbine blading, external disturbances frequently trigger transition conditions and these external influences cannot be easily incorporated into the theory.

With regard to turbines, there appears to be the necessity of accumulating more experimental data on the nature of the boundary layers on the suction and pressure surfaces of blades and vanes. There are so many simultaneous effects that bear upon the boundary layers to cause transition or detransition conditions. In fact, any of the flow factors mentioned in this paper will influence transition.

CONCLUSIONS AND RECOMMENDATIONS

From what is known about predicting the local heat transfer rates associated with flow over blades or vanes in turbines, the following listed effects contribute to excessive predictive uncertainty.

1. At the leading edge of a blade or vane there is insufficient knowledge about the interaction of the unsteady, turbulent free stream with the stagnation boundary layer. Research has confirmed that a system of vortices develop upstream with their axes normal to the axial flow plane. The interaction of the upstream vortex array with the growing boundary layer in the region of stagnation needs further experimental investigation and the results thereof incorporated into an analytical model of the boundary layer development for that region.

2. For the pressure and suction surfaces, curved channel experiments should be executed that simulate more closely the initial boundary layer history of flow over a blade or vane. The blade boundary layers begin differently from the curved channel boundary layers cited in the literature. Differing boundary layer history can affect local convective heat transfer rates appreciably.

3. The influence of secondary flows on the

structure of the surface boundary layers needs further assessment. It is recommended that development of an analytical model of the secondary flows in a turbine passage be given high priority. Once confirmed as an analytical tool, the results of this program can be used in the assessment of secondary flow effects on heat transfer. It is recognized that the secondary flows are particularly severe at the endwalls, but they are present in every zone of the blade or stator passage. A more quantitative representation of the secondary flow behavior is needed to improve the heat transfer prediction.

4. Of general interest for the entire blade or vane geometry is the effect of free stream turbulence scale and intensity on the structure of the boundary layers and the attendant heat transfer. A similar issue is the effect of free stream unsteadiness on the structure of the thermal and velocity boundary layers. In the real engine, flow unsteadiness and induced turbulence are what the turbine sees. The real engine flow unsteadiness and turbulence has to be characterized so that the experiments simulate actual conditions, including combustion turbulence. The major uncertainty in predicting gas side local heat transfer rates anywhere on the blade is the interaction of free stream unsteadiness and turbulence with the boundary layers on the blade. Such interaction will determine the nature of the boundary layer, control the mechanism of transition and in the last analysis, establish the levels of heat transfer.

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TURBULENT FLAT PLATE LAW $Nu_x = 0.0292\ Re_x^{1/5}\ Pr^{1/3}$

LAMINAR FLAT PLATE LAW $Nu_x = 0.332\ Re_x^{1/2}\ Pr^{1/3}$

STAGNATION POINT OF A CIRCULAR CYLINDER $Nu_d = 1.14\ Re_d^{1/2}\ Pr^{0.4}$
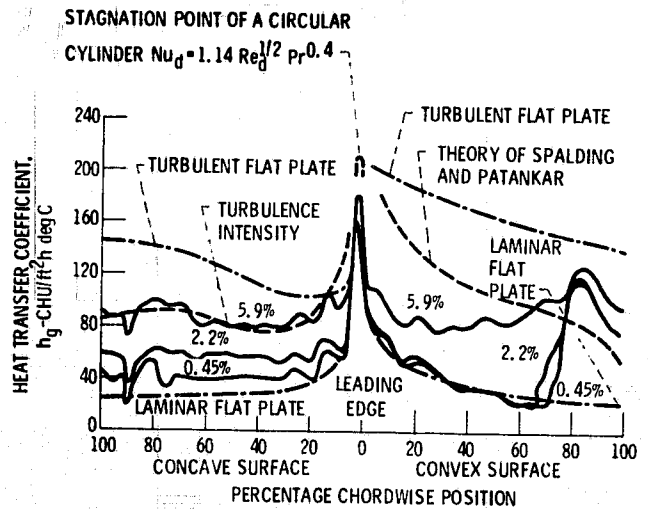
Figure 2. - Experimental distribution of turbine vane heat transfer coefficient for an exit Mach number of 0.75.

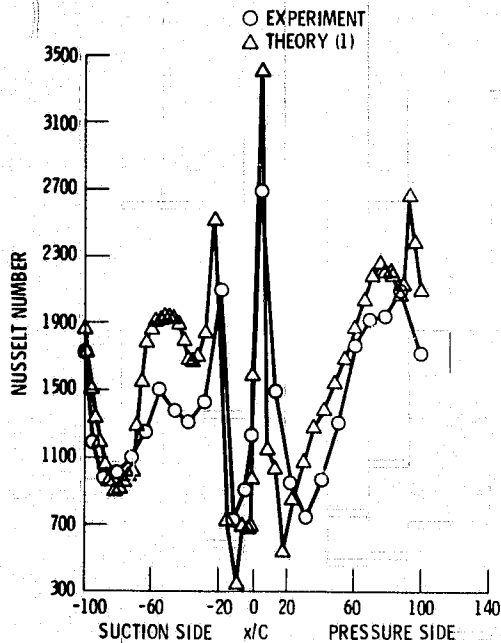Reprinted by permission of the Council of the Institution of Mechanical Engineers from reference 6.

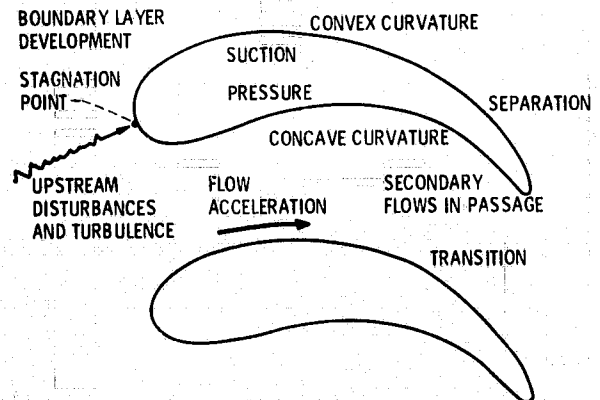Figure 1. - Comparison between $Nu_{experimental}$ and $Nu_{theoretical}$ prediction for a turbine blade in a cascade.

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Figure 3. - Schematic of turbine blade row and associated convective phenomena.
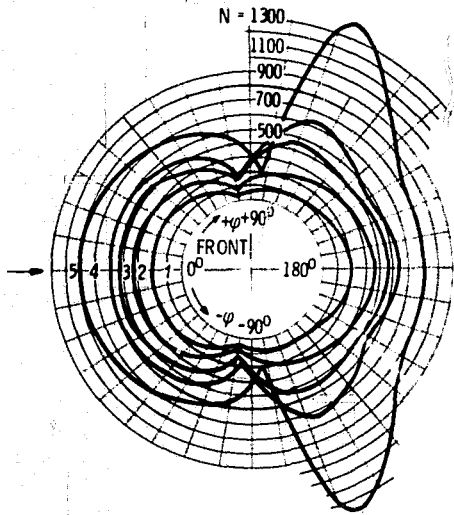
Figure 4. - Local coefficient of heat transfer for a circular cylinder at varying Reynolds numbers as measured by E. Schmidt and K. Wenner (ref. 7). Curves (1) and (2) refer to the region below the critical Reynolds number, curves (3) and (4) were measured in the critical range, and curve (5) above the critical range. (1) R = 39 800, (2) R = 101 300, (3) R = 170 000, (4) R = 257 600, and (5) R = 426 000.
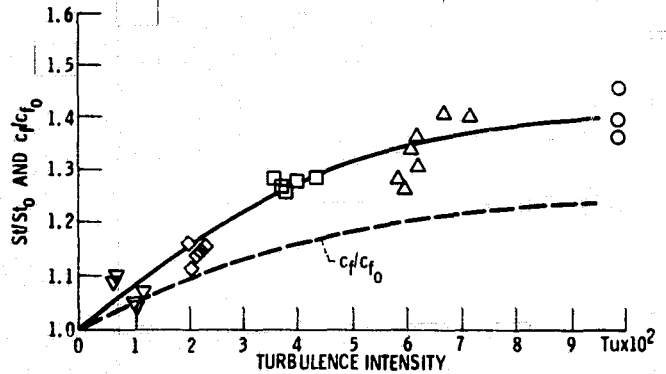
"From reference 8. Used with permission of McGraw-Hill Book Company."

Figure 7. - Variation of Stanton and coefficient of friction due to the free-stream turbulence.

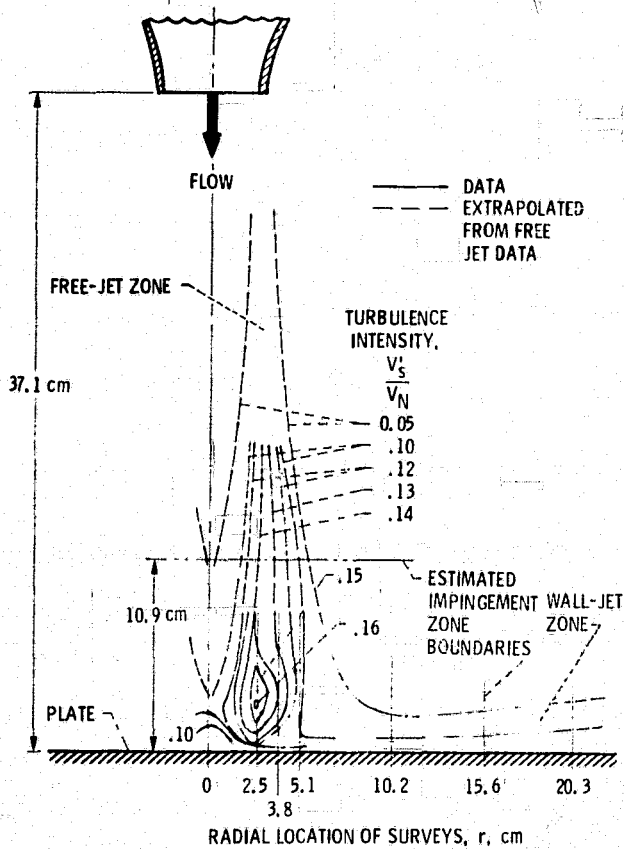From reference 21. Used with permission of Hemisphere Publishing Corporation.

Figure 6. - Contours of constant turbulence intensity in impinging flow. Nozzle-exit velocity, V_N, 138 meters per second.

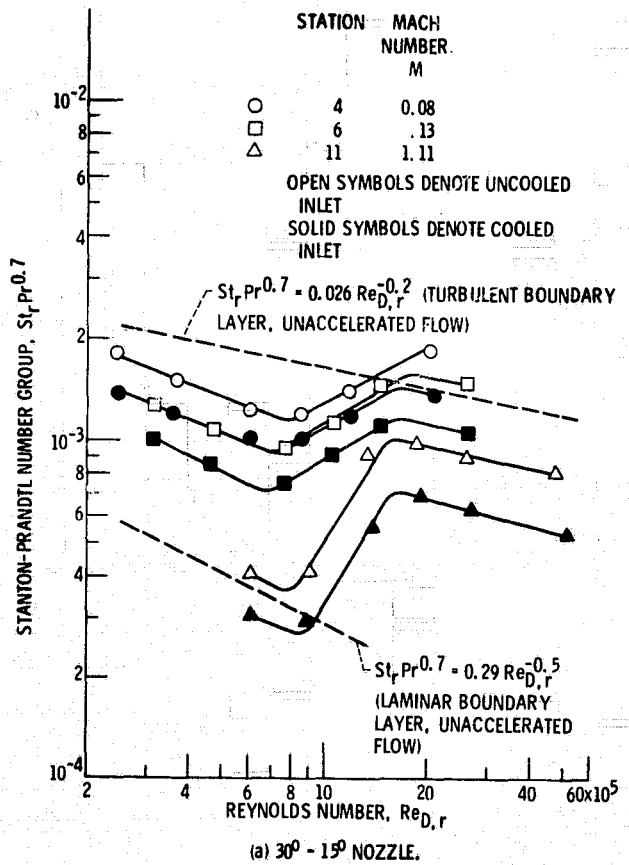(a) $30^0 - 15^0$ NOZZLE.

Figure 8. - Heat transfer change during flow acceleration in a nozzle.
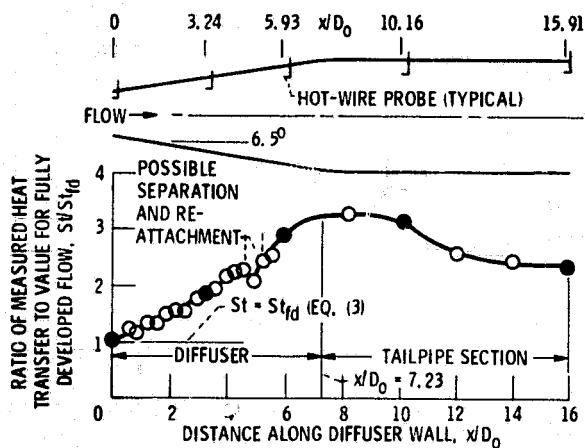
Figure 9. - Heat transfer in conical diffuser and tailpipe.
Entrance diameter, D_0 = 7.62 centimeters. Solid
symbols denote boundary layer survey locations.