179809

NASA
Technical Memorandum 106008
AIAA–91–2129

Army Research Laboratory
Memorandum Report ARL–MR–14

13 P

Two-Dimensional Navier-Stokes Heat Transfer Analysis for Rough Turbine Blades

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(NASA-TM-106008) TWO-DIMENSIONAL NAVIER-STOKES HEAT TRANSFER ANALYSIS FOR ROUGH TURBINE BLADES (NASA) 13 p

N94-13027

Unclas

G3/34 0179809

Prepared for the
27th Joint Propulsion Conference and Exhibit
sponsored by the AIAA, SAE, and ASME
Sacramento, California, June 24–27, 1991

U.S. ARMY
RESEARCH LABORATORY

TWO-DIMENSIONAL NAVIER-STOKES HEAT TRANSFER ANALYSIS FOR ROUGH TURBINE BLADES

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ABSTRACT

A quasi-three-dimensional thin-layer Navier-Stokes analysis was used to predict heat transfer to rough surfaces. Comparisons are made between predicted and experimental heat transfer for turbine blades and flat plates of known roughness. The effect of surface roughness on heat transfer was modeled using a mixing length approach. The effect of near-wall grid spacing and convergence criteria on the accuracy of the heat transfer predictions are examined. An eddy viscosity mixing length model having an inner and outer layer was used. A discussion of the appropriate model for the crossover between the inner and outer layers is included. The analytic results are compared with experimental data for both flat plates and turbine blade geometries. Comparisons between predicted and experimental heat transfer showed that a modeling roughness effects using a modified mixing length approach results in good predictions of the trends in heat transfer due to roughness.

Nomenclature

A_f - Frontal area
A_s - Windward wetted area
A^+ - Damping coefficient
C_f - Friction factor
D - Diameter of hemispherical roughness
h - Roughness height
h_{eq} - Equivalent roughness height
L - Distance between roughness elements
ℓ - Length scale
Pr_t - Turbulent Prandlt number
p - Pitch between roughness elements
Re - Unit Reynolds number
S - Reference surface area
S_f - Total frontal area
s - Surface distance
St - Stanton number based on inlet conditions
U - Inviscid velocity
u_τ - Friction velocity
y - Distance normal to surface
y^+ - Normalized distance, $(yU_e/\nu)\sqrt{C_f/2}$
δ - Full boundary layer thickness
κ - Von Karman's constant
Λ - Roughness density parameter
μ - Dynamic viscosity
μ_t - Turbulent eddy viscosity
ν - Kinematic viscosity
ν_t - Turbulent eddy viscosity, (μ_t/ρ)
ρ - Density
ω - Vorticity

Subscripts

$_e$ - Edge of boundary layer
$_m$ - Measured
$_p$ - Predicted

INTRODUCTION

The effect of inaccuracies in predicting turbine blade temperatures significantly affects turbine durability. As discussed by Graham[1], a $50°C$ underprediction of blade temperature can decrease service life by a factor of two. Taylor[2] measured the surface roughness of in-service turbine blades. His results showed that for typical Reynolds numbers the surfaces are not hydraulically smooth. Since actual turbine blades are rough, heat transfer predictions assuming smooth surfaces can significantly underpredict external heat transfer. This in turn could result in underprediction of the blade surface temperatures. Both two-dimensional boundary layer analyses, such as the STAN5 code of Crawford and Kays[3], and Navier-Stokes analyses, such as those of Chima[4] and Hah[5], have been used to predict smooth turbine blade heat transfer. Models to account for surface roughness in boundary layer analyses have been developed by Cebeci and Chang[6], and by Taylor, Coleman, and Hodge[7]. Navier-Stokes analyses have the advantage over boundary layer analyses in that they can be used to predict blade heat transfer beyond the point of separation. The present work discusses the results obtained when a surface roughness model was incorporated into the quasi-three-dimensional thin-layer Navier-Stokes analysis developed by Chima[4].

There are two different philosophies with regard to modeling surface roughness. The approach taken by Taylor et al. [7] explicitly accounts for roughness geometry. This approach accounts for blockage due to the roughness elements, the increased drag due to their presence in the flow field, as well as a correlation to account for additional heat transfer between the fluid and the discrete roughness elements. Consequently, this approach requires detailed knowledge of the surface roughness, as well as empirical correlations for the drag and heat transfer to the roughness elements. Hosni, Coleman, and Taylor[8] have used this model in a boundary layer analysis to predict heat transfer for a rough flat plate over a wide range of Reynolds numbers. The other philosophy, as expressed by Cebeci and Chang[6], is to model the roughness effect based on the offset in the near-wall velocity profile. In this model the mixing length is modified to account for the effective height of the roughness. The effective height is a function of both roughness density and shape. Computationally, the model of Cebeci and Chang is significantly easier to implement into a Navier-Stokes analysis than the more detailed model of Taylor et al. Also, Hosni et al.[8] reported that their boundary layer calculations, using the model of Taylor et al., were done using 250 points within the boundary layer. This is far more than is practical to use in a Navier-Stokes analysis for turbine blade heat transfer. Consequently, the model of Cebeci and Chang was implemented in the Navier-Stokes analysis.

Navier-Stokes heat transfer predictions can be affected by the near-wall grid spacing. The sensitivity of the heat transfer predictions for both smooth and rough surfaces will be examined. Because roughness promotes very early transition, heat transfer to rough surfaces can be assumed to be fully turbulent. The effect of different approaches to turbulent eddy viscosity modeling in the leading edge stagnation region is discussed. Comparisons are made with experimental data to demonstrate the suitability of the roughness models for predicting turbine blade heat transfer. The data of Hosni, Coleman, and Taylor[8] for flow over a roughened flat plate was used for comparisons. Comparisons were made for the heat transfer on turbine blades using both the data of Liebert[9] and the data of Blair and Anderson[10].

METHOD of ANALYSIS

The roughness model of Cebeci and Chang[6] was incorporated into the Navier-Stokes analysis of Chima[4] by modifying a version of the Baldwin-Lomax [11] turbulent eddy viscosity model. In this model there is an inner and outer region. The turbulent eddy viscosity in the inner region is given by:

$$(\mu_t)_{\mathrm{INNER}} = \rho \ell^2 |\omega| \qquad (1)$$

The length scale, ℓ, is given by:

$$\ell = \kappa y (1 - e^{-y^+/A^+}) \qquad (2)$$

Where $y^+ = y\nu/u_\tau$ and $A^+ = 26$. In the outer region the turbulent eddy viscosity is given by:

$$(\mu_t)_{\mathrm{OUTER}} = C_{\mathrm{OUTER}} \rho F_{\mathrm{WAKE}} F_{\mathrm{KLEB}} \qquad (3)$$

Where $C_{\text{OUTER}} = 0.0269$, and F_{WAKE} is given by:

$$F_{\text{WAKE}} = min(y_{\text{MAX}} F_{\text{MAX}}, .25 y_{\text{MAX}} v^2_{\text{DIF}} / F_{\text{MAX}}) \tag{4}$$

The term v_{DIF} is the maximum velocity along the near normal grid line out from the surface. The quantities y_{MAX} and F_{MAX} are found from:

$$F(y) = y|\omega|\left(1 - e^{-y^+/A^+}\right) \tag{5}$$

y_{MAX} is the value of y for which $F(y)$ is a maximum. F_{KLEB} is given by:

$$F_{\text{KLEB}} = \left[1 + 0.004(y/y_{\text{MAX}})^6\right]^{-1} \tag{6}$$

For rough surfaces Cebeci and Chang use an offset distance, Δy, which is added to the physical normal distance. This was implemented by replacing y by $y + \Delta y$ in the above equations. For example, equation 2 for a rough surface becomes:

$$\ell = \kappa(y + \Delta y)\left(1 - e^{-(y^+ + \Delta y^+)/A^+}\right) \tag{7}$$

At the wall $y = 0$, but $\ell > 0$ when the surface is rough. Therefore, $\mu_t > 0$ at the wall. The heat transfer predictions were made using an effective wall thermal conductivity. Pr_t was assumed equal to 0.9 for the determination of the effective wall conductivity.

Cebeci and Chang give Δy as:

$$\Delta y^+ = 0.9\left(\sqrt{h^+_{\text{eq}}} - h^+_{\text{eq}} e^{-h^+_{\text{eq}}/6}\right) \tag{8}$$

Cebeci and Chang state that equation 8 is valid in the range $4.535 < h^+_{\text{eq}} < 2000$. Figure 1 shows the variation of Δy^+ with h^+_{eq}. When the equivalent roughness height, h^+_{eq}, equals 4.535, Δy^+ is zero. This automatically accounts for the fact that small roughnesses, $(h^+ < 5)$, are hydraulically smooth. For large values of h^+_{eq} the slope of the curve progressively decreases. Consequently, the effect of increasing roughness height becomes less significant as the roughness becomes large.

In the boundary layer analysis of smooth walls the length scale, ℓ, is held constant when κy exceeds 0.086δ. In the Navier-Stokes analysis the boundary layer thickness, δ, is not clearly defined, and is not used in the Baldwin-Lomax turbulence model. For rough surfaces the turbulent eddy viscosity in the outer region was calculated in two ways. First, y was unmodified in the outer-region eddy viscosity equations. In the second way, y was replaced by $y + \Delta y$ in the outer-region eddy viscosity equations. The consequence of these assumptions on the rough surface heat transfer will be discussed.

The Baldwin-Lomax turbulence model has an abrupt crossover model. In this model, when $(\mu_t)_{\text{OUTER}} > (\mu_t)_{\text{INNER}}$, μ_t is taken as the outer value. Granville[12] discusses two other crossover models which blend the inner and outer regions. These two models are:

$$\frac{\nu_\tau}{\nu} = \left(\frac{\nu_\tau}{\nu}\right)_{\text{OUTER}} \tanh \frac{(\nu_\tau/\nu)_{\text{INNER}}}{(\nu_\tau/\nu)_{\text{OUTER}}} \tag{9a}$$

and

$$\frac{\nu_\tau}{\nu} = \left(\frac{\nu_\tau}{\nu}\right)_{\text{OUTER}} \left(1 - e^{\frac{-(\nu_\tau/\nu)_{\text{INNER}}}{(\nu_\tau/\nu)_{\text{OUTER}}}}\right) \tag{9b}$$

The effect on the predicted heat transfer using each of these models will be discussed.

RESULTS and DISCUSSION

Heat transfer predictions are known to be sensitive to the near-wall grid spacing in a Navier-Stokes analysis. This sensitivity could be different depending on whether the surfaces are rough or smooth. Figure 2 shows the sensitivity of heat transfer predictions to the near-wall spacing for a representative rotor geometry for both smooth and rough surfaces. The results are given in terms of a reference y^+. The same definition is used as was used by Boyle in reference 13. The definition is:

$$y^+_{\text{REF}} = 0.17 y_1 Re^{0.9}/s^{0.1} \tag{10}$$

Where y_1 is the distance from the surface of the first grid line. Re is the exit Reynolds number per unit of length. y^+_{REF} is only a weak function of the surface distance, s, and s is taken as the axial chord. The same definition is used for the reference surface roughness, h^+_{REF} except that y_1 is replaced by h_{eq}.

$$h^+_{\text{REF}} = 0.17 h_{\text{eq}} Re^{0.9}/s^{0.1} \tag{11}$$

The calculations are for fully turbulent flow. The smooth wall results shown in figure 2a are not very sensitive to the near wall spacing. A greater sensitivity was shown in reference 13 for a similar comparison. The sensitivity was greater when the flow was laminar, or close to transition.

Figure 2 shows a somewhat greater sensitivity to the near wall spacing for the rough surface than for the smooth one. The reference values were calculated assuming a smooth surface, and the calculation depends on the inverse of the friction factor. Thus, for rough surfaces the reference values is lower than one based on the correct friction factor by the ratio $\sqrt{(C_f)_{\mathrm{ROUGH}}/(C_f)_{\mathrm{SMOOTH}}}$. The rough surface friction is not known before the analysis is done, and equation 11 is useful for determining the grid spacing. Consequently, the rough wall heat transfer shown in figure 2b is more sensitive to the near-wall spacing than the smooth wall data shown in figure 2a.

The primary purpose of this investigation was to examine the effects of surface roughness on heat transfer predictions. Consequently, grids with close near-wall spacing were used. Calculations were typically done using 145 × 54 C-grids. Approximately 5000 iterations were needed to insure convergence. The computations took about 700 CPU seconds on a Cray-XMP. It is expected that further optimization of the code and calculation procedure would result in faster convergence.

Figure 3 shows the effect of the different crossover models for fully turbulent flow. The calculations are for a smooth wall. The difference in the calculated heat transfer among the different models is not great. Of the two blending models, equation 9b is closer to the Baldwin-Lomax crossover model. The Baldwin-Lomax model results in a heat transfer prediction that has greater fluctuations between adjacent nodes than either of the other crossover models. This is consistent with the abrupt nature of the crossover model. For some cases the solution stability or convergence might be enhanced using a blended crossover model.

The equivalent roughness height, h_{eq}, is a function of the actual roughness and the roughness density. A number of correlations have been proposed to obtain h_{eq}. Sigal and Danberg[14] discuss various correlations for obtaining the equivalent height. The ratio of equivalent to actual roughness height, h_{eq}/h, is correlated as a function of the roughness density parameter, Λ. Figure 4 shows their results with different correlations for both two- and three-dimensional roughness. Λ is defined differently for the different correlations. The definition used by Sigal and Danberg was:

$$\Lambda_{\mathrm{SD}} = \frac{S}{S_f}\left(\frac{A_f}{A_s}\right)^{-1.6} \tag{12a}$$

The definition used by Dvorak[15] as interpreted by Simpson[16] was:

$$\Lambda_{\mathrm{DV}} = \frac{S}{S_f} \tag{12b}$$

The definition used by Dirling[17] was:

$$\Lambda_{\mathrm{DR}} = \frac{p}{h}\left(\frac{A_f}{A_s}\right)^{-1.33} \tag{12c}$$

Sigal and Danberg reasoned that three-dimensional roughness has a lower h_{eq} because flow is able to go around as well as over a three-dimensional obstacle. Since different definitions were used for Λ, Dvorak's correlation does not necessarily result in a lower value of h_{eq}.

In the correlation of Sigal and Danberg, and in the correlation of Dvorak, for the same roughness shape, Λ is proportional to the height-to-pitch ratio for two-dimensional roughness. However, for three-dimensional roughness Λ is proportional to the square of this ratio. Consequently, even for closely spaced roughness, only two-dimensional roughness results in Λ values in the region where the slope of the correlation is positive.

Hosni et al.[8] conducted experiments on smooth and rough flat plates. The rough surface test sections were formed from machined aluminum plate, and had integral hemispherical roughness elements. They were of diameter D, and were spaced one of three distances, L, apart to form a three-dimensional roughness array. This resulted in three different roughness densities. Table I shows the equivalent height ratio for different spacings using the three definitions of Λ. Because Λ was defined differently by the different investigators, there is less variation in the equivalent height ratio than might be expected from figure 4. The equivalent height ratio varied from about 0.02 to somewhat greater than 1.0 as L/D varied from 2 to 10.

Figure 5 shows comparisons of predicted and measured heat transfer over a flat plate for different equivalent roughness heights. Since the Navier-Stokes analysis used here was developed for turbomachinery applications, the flat plate results were calculated as a thin uncambered airfoil in a low solidity cascade. The airfoil had an elliptical leading edge with a 10:1 radius ratio. This was done to avoid any separation in the leading edge region, as might occur with a circular leading edge. The experimental data are shown for clarity as individual points, but were derived from a curve fit through data. As the Reynolds number increases the predicted heat transfer becomes greater than the experimental data for the smooth surface. The predicted heat transfer is greater by nearly 10%. In the Navier-Stokes analysis the solidity may not have been low enough. There was some small acceleration of the freestream flow due to boundary layer growth. This would increase the heat transfer above that for an isolated flat plate.

Table II compares the ratio of predicted to measured heat transfer for the different roughness spacings at a Reynolds number of 400000. Comparisons are given for the values of h_{eq}/h from the different roughness correlations. Also shown in this table is the ratio of predicted to measured heat transfer for the smooth surface. The analysis predicts the trends in the experimental data. Generally, the best agreement is found when the lowest value of h_{eq}/h for a given spacing is used in the analysis. The most recent correlation of roughness data is that of Sigal and Danberg. This is also the correlation that agrees best with the data. Sigal and Danberg showed that spherical segment data might deviate significantly from their proposed correlation. This is the roughness geometry that most closely resembles the roughness elements used by Hosni et. al. A line using the extremes of the spherical segment data would be about a factor of 3 less in equivalent height ratio than the three-dimensional correlation of Sigal and Danberg. Consequently, it is not suprising that a heat transfer analysis using the correlation of Sigal and Danberg would overpredict the heat transfer when hemispherical roughness elements are used.

There is little rough surface turbine blade heat transfer data available that is suitable for comparisons with analytic predictions. Suitable data would have well defined information on the roughness characteristics, and not just the roughness height. Measurement techniques suitable for smooth blades may not be suitable for rough surfaces. For example, Taylor, Taylor, Hosni, and Coleman[18] showed that surface mounted sensors which do not have the same roughness characteristics as the blade itself may indicate heat transfer rates different from that of the blade. If the surface of the blade is artificially roughened using a low conductivity material such as sand, this material may effectively move transition forward on the surface, but also form a thermal resistance. These requirements suggest that experimental heat transfer data for rough turbine blades should include baseline smooth surface data. The experimental data of Tarada[19] showed that for many cases there was no increase in heat transfer for rough blades when the flow is turbulent. In these data, in which roughness was achieved by coating a smooth blade with sand, the roughness only tripped the boundary layer close to the leading edge. The early tripping of the boundary layer resulted in an increased average surface heat transfer. However, for many cases the heat transfer was lower in the fully turbulent region for rough blades than for smooth blades. The analysis does not predict this behavior. The predicted effect of roughness for turbine blades is similar to the predictions for a flat plate shown in figure 5.

In contrast to the data of Tarada, the data of Blair and Anderson[10] show higher heat transfer for a roughened blade in the fully turbulent region. While the roughness height was given, no information was given to determine the roughness density. Sand was used to roughen the surfaces, but the tests were conducted in a much different thermal environment. The significance of an insulating layer may have been less in this environment than in the tests of Tarada. The tested blade was the rotor of a large-scale low-speed annular cascade. The turbulence intensity is high in front of the blade due to the presence of the upstream stator. Figure 6 compares predicted and measured heat transfer for the smooth surface. Two predictions are shown in figure 6a. The one in best agreement with the data was for a turbulence intensity of 10%. The augmentation of laminar heat transfer was included until the flow was fully turbulent.

The transition model, which is very sensitive to the turbulence intensity, is the one described in reference 13. The other calculation neglects the effect of turbulence with respect to both transition and augmentation of laminar heat transfer. Comparisons between the two calculations show the effect of freestream turbulence to be large.

Rough surface predictions are shown in figure 6b. Calculations are shown for different values of the equivalent height ratio. Since sand was used to roughen the blade, an equivalent height ratio near unity is expected. In fact this does give reasonably good agreement with the data. The pressure surface heat transfer calculation is less sensitive than the suction surface to the assumed value of equivalent height ratio.

Figure 7 shows heat transfer predictions for both smooth and rough surfaces of the turbine rotor blade tested by Liebert[9]. Only a rough surface blade was tested. The primary purpose of the test was to verify a heat flux measurement technique. Measurements were made on a 1:1 scale SSME high-pressure fuel turbine rotor blade. Because of the relatively small size only one sensor was installed on each blade. Only three blades were instrumented in the test. To obtain a complete mapping of the blade heat transfer, several blades could be instrumented. The surface roughness profiles were measured. Both the surface roughness height, h, and the roughness density, Λ, were calculated. The equivalent height ratio, h_{eq}/h, was determined to be between 0.2 and 0.4. Predictions are shown for different height ratios as well as for a smooth blade. The tests were conducted in a turbine tester rig. This rig matched the engine turbine inlet temperature, but operated at a lower pressure than the actual engine. The lower pressure resulted in lower than engine Reynolds number. If the roughness height were the same, the value of h^+ in the engine would be higher than in these tests. These tests, like those of Blair and Anderson, were also in a high turbulence environment. Because of the high turbulence intensity and high Reynolds number, fully turbulent predictions are shown. The data are in reasonably good agreement with the prediction for an equivalent height ratio of 0.1. The figure also shows that had the roughness been of the same height, but of a different density, the heat transfer might have been significantly higher.

The fully turbulent predictions shown in figure 7 used an averaging process near the stagnation point. There is symmetry in flow properties on either side of the grid line that intercepts the stagnation point. Consequently, the calculated vorticity is nearly zero along this line. This in turn results in a turbulent eddy viscosity, μ_t, that is nearly zero along the grid line which leads up to the stagnation point. For grid lines adjacent to this line, the vorticity is large, as is μ_t. In the analysis μ_t was taken as the average of the values at five grid lines in the stagnation region.

CONCLUSIONS

Predictions for rough surface heat transfer exhibit a greater sensitivity to the near wall spacing than do those for smooth surfaces. This results in the need for closer near wall grid spacing for the rough surface predictions.

The modified mixing length approach of Cebeci and Chang can be used to predict the trends in heat transfer rates with roughness. The disagreement between the analysis and the experimental data could generally be accounted for by the uncertainty in the assumptions regarding roughness. The primary uncertainty is the equivalent height.

The results of this investigation illustrate the importance of having a well characterized surface definition for experiments used to validate rough turbine blade heat transfer predictions. Not only should the roughness height and density be known, but the thermal characteristics of the roughness should also be known. For the data to be suitable for comparison with analytic predictions it is necessary to quantify any additional thermal resistivity caused by the roughness.

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Table I. - Equivalent height ratios for roughness geometries of Hosni et al.

	Equivalent Height Correlation					
	Sigal &Danberg		Dvorak		Dirling	
L/D	Λ	h_{eq}/h	Λ	h_{eq}/h	Λ	h_{eq}/h
2	30.8	1.58	10.2	2.43	10.0	1.69
4	123	0.257	40.7	0.50	20.1	0.45
10	772	0.023	257	0.061	50.3	0.079

Table II. - Ratio of predicted to measured heat transfer for a flat plate at $Re_x = 400000$. Data of Hosni et al.

	Equivalent Height Correlation					
	Sigal & Danberg		Dvorak		Dirling	
L/D	h_{eq}/h	St_p/St_m	h_{eq}/h	St_p/St_m	h_{eq}/h	St_p/St_m
2	1.58	1.64	2.43	1.85	1.69	1.67
4	0.26	1.36	0.50	1.57	0.45	1.52
10	0.023	1.01	0.061	1.15	0.079	1.22
Smooth		1.10		1.10		1.10

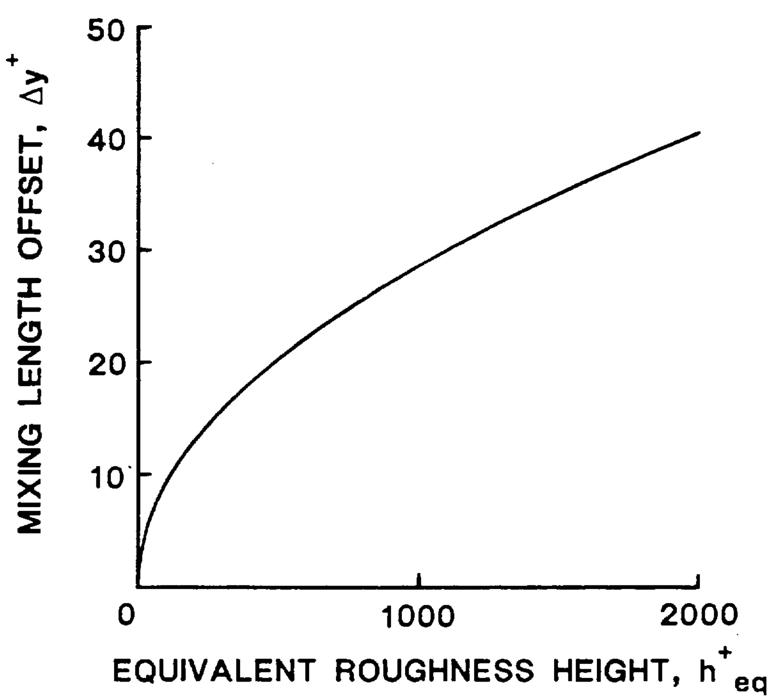
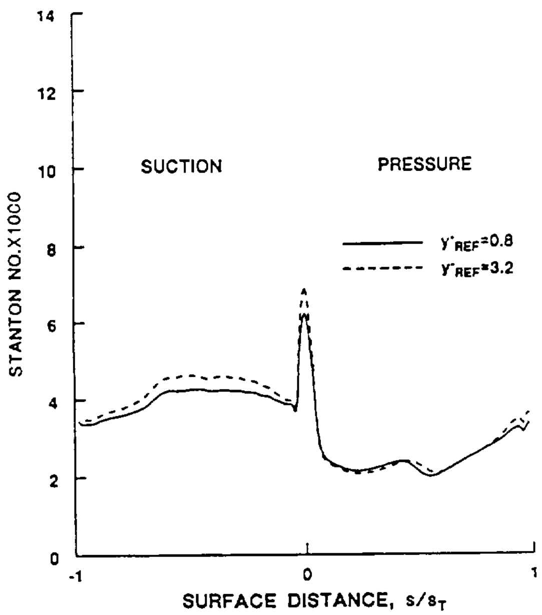


Fig. 1 Offset in mixing length as a function of roughness height.



a) Smooth surface

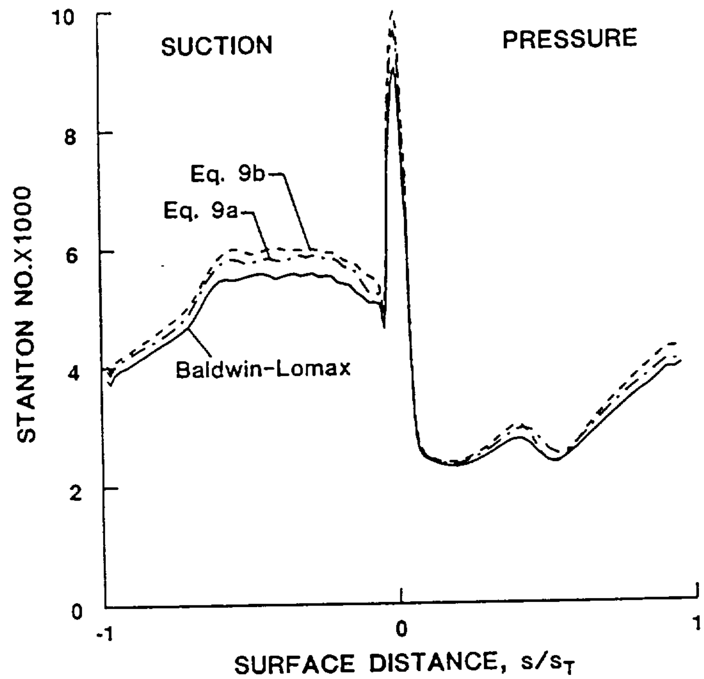
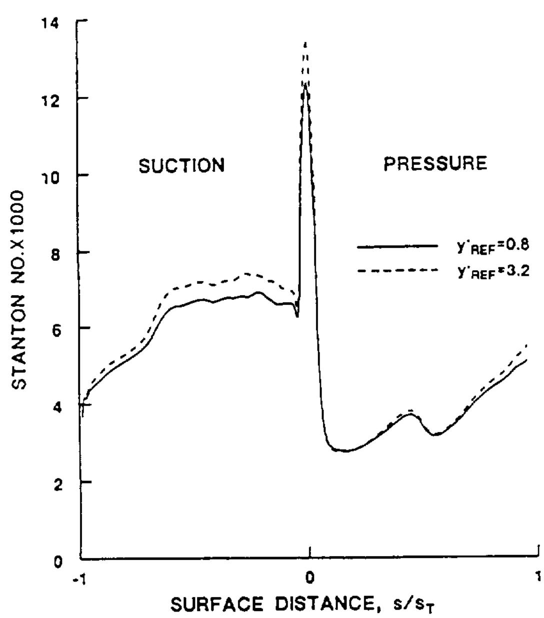


Fig. 3 Effect of crossover model on blade heat transfer for smooth blade



b) Rough surface,h^+_{REF}=60.

Fig. 2 Effect of near-wall spacing on calculated blade heat transfer distribution.

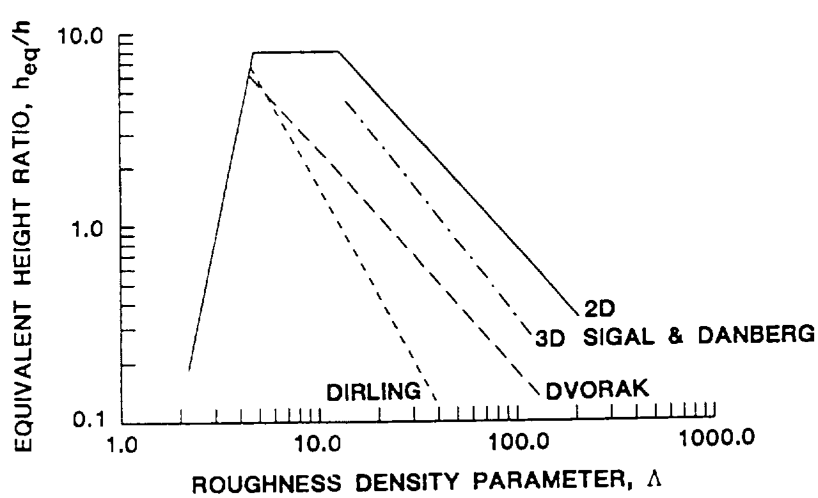


Fig. 4 Equivalent roughness height ratio as a function of roughness density parameter

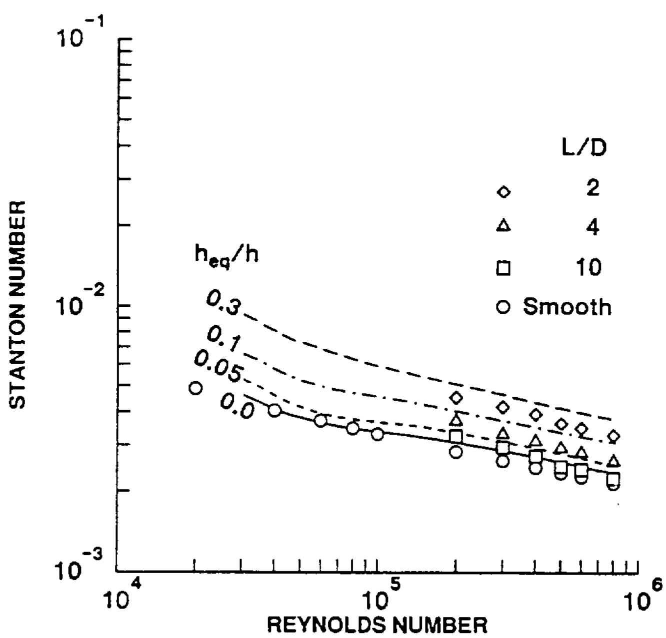


Fig. 5 Calculated and measured heat transfer for a smooth and rough flat plate

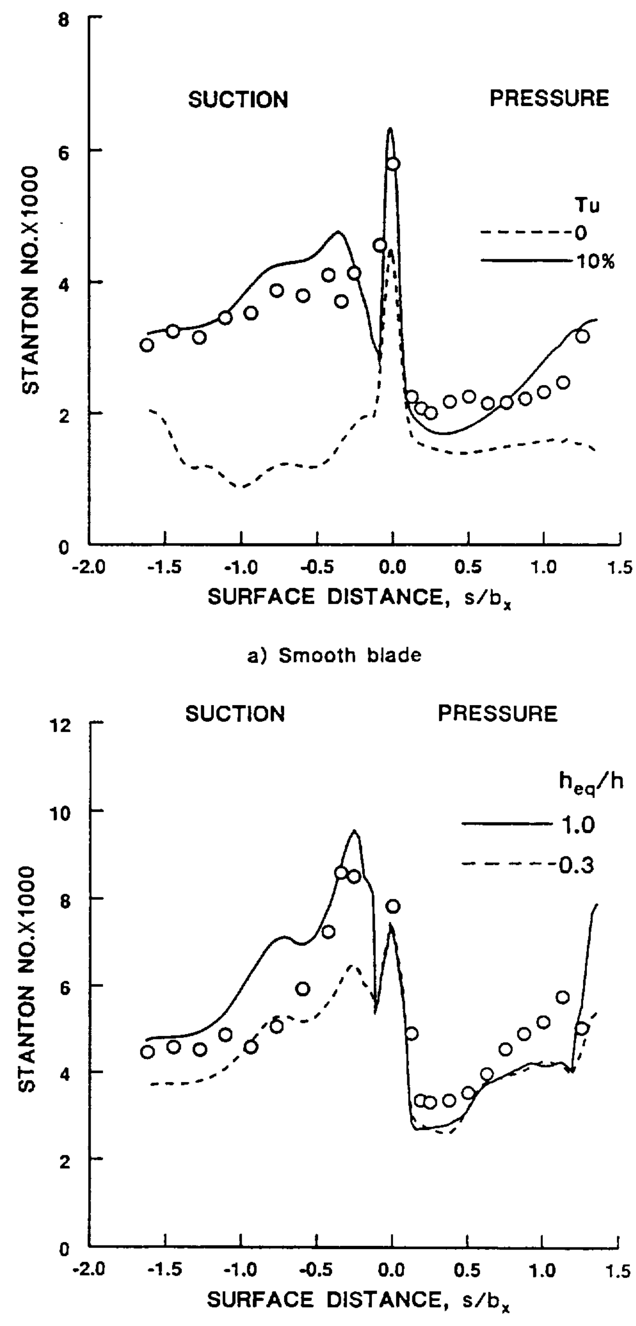


Fig. 6 Comparison of predicted and measured rotor blade heat transfer. Data of Blair and Anderson.

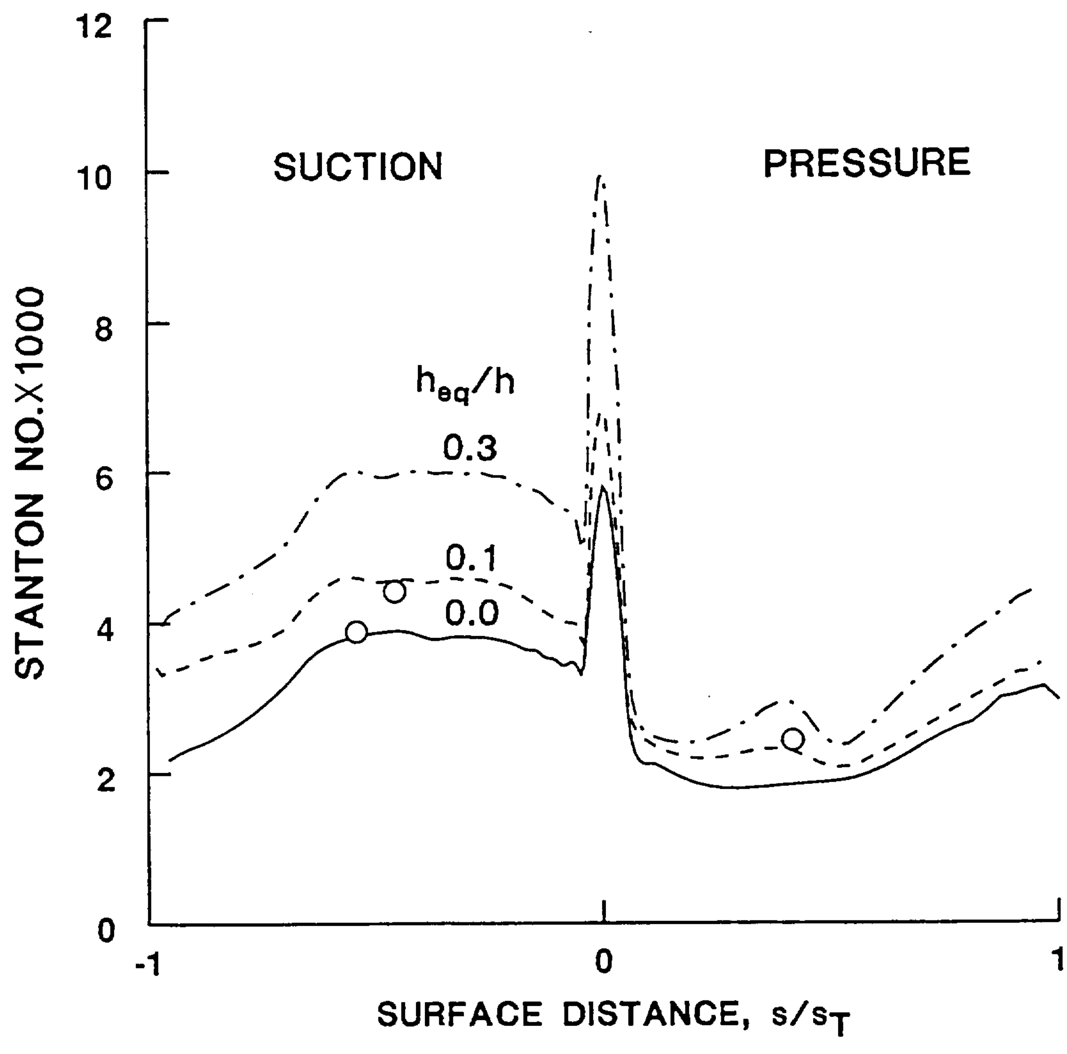


Fig. 7 Comparison of predicted and measured rotor blade heat transfer. Data of Liebert.

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY (*Leave blank*)	2. REPORT DATE	3. REPORT TYPE AND DATES COVERED
	June 1991	Technical Memorandum

4. TITLE AND SUBTITLE
Two-Dimensional Navier-Stokes Heat Transfer Analysis for Rough Turbine Blades

5. FUNDING NUMBERS
WU–505–62–52

6. AUTHOR(S)
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8. PERFORMING ORGANIZATION REPORT NUMBER
E–7755

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)
National Aeronautics and Space Administration
Washington, D.C. 20546–0001
and
U.S. Army Research Laboratory
Adelphi, Maryland 20783–1145

10. SPONSORING/MONITORING AGENCY REPORT NUMBER
NASA TM–106008
AIAA–91–2129

11. SUPPLEMENTARY NOTES
Prepared for the 27th Joint Propulsion Conference and Exhibit sponsored by the AIAA, SAE, and ASME, Sacramento, California, June 24–27, 1991. R.J. Boyle, NASA Lewis Research Center; and K.C. Civinskas, Vehicle Propulsion Directorate, U.S. Army Research Laboratory, Lewis Research Center, Cleveland, Ohio 44135. Responsible person, R.J. Boyle, (216) 433–5889.

12a. DISTRIBUTION/AVAILABILITY STATEMENT
Unclassified - Unlimited
Subject Category 34

12b. DISTRIBUTION CODE

13. ABSTRACT *(Maximum 200 words)*

A quasi-three-dimensional thin-layer Navier-Stokes analysis was used to predict heat transfer to rough surfaces. Comparisons are made between predicted and experimental heat transfer for turbine blades and flat plates of known roughness. The effect of surface roughness on heat transfer was modeled using a mixing length approach. The effect of near-wall grid spacing and convergence criteria on the accuracy of the heat transfer predictions are examined. An eddy viscosity mixing length model having an inner and outer layer was used. A discussion of the appropriate model for the crossover between the inner and outer layers is included. The analytic results are compared with experimental data for both flat plates and turbine blade geometries. Comparisons between predicted and experimental heat transfer showed that a modeling roughness effects using a modified mixing length approach results in good predictions of the trends in heat transfer due to roughness.

14. SUBJECT TERMS
Turbine; Heat transfer; Rough surface

15. NUMBER OF PAGES
12

16. PRICE CODE
A03

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
Unclassified	Unclassified	Unclassified	