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A Comparison of the Analytical and Experimental Performance of the Solid Version of a Cooled Radial Turbine

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A COMPARISON OF THE ANALYTICAL AND EXPERIMENTAL PERFORMANCE OF THE SOLID VERSION OF A COOLED RADIAL ROTOR

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Abstract

An evaluation of the aerodynamic performance of the solid version of an Allison-designed cooled radial turbine was conducted at NASA Lewis' Warm Turbine Test Facility. The resulting pressure and temperature measurements are used to calculate vane, rotor, and overall stage performance. These performance results are then compared to the analytical results obtained by using NASA's MTSB (MERIDL-TSONIC-BLAYER) code.

Introduction

Because of its high stage work and efficiency advantage over the axial turbine, the radial turbine offers a venue for improvement in small engine performance. However, to capitalize on the performance advantage of the radial turbine, an increase in inlet temperature capability is required. Recently, many joint Army-NASA research efforts have been conducted in the high temperature radial turbine area. Reference 1 discusses a split blade fabrication method for a cooled radial turbine conducted by Solar Turbines Incorporated. In reference 2, Allison presents the results of an aerodynamic test using a cooled rotor made by bonding a cast MAR-M247 air-cooled shell to a P101 powder metal hub. Garrett summerizes their attempt at fabricating and evaluating a cooled radial turbine featuring directionally solidified (DS) MAR-M247 laminated blades in reference 3. Finally, Pratt & Whitney describes the fabrication and testing of a turbine stage with air cooled nozzle and rotor sections in reference 4. The results from each of the reports indicate the feasability of a cooled radial turbine in a rotorcraft application.

As part of NASA's research program to improve turbine performance for rotorcraft application, Snyder and Roelke, reference 5, reported on the design of an air-cooled metallic radial turbine that was to be tested in the Lewis Small Warm Turbine Facility. Allison fabricated a solid and a cooled version of this rotor. The external geometries of the two rotors are identical. The test rotors were scaled up approximately 1.8 times size to allow more space for instrumentation with which to take more detailed information. The experimental plan for these rotors include an overall stage aerodynamic evaluation and analysis of the stage including rotating blade surface pressure measurements, calculation of blade heat flux, and a detailed mapping of the blade external flow using LDV measurement techniques.

This paper details the completion of the first step in the experimental plan using the solid version of the rotor. It focuses on the data obtained at design speed. At design speed the stage total to total pressure ratio was varied from 2:1 to 5.5:1 with the design at 4:1. The experimental aerodynamic results are compared herein with calculations obtained by using MTSB. Mass flow, efficiency, and blade loading define the

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comparison characteristics.

Stage Description

Table 1 describes the engine and rig design values for the solid version of this turbine. With a test design speed of 19475 rpm, the rotor ran at speeds between 80 and 120 percent of design. The physical characteristics of the stage include a stator with fifteen blades, with a chord of 5 inches, that turn the flow approximately 73 degrees. The rotor has 13 blades made of cast MAR-M247 with a tip diameter of 14.4 inches and an exit shroud diameter of 9.39 inches.

	Engine	Test
T_0', ^oR	2760.0	859.7
P_0', psia	200.0	29.8
$\dot{m}$, lbm/sec	4.56	3.998
$\Delta h'$, Btu/lbm	186.846	59.613
N, rpm	61900.0	19475.34
P_0' / P_4'	3.6648	4.049
η',uncooled	0.87	0.87
Re	381622	381622
POWER, hp	1205.00	337.21
Dt , inches	8.021	14.4

Table 1. Design conditions

Apparatus, Instrumentation, and Procedure

Reference 6 describes the test equipment and capabilities of NASA Lewis' Warm Turbine Test Facility. Figure 1 shows a cross section of the turbine test package. The instrumentation provided information at six locations from the scroll inlet, station 0, to the far rotor exit, station 4.

The equipment involved in determining the overall efficiency include the stationary rakes at stations 0 and 4. The flow rakes at station 4 were aligned to the exiting flow within the probe incidence limits. The actual specific work was calculated from the total temperatures obtained from the inlet and exit rakes at stations 0 and 4. An ideal value of enthalpy change was also calculated using the measured total pressures. The efficiency

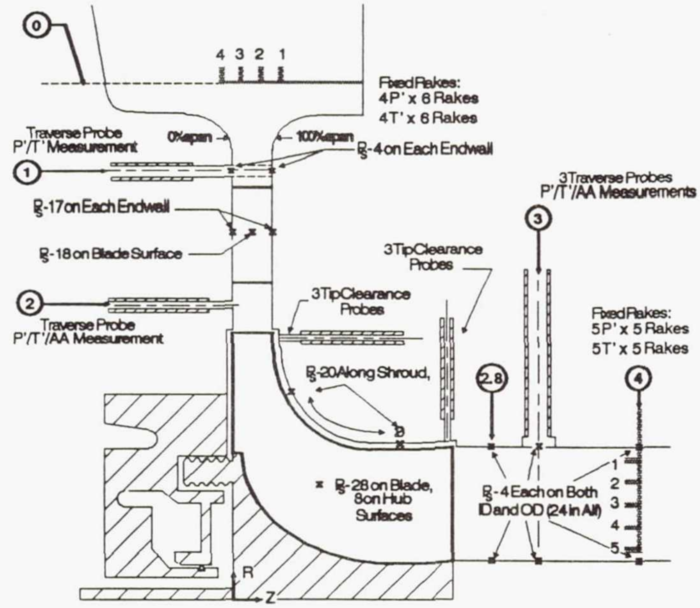

Figure 1. Turbine Test Package

presented in this paper is the observed change in enthalpy divided by the ideal change.

A venturi flow meter measured the mass flow upstream of the inlet plenum. An equivalent mass flow was obtained by multiplying the measured mass flow by the equivalent parameters obtained by normalizing the total pressure, temperature and the ratio of specific heats. The equivalent mass flow is presented in terms of pounds per second.

Between stations 1 and 2, a series of pressure taps allowed for static pressure measurement on both endwalls and the stator surface. Fourteen static taps circumscribed the meridional streamline of one stator blade with two additional taps located at the leading edge, one near the hub and one near the tip. Two taps were similarly situated at the throat. Figure 2 shows the profile of a stator blade with the location of the static pressure taps. The static pressure measurements were defined as a ratio of the measured static pressure compared to the averaged total inlet pressure obtained from the rakes at station 0.

The rotor was instrumented with 36 static pressure taps. Twenty-eight taps were distributed on the pressure and suction sides of the blade at approximate streamline locations of 20 percent and 70 percent span. The remaining 8 taps were placed in the mid-channel of the hub region. Figure 3 shows the location of the 36 taps. Six tip clearance probes were distributed at the inlet and exit of the rotor (after station 2 and before station 2.8). They recorded tip clearances of 34 mills at the inlet and 16.5 mills at the exit. The backface clearance measured 58 mills.

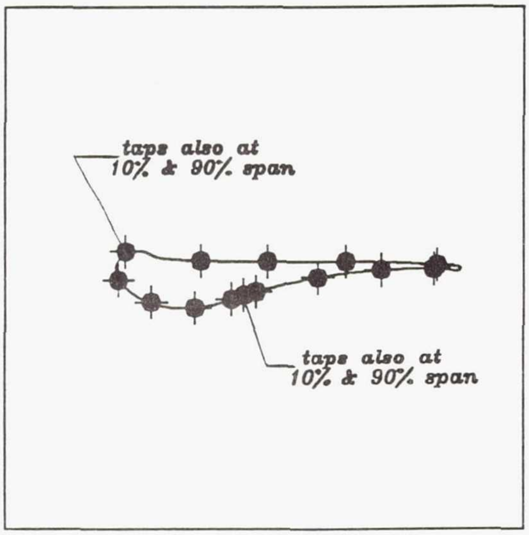

Figure 2. Stator pressure tap locations

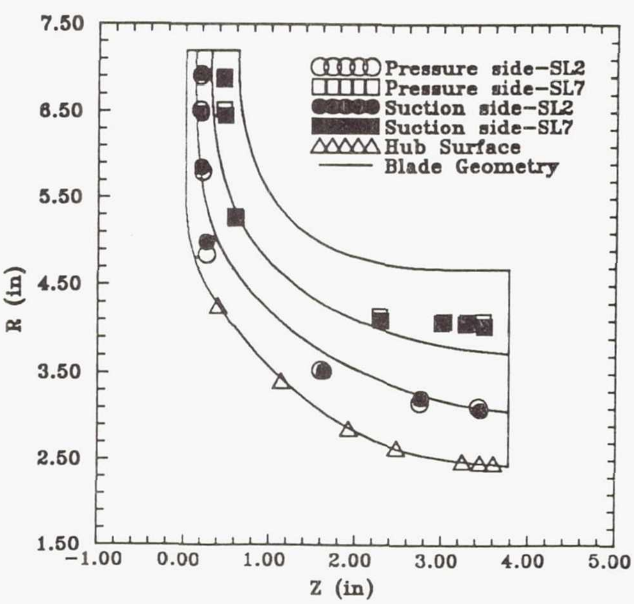

Figure 3. Rotor pressure tap locations

A rotating Scanivalve system was used to record the 36 surface static pressures. With the exception of the port identification method, reference 7 describes the Scanivalve system. The port location, as the unit steps through its cycle, was identified by an output voltage that varied linearly between known values from the first to the last port. Reference 7 also describes the centrifugal pumping correction,

$$P_{r,corr} = P_{r,i} \exp\left[\frac{\omega^2 (r_i^2)}{2RTg_c}\right]$$

$P_{r,i}$ = measured static pressure at port i, psi
r_i = radial distance from the shaft centerline, ft.
ω = rotational speed, radians per second
R = universal gas constant
g_c = acceleration due to gravity
T = averaged total inlet & exit temperature, R

employed in the data reduction program. The corrected surface static pressures were ratioed to the averaged total inlet pressure obtained from the rakes at station 0.

Analytical Method

The analysis method used to compare with the experimental data incorporates the coupling of the three codes, MERIDL, TSONIC, and BLAYER as described by Boyle, Haas, and Katsanis in reference 8. The coupled codes, MTSB, allowed for the prediction of overall losses in conjuction with the aerodynamic analysis. MTSB has been used to predict axial turbine performance, however, the radial turbine provides new and different challenges in the loss prediction. With access to blade surface static pressure data, the capability of surface pressure prediction can be evaluated as a first step in the evaluation of the loss model accuracy.

References 8 and 9 provided background information on the use of MTSB in calculating the overall stage performance. References 10-13 provided additional information on the use of

MTSB. References 10-12, the user's mannuals for MERIDL, TSONIC, and BLAYER respectively, provided information on preparing the input and explaining the output for each of the codes. Reference 13 details the algebraic loss correlation that was recently added to MTSB for the radial turbine case. Because of the unusually large backface clearance and the significant tip clearances for this research rotor, the loss coefficient calculated as per Ref. 13 was quite large and was dominated by the backface value.

The coupled programs operate individually starting with MERIDL. MERIDL is an inviscid 2D flow code that calculates the flow properties on the hub-to-shroud mid-channel stream surface. It includes an assumed pressure drop due to losses. TSONIC uses the stream sheet thicknesses generated by MERIDL to solve for flow conditions on blade-to-blade stream surfaces at various locations from hub to tip. Iteration between the two programs produces a solution with equal static pressures on both pressure and suction sides of the blade at the trailing edge. An integral method boundary layer code, BLAYER, can then use the resulting quasi-3D solution as input. BLAYER calculates the boundary layer growth along all four flow channel surfaces.

Using the same inlet conditions of temperature and pressure as the experimental portion of this study, MTSB solutions were obtained for pressure ratios of 2.5:1, 3:1, 4:1, 4.5:1 and 5:1 with the rotor operating at design speed.

Comparison and Discussion of Results

Stage Efficiency: Figure 4 compares the predicted total to total stage efficiencies with experimental values. The figure shows that at the lower pressure ratios MTSB predicts the stage efficiency very well with a variation of less than half a point. Larger variations between the prediction and experiment occur at the higher pressure ratios. Results at a pressure ratio of 5.5:1 show a maximum variation of 1.5 points. Although the maximum variation of efficiency is within a reasonable range, the unusually large backface clearance was the largest single loss affecting the overall efficiency.

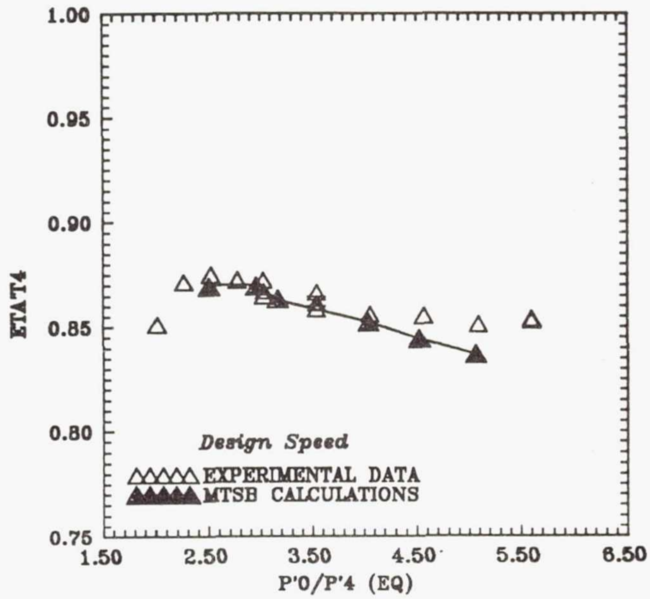

Figure 4. Total to total stage efficiencies

Figure 5 compares a breakdown of the predicted losses at design operation with the overall measured loss. The calculated loss breakdown shows that the clearance loss accounts for 71 percent of the rotor losses or 61 percent of the total stage loss. Figure 6 shows a breakdown of the clearance losses. The backface clearance loss for this rotor accounts for nearly 79

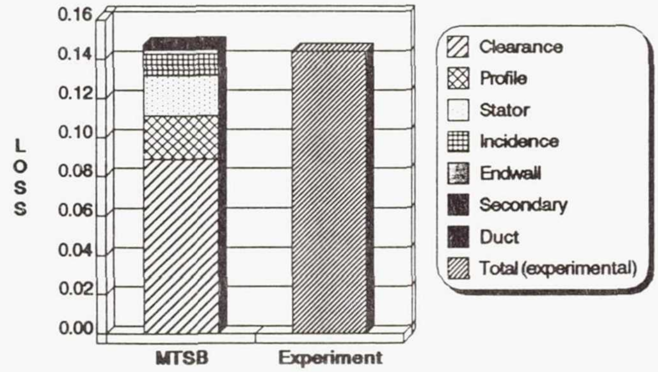

Figure 5. MTSB loss breakdown compared with total measured loss

percent of total clearance losses. Because of the 3D viscous nature of tip and backface leakage, the quasi-3D approach used by MTSB allows only a conservative estimate of the clearance losses. Figure 5 and 6 show that without capturing a detailed picture of the leakage effects MTSB estimates the overall losses within a tolerable range.

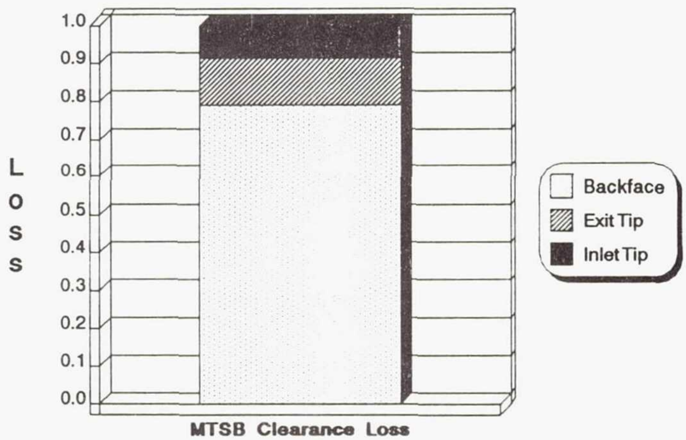

Figure 6. Clearance loss breakdown

<u>Mass Flow</u>: Figure 7 shows the equivalent mass flow as a function of pressure ratio. At the minimum and maximum pressure ratios, the input values of mass flow to MTSB yielded the same pressure ratios as in the experimental case. The two curves approach the same choked value, but the curvature of the two vary slightly. MTSB required a decrease in mass flow to achieve a converged solution at the intermediate pressure ratios. Overall, the calculated mass flow agrees well with the measured mass flow with a maximum difference of 1 percent.

<u>Surface Pressure Comparison</u>: The predicted surface static pressures were compared with the experimental values for both the stator and the rotor at the design pressure ratio of 4:1 . MTSB was used to calculate three streamlines for the stator. Since the stator cross-section is symmetric

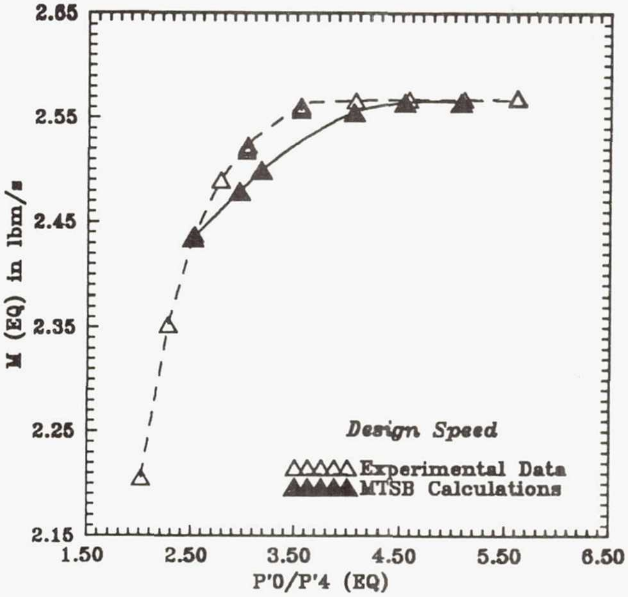

Figure 7. Mass flow at various pressure ratios

along the z-axis, results for only one streamline are shown. Figure 8 shows the predicted and measured surface static-to-inlet total pressure ratio as a function of relative radius. The relative radius is the actual radius, in feet, subtracted from unity. The spiked region on the suction surface was not expected. The MTSB solution indicated the spike on the suction side near the throat region. As seen in Fig. 8, the measured values show that the spike was indeed there, occurring in the throat region. By plotting the experimental stator data for all

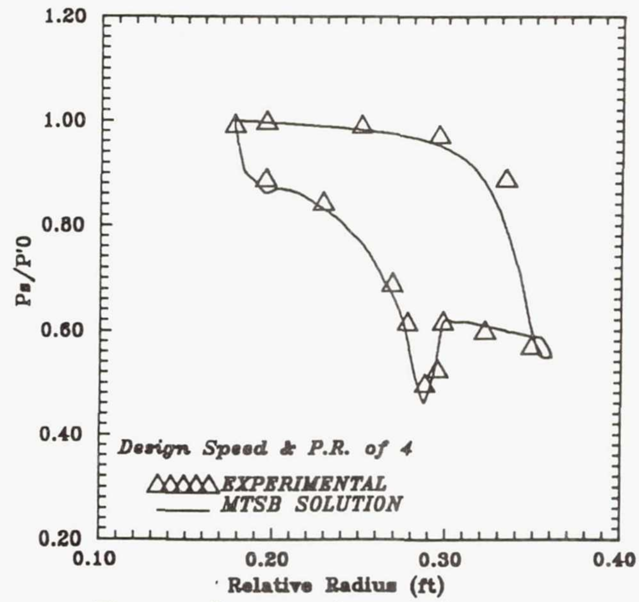

Figure 8. Stator surface static pressure comparison

tested pressure ratios, the nature of the spike could be determined. Figure 9 shows that some form of the spike occurs at all pressure ratios including a pressure ratio of 2:1, where the flow is subsonic throughout the stage. This figure indicates that the spike is not a shock region, but instead the result of a surface curvature inflection near the throat. MTSB predicted the spike accurately over the range of pressure ratios even though its flow solution methodology is less accurate for Mach numbers above unity.

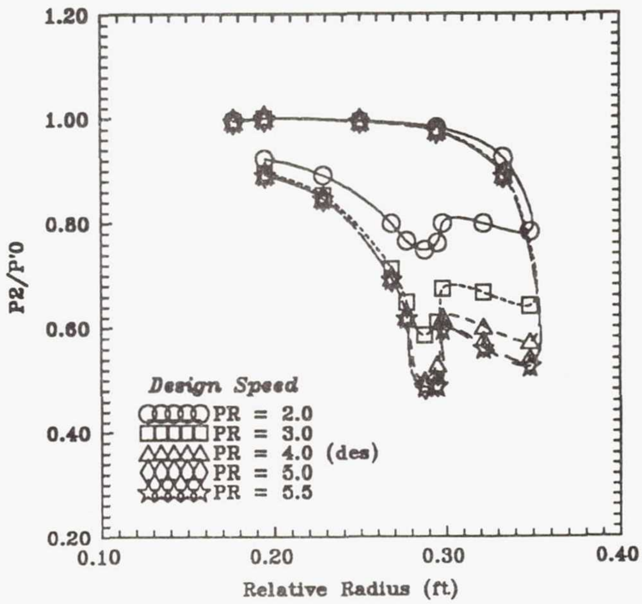

Figure 9. Measured stator surface pressures

MTSB was also used to calculate four streamline solutions for the rotor. The results correspond to streamlines near the pressure tap locations. Streamlines 1 and 4 follow the hub and tip boundaries respectively. Streamline 2 is for a flow function of 20 percent and streamline 3 is for a flow function of 70 percent. Figure 10 compares the measured and the calculated relative static-to-inlet-total pressure ratio versus distance along the blade at the 20 percent streamline. Figure 11 illustrates those values at the 70 percent streamline. The two figures show that MTSB over-predicts the surface pressures at the inlet on the pressure side and that the variation is larger on the 20 percent streamline than at the 70 percent streamline. Proving that these results were not due to the fact that the taps do not fall on a specific streamline, figures 12 and 13 display contours generated by using MTSB data from all four streamlines. Overlaying the measured pressure values provides another perspective of the comparison. The contours also illustrate the over-prediction of surface pressure at the inlet on the pressure side of the blade.

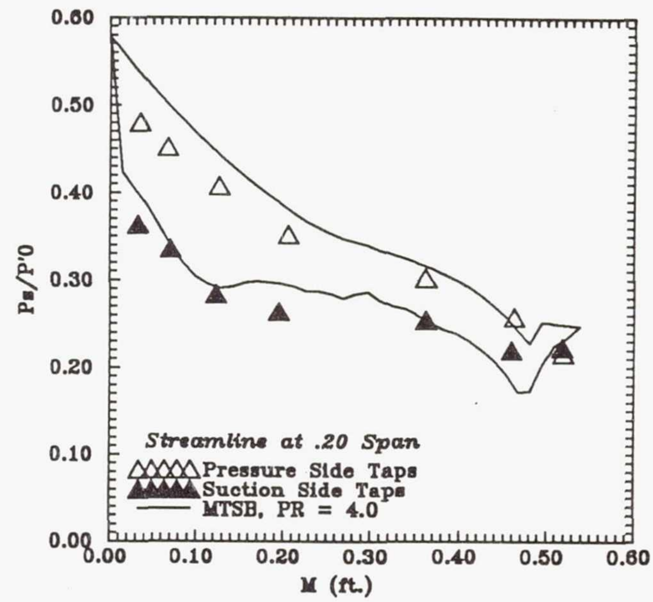

Figure 10. Rotor surface static pressures at 20% streamline

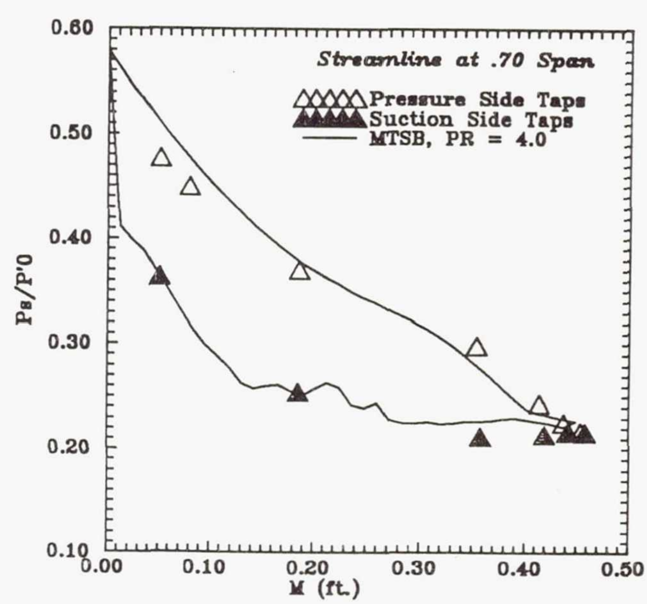

Figure 11. Rotor surface static pressures at 70% streamline

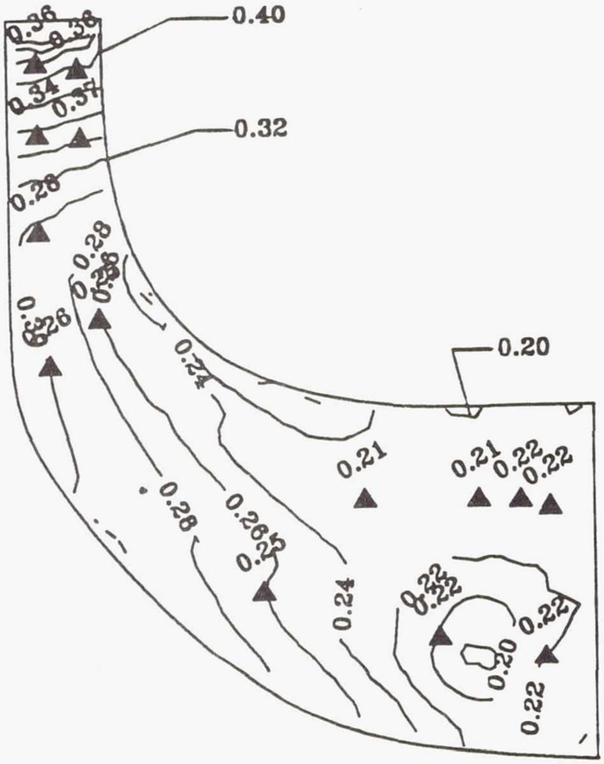

Figure 12. Rotor surface pressure contour, suction side

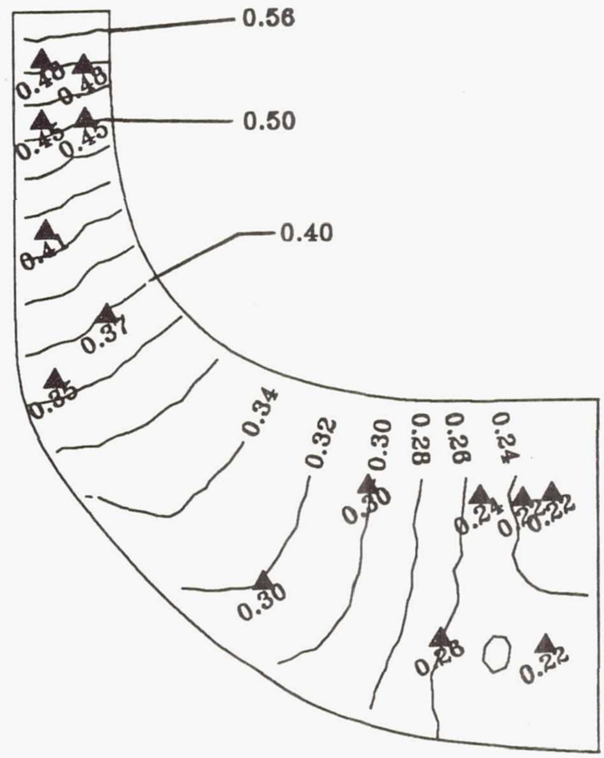

Figure 13. Rotor surface pressure contour, pressure side

As shown in figure 3 the first four pressure taps are located before or at the scalloped backface. The first four experimental static values in Fig. 10 show the greatest variation in the comparisons. The leakage from the pressure side of the blade to the suction side due to the scalloped backface and the large backface clearance is the most probable reason for the discrepancy between the calculated and experimental values. Looking again at the solution comparison at the 70 percent streamline, figure 11, the agreement between the two is good for an inviscid flow calculation. The good agreement in this figure indicates that leakage may indeed be occuring and that its affect is not as severe toward the shroud region.

Conclusion

With the newer loss model modifications, MTSB provides a more accurate aerodynamic analysis for the radial turbine. The quasi-3D inviscid flow analysis method predicted the stage efficiency within 1.5 points. This prediction is remarkably good considering the unusually large backface clearance that dominated the overall loss. The large backface clearance may have caused the variations between the calculated and measured surface pressure data at the rotor inlet. The streamline and pressure contour comparisons indicate that the clearance loss is indeed a factor in prediction accuracy. However, the accurate results produced by the stator comparison provide motivation for the continued use of MTSB as tool for obtaining an aerodynamic analysis of the radial turbine configuration. Experimental work on the effect of tip and backface gap size on the overall loss would yield increased loss model accuracy. Keeping in mind that MTSB is an inviscid code, one can see that it is already an extremely useful tool that is continuously improving.

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