Performance Degradation due to Blade Surface Roughness in a Single-Stage Axial Turbine

Turbine blades experience significant surface degradation with service. Previous studies indicate that an order-of-magnitude or greater increase in roughness height is typical, and these elevated levels of surface roughness significantly influence turbine efficiency and heat transfer. This paper presents measurement and a mean-line analysis of turbine efficiency reduction due to blade surface roughness. Performance tests have been conducted in a low-speed, single-stage, axial flow turbine with roughened blades. Sheets of sandpaper with equivalent sandgrain roughnesses of 106 and 400 μm have been used to roughen the blades. The roughness heights correspond to foreign deposits on real turbine blades measured by Bons et al. [1]. In the transitionally rough regime (106 μm), normalized efficiency decreases by approximately 4% with either roughened stator or roughened rotor and by 8% with roughness on both the stator and rotor blades. In the fully rough regime (400 μm), normalized efficiency decreases by 2% with roughness on the pressure side and by 6% with roughness on the suction side. Also, the normalized efficiency decreases by 11% with roughness only on stator vanes, 8% with roughness only on rotor blades, and 19% with roughness on both the stator and rotor blades.

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Introduction

Turbine blades experience surface degradation with use, and an order-of-magnitude or greater increase in root-mean-square (rms) roughness is typical for the first stage vanes in high-pressure turbines [1–3]. Much effort has been expended to characterize the randomly distributed nature of turbine blade surface roughness and to translate their characterization into more convenient, deterministic forms (e.g., distributed cylinders, hemispherical segments, and equivalent sandgrain roughness). Taylor [2] used a stylus profilometer to measure the first-stage turbine blade roughness from two military engines. He measured several statistical quantities, such as rms, skewness, and kurtosis of the roughness height distribution, and found great variation from point to point on the blade. Bogard et al. [3] made profilometer measurements of two sample turbine vanes from military engines. Bons et al. [1] measured a stylus profilometer to measure the first-stage turbine blade roughness from two military engines. He measured several statistical quantities, such as rms, skewness, and kurtosis of the roughness height distribution, and found great variation from point to point on the blade. Bogard et al. [3] made profilometer measurements of two sample turbine vanes from military engines. Bons et al. [1] made measurements, including the centerline averaged roughness $R_a$, maximum peak-to-valley roughness $R_N$, and roughness shape/density parameter $L_s$, in many different turbines. Bons et al. [1] also categorized turbine blade surface roughnesses according to their causes—foreign deposits, pitting/erosion, and thermal barrier coating spallation. Each roughness-producing mechanism produced unique surface textures. For example, pitting and spallation had large roughness recesses below the surface mean line while deposits were characterized by peaks above the mean line.

However, since two surfaces with the same roughness height can have different aerodynamic characteristics, roughness height alone is insufficient to characterize surface roughness. Sigal and Danberg [4] have indicated that equivalent sandgrain roughness is sensitive to not only the shape, but also spacing of roughness elements. Therefore, they measured roughness shape/density parameter $L_s$ as well as actual roughness $k_s$ and correlated them with equivalent sandgrain roughness $k_s$. Bogard et al. [3] estimated equivalent sandgrain roughness height for their measured vane surface roughness by using Sigal and Danberg’s correlation. Also, to convert roughness $k$ into equivalent sandgrain roughness $k_s$, Bons [5] measured $c_f$ values for scaled replicas of “real” turbine surface roughness $k$ in a wind tunnel. Then, he used Sigal and Danberg’s correlation to calculate equivalent sandgrain roughness $k_s$. Bons [5] found that $c_f$ values for scaled replicas of “real” rough surfaces, and this conclusion justified previous studies that used uniform emery grains [7,8] and sand grains [9] to simulate rough surfaces in cascades and turbines. Bons [5] also attempted correlations of a number of roughness parameters and recommended a modified form of Sigal and Danberg’s correlation.

Studies have been conducted to measure performance penalty due to surface roughness in turbines. Bammert and Sandstede [8] measured boundary layer parameters (momentum and displacement thicknesses) and static pressure distribution on vane surfaces in a cascade. Roughness on the blade surface caused a premature transition from laminar to turbulent boundary layers, and the roughness on the suction side increased momentum thickness two to three times more than that on the pressure side. Kind et al. [9] measured profile loss and deviation angle for flows through roughened cascade vanes. The blade-loading and deviation angle were not sensitive to surface roughness. The profile loss increased drastically due to the suction-side roughness while there was not much change due to the pressure-side roughness, reconfirming the results of Bammert and Sandstede [8]. Boyle and Senyitko [10] measured total pressure loss in a linear cascade for a wide range of Reynolds number values and proposed a roughness transition model to match their experimental results. Bammert and Sandstede [7] found that efficiency decreased by up to 14% in a four-stage turbine with all blades uniformly roughened with 580 μm emery grains. Boynton et al. [11] found that a decrease in rotor blade surface roughness from 10.16 μm (rough) to 0.76 μm (polished) increased total-to-total efficiency by 2.4% in a double-stage high-pressure fuel turbopump turbine.

Nevertheless, the influence of roughness corresponding to the "real" roughness [1] on turbine performance has not yet been investigated. In addition, the effects of surface roughness location on performance in rotating turbine rigs have not yet been studied. To the authors' knowledge, only the cascade experiments performed by Bammert and Sandstede [8] and Kind et al. [9] explored suction-pressure side differentiation. Therefore, this study has been conducted to measure, for the first time, the relative influence of surface roughness location—suction side versus pressure side and stator versus rotor—on turbine performance in a rotating environment.

Specifically, this paper addresses the following questions.

- How does equivalent sandgrain roughness $k_s$ corresponding to the "real" blade surface roughness affect turbine performance?
- What are the relative effects of suction- and pressure-side roughness in a rotating environment?
- Is turbine performance more sensitive to stator roughness or rotor roughness?

Experimental Facility

Flow Loop. The open-loop experimental facility consists of a blower, flow conditioning section, test section, and exit. The overall view of the test rig is shown in Fig. 1. A 75 kW main blower provides a mass flow rate of 3.1 kg/s to the test section. Upstream of the test section, the flow passes through a mass flow meter and a flow conditioning section (honeycomb and screens) before entering the test section.

Instrumentation. Mass flow rate is determined upstream of the turbine from the pressure difference between upstream and downstream of a bank of nine calibrated venturi nozzles using pressure transducers with an accuracy of $\pm 0.25\%$ of the full scale values. The overall accuracy of the mass flow meter is $\pm 1\%$ of the measured value. The axial locations of aerodynamic instrumentation stations are shown in Fig. 2. The axial distance $x$ is measured from the rotor leading edge and is positive in the downstream direction. The distances are normalized by the axial chord measured from the rotor leading edge and is positive in the downstream direction. The distances are normalized by the axial chord measured from the rotor leading edge and is positive in the downstream direction.

Test Turbine. The test turbine geometry is given in Table 1. The maximum rotating speed is 1600 rpm, and the maximum shaft power is 15 kW. The degree of reaction of this turbine is approximately 30%. At the design point, the inlet flow velocity is 20 m/s and the Reynolds number is $2.0 \times 10^5$ based on rotor chord length and relative exit velocity.

Surface Preparation. Several types of turbine blade surface roughness (e.g., foreign deposits, pitting/erosion) can be represented by equivalent sandgrain roughness $k_s$. However, foreign deposits affect flow in two ways—first, by increasing surface roughness and second, by reducing the blade passage area. On the other hand, pitting/erosion increases surface roughness only. This study’s scope is limited to the effects of foreign deposits because foreign deposits are known to be more important in determining turbine efficiency [1]. For current investigation, the surface data corresponding to foreign deposits on the suction side, midspan, and leading edge part have been chosen (Table 2).

"Real" roughness height cannot be directly matched with densely distributed equivalent sandgrain roughness. Previous studies have used a vague concept of the ratio of sand grain size to chord length to determine the roughness height (e.g., [7]). However, same roughness has different effects in machines operating at different Reynolds numbers. Therefore, the absolute value of roughness height alone is insufficient.

To overcome this problem, Bons [5] recommended the following correlation to estimate equivalent sandgrain roughness, taking into account the spacing of roughness elements.

<table>
<thead>
<tr>
<th>Table 1 Test turbine geometry</th>
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<tbody>
<tr>
<td>Parameter</td>
</tr>
<tr>
<td>Axial chord (mm)</td>
</tr>
<tr>
<td>Hub diameter (mm)</td>
</tr>
<tr>
<td>Tip diameter (mm)</td>
</tr>
<tr>
<td>Number of vanes/blades</td>
</tr>
<tr>
<td>Tip clearance (mm)</td>
</tr>
<tr>
<td>Annulus area (m²)</td>
</tr>
<tr>
<td>Solidity</td>
</tr>
<tr>
<td>Inlet angle (deg)</td>
</tr>
<tr>
<td>Exit angle (deg)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 2 Representative turbine blade surface data (Bons et al. [1])</th>
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<tbody>
<tr>
<td>Region</td>
</tr>
<tr>
<td>SS/MS/LE</td>
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</tbody>
</table>
where, according to Sigal and Danberg [4],
\[ \Lambda = \left( \frac{A_f}{A_s} \right)^{-1.6} \]  
(2)

The average roughness height, \( k \) in Eq. (1), is commonly estimated to be \( R_z \), or the average of the local \( R_s \) values over the entire map [5]. According to Bons [5], \( R_z \) is approximately five times the centerline average roughness \( R_a \). Therefore, for the case in Table 2,

\[ R_z = k = 70.5 \ \mu m \]  
(3)

Substituting \( k = 70.5 \ \mu m \) and \( \Lambda_s = 45 \) into Eq. (1), \( k_s = 90.64 \ \mu m \) for the Bons et al. [1] case. At this Reynolds number, the roughness number \( k^+ \) can be determined as,

\[ k^+ = Re \frac{k_s}{c} \sqrt{c_f} = 149.2 \]  
(4)

where \( c_f = \left[ 2.87 + 1.58 \log \left( \frac{c}{k_s} \right) \right]^{-2.5} \)  
(Ref. [6])

To roughen the blades, sheets of sandpaper are attached to blade surfaces. The sandpaper is composed of 500 \( \mu m \) thick base paper and sand grits glued to the base paper. The effects of surface roughness have been isolated as follows. For the baseline case, the blades are wrapped with two sheets of double-sided tape with each sheet having the same thickness as the base paper of the sandpaper (Fig. 3). For rough blades, only one sheet of double-sided tape is attached to the blades, and sandpaper is attached to the double-sided tape. For roughness only on the suction side, one sheet of double-sided tape is attached to the blades, sandpaper is attached to the tape on the suction side, and another layer of double-sided tape is added on the pressure side. The roughness

![Fig. 3 Blade roughening process](image)

![Fig. 4 Roughened test turbine blade set](image)

Table 3 Test matrix

<table>
<thead>
<tr>
<th></th>
<th>Baseline Smooth</th>
<th>Stator SS Smooth</th>
<th>Stator PS Smooth</th>
<th>Rotor SS Smooth</th>
<th>Rotor PS Smooth</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fully rough</td>
<td>Stator SS Fully rough</td>
<td>Stator PS Fully rough</td>
<td>Rotor SS Fully rough</td>
<td>Rotor PS Fully rough</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Stator PS Fully rough</td>
<td>Stator PS Fully rough</td>
<td>Rotor PS Fully rough</td>
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<td></td>
<td>Stator SS Smooth</td>
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<td>Rotor SS Smooth</td>
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The chord length \( c \) is inferred to be 70 mm from the Bons [5] data. According to Nikuradse [13], \( k^+ = 149.2 \) is sufficiently large to be in the fully rough regime \( (k^+ > 70) \).

For the test facility used in this study, the blade Reynolds number is \( 2.0 \times 10^5 \) and the true chord length of rotor blades is 60 mm. Therefore, to match \( k^+ = 149.2 \), the equivalent sandgrain roughness height for the fully rough test case is determined to be \( k_s = 400 \ \mu m \). For intermediate levels of roughness \( (5 < k^+ < 70) \), \( k^+ = 30 \) has been selected. Assuming that Eq. (5) is valid in the transitionally rough regime as well, the corresponding \( k_s \) is 106 \( \mu m \).
starts from the design stagnation point of the smooth blade. Thus, foreign deposit effects are introduced, and a roughened blade row is shown in Fig. 4.

Table 3 shows the test matrix to differentiate the effects of transitional ($k_s = 106 \text{ mm}$) and full ($k_s = 400 \text{ mm}$) roughnesses. The stator has been chosen for suction-pressure side differentiation because the roughened stator causes more performance penalty than the roughened rotor.

The normalized efficiency decreases by 8% when both the stator and rotor blade rows are roughened with sandpaper of $k_s = 106 \text{ mm}$ and by 19% with sandpaper of $k_s = 400 \text{ mm}$. Thus, there is more of an efficiency penalty with larger equivalent sandgrain roughness $k_s$. Furthermore, the flow coefficient corresponding to maximum efficiency decreases with increasing roughness.

Previously, Bammert and Sandstede [7] made measurements in a four-stage turbine with all blades uniformly roughened with the same size emery grains. Their Reynolds number based on the blade chord and exit velocity varied between $3.5 \times 10^5$ and $6.0 \times 10^5$, compared to $2.0 \times 10^5$ in the current study. Therefore, according to Eqs. (4) and (5), $k_s = 106 \text{ mm}$ in the current study corresponds to $k_s = 60 \text{ mm}$ in Bammert and Sandstede’s turbine. As listed in Table 4, the new test results are comparable to those of Bammert and Sandstede [7]. Differences may be due to multistage effects, which are absent in the current study.

Figure 6 shows the results of stator suction side-pressure side differentiation in the fully rough regime. The normalized efficiency decreases by 2% when the stator’s pressure side is roughened and by 6% when the stator’s suction side is roughened. Thus, turbine performance is more sensitive to roughness on the stator suction side than that on the stator pressure side. This result is similar to the trend found in cascades by Bammert and Sandstede [8] and Kind et al. [9]. They also found that the roughened suction side losses were about three times as large as those due to roughened pressure side.

Figure 7 shows stator-rotor differentiation results in the transitionally rough regime. The normalized efficiency decreases by approximately 4% with either roughened stator or roughened ro-

![Fig. 5 Normalized efficiency degradation with transitional and full roughnesses](image)

**Experimental Results and Discussion**

To evaluate surface roughness effects on turbine performance, total-to-static efficiency $\eta_{ts}$ has been measured at different values of flow coefficient $\phi$. The two parameters are defined as

$$\eta_{ts} = \frac{Q \omega}{mc_p T_i (1 - (P_{st}/P_t)^{1+\gamma})}$$

$$\phi = \frac{U_{st}}{U}$$

Repeatability tests have been conducted for each flow coefficient, and efficiency has been found to be repeatable to within 1%. This variation is less than the 1.4% uncertainty (95% confidence interval) in $\eta_{ts}$ estimated according to the method of Kline and McClintock [14]. The turbine performance is represented by normalized efficiency $\eta/\eta_0$ plotted versus normalized flow coefficient $\phi/\phi_0$, where $\eta_0$ and $\phi_0$ are the design efficiency and design flow coefficient of the baseline case, respectively.

Figure 5 shows the plots of normalized efficiency versus flow coefficient for the baseline, transitionally rough, and fully rough cases. As expected, blade surface roughness significantly degrades turbine efficiency. The normalized total-to-static efficiency decreases by 8% when both the stator and rotor blade rows are roughened with sandpaper of $k_s = 106 \text{ mm}$ and by 19% with sandpaper of $k_s = 400 \text{ mm}$. Thus, there is more of an efficiency penalty with larger equivalent sandgrain roughness $k_s$. Furthermore, the flow coefficient corresponding to maximum efficiency decreases with increasing roughness.

![Fig. 6 Suction side-pressure side differentiation with fully roughened stator ($k_s = 400 \text{ mm}$)](image)

![Fig. 7 Stator-rotor differentiation with transitional roughness ($k_s = 106 \text{ mm}$)](image)
tor, and 8% with roughness on both the stator and rotor. In this case, the relative influence of stator or rotor roughness cannot be resolved because the data in both cases are within each other’s uncertainty range.

However, similar data with full roughness clarify this issue. Figure 8 shows stator-rotor differentiation results in the fully rough regime ($k_s = 400 \mu m$). The normalized efficiency decreases by 11% when stator is roughened and by 8% when rotor is roughened. When both are roughened, the efficiency decreases by 19%.

Thus, compared to rotor roughness, stator roughness results in an additional 3% efficiency deterioration. This is significant because the stator of the first stage of turbines may be the most vulnerable to surface degradations. Also, the efficiency drop with a roughened stator added to the drop with a roughened rotor is approximately equivalent to the efficiency drop with both rows roughened. Furthermore, the decrease in flow coefficient corresponding to maximum efficiency also occurs here as each row is roughened.

Recently, Boynton et al. [11] measured total-to-total efficiency in a double-stage turbine. Their Reynolds number was $1.5 \times 10^6$, and they found that the total-to-total efficiency decreased by 2.4% when rotor roughness was increased from 0.76 $\mu m$ rms to 10.16 $\mu m$ rms. In the current study, total-to-static efficiency decreased by 4% and 8% with transitionally rough and fully rough rotor blades, respectively (Table 5). The difference can be due to several reasons. First, Boynton et al. measured total-to-total efficiency versus the total-to-static efficiency measured in the current study. Second, Boynton et al. did not report roughness shape/density parameter. Therefore, $k_s$ and $k^*$ have been estimated for the Boynton et al. turbine in Table 5 by using a possible range of roughness shape/density parameter values from the Bons et al. [1] measurements. Third, multistage effects are absent in the current study. Therefore, a direct comparison is difficult.

The efficiency degradation and decrease in flow coefficient corresponding to maximum efficiency due to blade surface roughness can be explained through a mean-line analysis as follows. Figure 9 shows time-averaged stator pressure coefficient (the ratio of static pressure drop across the stator and the stator inlet dynamic head $0.5pu_{11}^2$) normalized by its value at the design point of the baseline case. These results have been obtained from the casing static pressure data. The stator pressure coefficient shows a negligible change due to roughness on the rotor blades. However, roughness on the stator vanes increases the stator pressure coefficient. Based on the velocity triangles in Fig. 10, the following equation can be obtained for the stator:

$$P_{s1} + \frac{1}{2}P_{u1}^2 = P_{s2} + \frac{1}{2}P_{u2}^2 + \Delta P_s^S$$

(8)

Here, $\Delta P_s^S$ is the total pressure loss across the stator, and because the current focus is on steady flow properties, total pressure change indicates loss. Equation (8) can then be non-dimensionalized as follows:

$$C_{ps} = \frac{P_{s1} - P_{s2}}{\frac{1}{2}P_{u1}^2} = \frac{1}{\cos \alpha_2} - 1 + \xi^S$$

(9)

Kind et al. [9] have already shown that roughness does not significantly alter flow angles. Thus, the first two terms on the right-hand side of Eq. (9) remain constant, and, consequently, the increase in stator pressure coefficient due to stator roughness (Fig. 9) must result from the increase in stator total pressure loss. Furthermore, Figure 11 shows graphs of estimated stator loss coefficient $\xi^S$, using Eq. (9) and the data from Fig. 9. Here, the stator loss coefficient is clearly independent of flow coefficient.

Figure 12 shows graphs of rotor pressure coefficient (the ratio of static pressure drop across the rotor and the stator inlet dynamic head $0.5pu_{11}^2$) normalized by its value at the design point of the baseline case. The stator pressure coefficient shows a negligible change due to roughness on the rotor blades. However, roughness on the rotor vanes increases the stator pressure coefficient. Based on the velocity triangles in Fig. 10, the following equation can be obtained for the rotor:

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(8)

Here, $\Delta P_s^S$ is the total pressure loss across the stator, and because the current focus is on steady flow properties, total pressure change indicates loss. Equation (8) can then be non-dimensionalized as follows:

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$$P_{r1} + \frac{1}{2}P_{u1}^2 = P_{r2} + \frac{1}{2}P_{u2}^2 + \Delta P_r^S$$

(8)

Here, $\Delta P_r^S$ is the total pressure loss across the rotor, and because the current focus is on steady flow properties, total pressure change indicates loss. Equation (8) can then be non-dimensionalized as follows:

$$C_{pr} = \frac{P_{r1} - P_{r2}}{\frac{1}{2}P_{u1}^2} = \frac{1}{\cos \alpha_2} - 1 + \xi^R$$

(9)

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Table 5 Current test results compared with data from Boynton et al. [11]

<table>
<thead>
<tr>
<th></th>
<th>$k_s$ (mm)</th>
<th>$k^*$</th>
<th>$\eta/\eta_0$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotor</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Roughened</td>
<td>Current</td>
<td>106</td>
<td>30</td>
</tr>
<tr>
<td></td>
<td>Boynton study</td>
<td>400</td>
<td>149</td>
</tr>
<tr>
<td></td>
<td>Kind et al.</td>
<td>13–72</td>
<td>40–280</td>
</tr>
</tbody>
</table>

Fig. 9 Measured stator pressure coefficient

Fig. 10 Stage velocity triangles
head \(0.5\rho v_{11}^2\) normalized by its value at the design point of the baseline case. Two differences from Fig. 10 stand out. First, the rotor pressure coefficient shows dependence on flow coefficient. Second, roughness on stator vanes increases the rotor pressure coefficient even when rotor blades themselves are not roughened. Since rothaly is conserved across the rotor and the blade’s rotational speed remains constant in axial turbomachines (i.e., \(U_2 = U_3\)), the following equation can be obtained for the rotor:

\[
C_{ps}^R = \frac{P_{s2} - P_{s3}}{1/\rho v_{11}^2 - \frac{1}{\cos^2 \beta_3}} - \frac{1}{\cos^2 \alpha_2} \frac{1}{\phi} + \frac{2 \tan \alpha}{\phi} + C_{\xi R}^R 
\]  
(10)

Again, according to Kind et al. [9], the first four terms on the right-hand side of Eq. (10) remain constant for given blade angles and flow coefficient. Therefore, the increase in the rotor pressure coefficient due to roughness on stator vanes (Fig. 12) must result from the increase in total pressure loss across the rotor blades. Thus, stator roughness increases rotor loss, and, consequently, stator roughness results in more efficiency deterioration than rotor roughness (Fig. 8).

Figure 13 shows graphs of calculated rotor loss coefficient \(\xi^R\) using Eq. (10) and the data from Fig. 12. Unlike the stator, the loss across the rotor is a function of flow coefficient because the rotor incidence angle varies with flow coefficient. Furthermore, the rotor loss coefficient increases more than the stator loss coefficient with roughness. The rotor loss coefficient can be approximated with a second-order polynomial, which can be used for efficiency estimation.

For incompressible flows, total-to-static efficiency can be shown to have the following form:

\[
\eta_{ts} = \frac{\rho U v_{t1}(\tan \alpha_3 + \tan \beta_3) - \rho U^2}{\rho U v_{t1}(\tan \alpha_3 + \tan \beta_3) - \rho U^2 + \frac{1}{2} \rho v_{11}^2 \Delta P_s^R + \Delta P_t^R}
\]
and, in terms of nondimensional parameters,

\[
\eta_{ts} = \frac{2(\tan \alpha_3 + \tan \beta_3) \phi - 2}{\left(\xi^S + \xi^R + \frac{1}{\cos^2 \beta_3}\right) \phi^2 + 2 \phi \tan \alpha_3 - 1}
\]
(12)

Blade angles and flow coefficient are known a priori. Stator loss coefficient \(\xi^S\) is given in Fig. 11, and rotor loss coefficient \(\xi^R\) is given in Fig. 13. Finally, Eq. (12) can be used to obtain the calculated efficiency graphs in Fig. 8.

In Fig. 8, estimated and measured efficiencies agree well with each other. The magnitudes of degradation are well captured in all cases. Stator roughness causing greater loss than rotor roughness is also captured. Also, for a given roughness, dependence on flow coefficient is well estimated. Finally, the decrease in the flow coefficient corresponding to maximum efficiency with increasing roughness is also predicted. Thus, combined with a simple mean-line analysis, efficiency data and static pressure data provide a consistent picture of the relative and cumulative effects of roughness on stator and rotor blades. According to this result, one can think of wall pressure data as indicators of performance degradation due to blade surface roughness.

Conclusions

Performance measurements have been carried out in a low speed, single-stage, axial turbine to assess the effects of blade surface roughness due to foreign deposits. New conclusions from this investigation are as follows:

1. Blade surface roughness severely degrades turbine efficiency. The normalized total-to-static efficiency decreases by 8% when both the stator and rotor blade rows are roughened with transitional roughness \((k^* = 30)\) and by 19% with full roughness \((k^* = 149.2)\). Thus, efficiency penalty increases with increasing roughness.

2. As in cascades, a rotating turbine’s performance is more sensitive to roughness on the suction side of stator vanes than that on the pressure side.
3. Roughness on stator vanes increases loss through the rotor even when the rotor blades are not roughened; however, roughened rotor does not affect loss across the stator. Thus, roughened stator induces a higher performance penalty than roughened rotor.

4. Efficiency drop with a roughened stator added to the drop with a roughened rotor is approximately equivalent to the efficiency drop with both rows roughened.

5. With increasing roughness, the flow coefficient corresponding to maximum efficiency decreases, and this trend is captured with a simple mean-line analysis.

6. According to a simple mean-line analysis, wall static pressure data can be indicators of performance degradation due to blade surface roughness.

Acknowledgments
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Nomenclature

\[ A_f = \text{windward frontal surface area of roughness elements} \]
\[ A_w = \text{windward wetted surface area of roughness elements} \]
\[ c = \text{true chord length} \]
\[ c_s = \text{axial chord length} \]
\[ c_f = \text{skin friction coefficient} \]
\[ c_p = \text{specific heat} \]
\[ C_p^{ps} = \text{pressure coefficient (} = \Delta P_t/(1/2 \rho V_{11}^2) \]
\[ k = \text{average roughness height (} \approx R_z \)
\[ k_s = \text{equivalent sandgrain roughness} \]
\[ k^* = \text{roughness Reynolds number} \]
\[ m = \text{mass flow rate} \]
\[ P_t = \text{total pressure} \]
\[ P_s = \text{static pressure} \]
\[ P_{ts} = \text{centerline average roughness} \]
\[ P_{r} = \text{maximum peak to valley roughness} \]
\[ P_{r_1} = \text{average peak to valley roughness} \]
\[ R_{ts} = \text{Reynolds number based on true chord length and exit velocity} \]
\[ S = \text{sample surface area without roughness} \]
\[ S_f = \text{total frontal surface area of sample} \]
\[ T_t = \text{total temperature} \]
\[ U = \text{blade rotating speed} \]
\[ v = \text{absolute velocity} \]
\[ v_s = \text{axial velocity} \]
\[ w = \text{relative velocity} \]
\[ x = \text{axial distance from rotor leading edge} \]

Greek

\[ \alpha = \text{absolute flow angle from the axial direction} \]
\[ \beta = \text{relative flow angle from the axial direction} \]
\[ \gamma = \text{ratio of specific heats} \]
\[ \Delta = \text{difference across the blade row} \]
\[ \phi = \text{flow coefficient} \]
\[ \eta_s = \text{total-to-static efficiency} \]
\[ A_s = \text{roughness shape/density parameter} \]
\[ \rho = \text{density} \]
\[ \omega = \text{angular velocity} \]
\[ \kappa = \text{total pressure loss coefficient (} = \Delta P_t/(1/2 \rho V_{11}^2) \]

Subscripts and Abbreviations

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<tr>
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Superscripts

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References