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Back, Seung Chul



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Effects of Reynolds Number and Surface Roughness Magnitude and Location on Compressor Cascade Performance

Seung Chul Back

Technology Development Center,
Samsung Engineering,
Suwon 443-823, Korea

Garth V. Hobson

Mechanical and Aerospace Engineering,
Naval Postgraduate School,
Monterey, CA 93943

Seung Jin Song

Mechanical and Aerospace Engineering
Seoul National University,
Seoul 151-744, Korea

Knox T. Millsaps

Mechanical and Aerospace Engineering,
Naval Postgraduate School,
Monterey, CA 93943

An experimental investigation has been conducted to characterize the influence of Reynolds number and surface roughness magnitude and location on compressor cascade performance. Flow field surveys have been conducted in a low-speed, linear compressor cascade. Pressure, velocity, and loss have been measured via a five-hole probe, pitot probe, and pressure taps on the blades. Four different roughness magnitudes, R_a values of 0.38 μm (polished), 1.70 μm (baseline), 2.03 μm (rough 1), and 2.89 μm (rough 2), have been tested. Furthermore, various roughness locations have been examined. In addition to the as manufactured (baseline) and entirely rough blade cases, blades with roughness covering the leading edge, pressure side, and 5%, 20%, 35%, 50%, and 100% of suction side from the leading edge have been studied. All of the tests have been carried out for Reynolds numbers ranging from 300,000 to 640,000. For Reynolds numbers under 500,000, the tested roughnesses do not significantly degrade compressor blade loading or loss. However, loss and blade loading become sensitive to roughness at Reynolds numbers above 550,000. Cascade performance is more sensitive to roughness on the suction side than pressure side. Furthermore, roughness on the aft 2/3 of suction side surface has a greater influence on loss. For a given roughness location, there exists a Reynolds number at which loss begins to significantly increase. Finally, increasing the roughness area on the suction surface from the leading edge reduces the Reynolds number at which the loss begins to increase. [DOI: 10.1115/1.4003821]

Introduction

Gas turbine performance, especially power output and efficiency, deteriorates during extended operation for many reasons, and therefore, continuous maintenance is needed to retain a high performance level. Such maintenance is critical to minimizing both operation costs and greenhouse gas emissions. Among many factors, the aerodynamic performance of gas turbines is significantly influenced by the condition of compressor and turbine blade surfaces. Therefore, the impact of blade surface roughness on performance has long been an important research issue.

Bammert and Woelk [1] measured performance degradation due to surface roughness in a three-stage axial compressor. For a roughness-to-chord ratio, k_s/c of 4.51e-3, they found a 30% static pressure ratio deterioration and a 15–20% volume flow rate decrease. Schaffler [2] experimentally and analytically investigated boundary layer characteristics in roughened axial flow compressors. Mal'tsev and Shakhov [3] found increased deviation in a roughened compressor cascade for roughness magnitudes found in helicopter gas turbine engines.

More recently, measurement of real gas turbine blade roughness magnitudes and locations, along with experimental determination and analytical prediction of performance degradation due to roughness have been conducted. Taylor [4] measured surface roughness at various locations in gas turbine vanes. Bons et al. [5] also measured the roughnesses of nearly 100 turbine blades, found spatial variation, and categorized the roughness characteristics

according to their causes. Experimentally, Leipold et al. [6] measured loss and boundary layer parameters in a highly loaded transonic compressor cascade for Reynolds numbers between 300,000 and 1×10^6 . Schreiber et al. [7] reported performance dependence of a roughened controlled-diffusion airfoil (CDA) on Reynolds numbers ranging from 100,000 to 3×10^6 . Analytical models to predict performance deterioration of gas turbine compressors have been developed by many researchers including Millsaps et al. [8], Aker and Saravanamottoo [9], Massardo [10], Syverud and Bakken [11], and Song et al. [12]. More recently, Back et al. [13] measured the effects of roughness on loss and deviation in a low-speed compressor cascade with blades representative of modern industrial gas turbines.

In addition, even with uniform roughness, boundary layers on blade surfaces are affected by the local pressure gradient. Therefore, the aerodynamic performance will be affected by the location as well as the magnitude of roughness. In this light, Tay et al. [14] used particle image velocimetry to investigate turbulent flow characteristics on a rough plate with a favorable pressure gradient. They found that surface roughness increased the mass and momentum flux deficits. With a favorable pressure gradient, friction velocity and skin friction coefficient increased. Pailhas et al. [15] studied the effect of Reynolds number and adverse pressure gradients on two kinds of rough plates with a hot-wire.

However, studies of roughness location on actual gas turbine blades are less common, and such studies are focused on the performance degradation in rotating rigs. Bammert and Sandstede [16] measured boundary layers for roughened turbine blades and found that the momentum thickness on the suction side is considerably thicker than that on the pressure side. Zhang and Ligrani [17] measured loss in the wake with roughened turbine blades. They applied various types of roughnesses, two fixed sizes and one variable roughness on the pressure side. The total pressure loss and kinetic energy were found to increase with roughness. Also, the suction

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side was found to contribute mostly to wake thickening. Yun et al. [18] measured performance degradation in a single-stage axial turbine with roughened blades. The normalized efficiency decreased by 2% when the pressure side was roughened and by 6% when the suction side was roughened. Thus, the degradation of turbine efficiency was three times bigger with roughness on the suction side.

In compressors, Suder et al. [19] conducted a survey in a transonic compressor rotor with fully and partially roughened blades. Blades with roughness on the leading edge and fully roughened blades showed approximately equal degradation in pressure ratio. Thus, roughness on the leading edge contributed to most of the performance deterioration. Gbadebo et al. [20] conducted a numerical and experimental survey in a single-stage low-speed compressor with roughened controlled-diffusion airfoils. They focused on 3D separation and reported that roughness induced a large hub corner separation, and increased blockage and deviation. In turn, the increased blockage significantly reduced the stage stagnation pressure rise.

Despite such research, reliable experimental data on roughness effects in compressor cascades are still scarce. Especially, how roughness location affects aerodynamic performance and how the location effect varies with Reynolds number remain unknown. Therefore, this paper presents a systematic experimental investigation of the impact of roughness magnitude, location, and Reynolds number on blade loading and loss on a highly loaded blade in a low-speed compressor cascade. The specific research questions are as follows:

In a low-speed compressor cascade,

- (1) How do roughness magnitude and Reynolds number affect compressor cascade performance?
- (2) How does the chordwise location of roughness affect compressor cascade performance?
- (3) How are the roughness location influences affected by Reynolds number?

Experimental Setup

Test Facility. Experiments have been conducted in a low-speed cascade wind tunnel (LSCWT) at the Naval Postgraduate School (NPS). Flow generated from a 410 kW centrifugal fan blows upwards and exits via an exhaust vent. The freestream turbulence level at the inlet of the wind tunnel was measured to be between 3–3.5%. A test section with a 254 mm height and 1524 mm width has been mounted at the end of the wind tunnel. Upstream of the test section, 60 inlet guide vanes are mounted to guarantee a uniform inlet flow angle of 36.3 deg. A schematic of the test section is shown in Fig. 1. A total of ten blades were mounted in the test section, and a total of seven blades (Numbers 1–7) have been roughened.

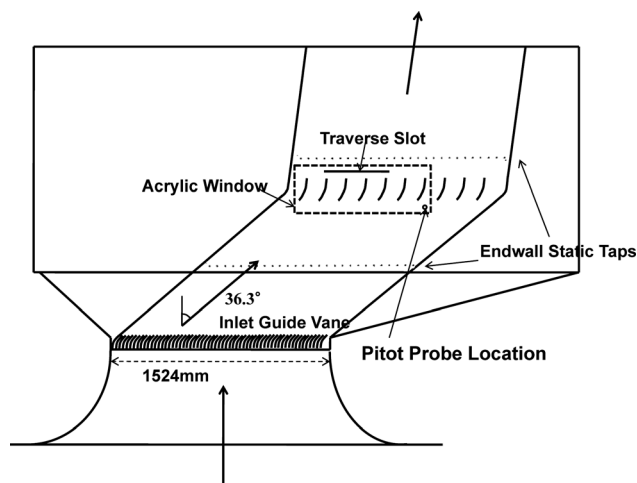


Fig. 1 Test section

Parameter	Value
Number of blades	10
Chord, c	127.3 mm
Span	254.0 mm
Blade spacing, S	152.4 mm
Solidity, c/S	0.835
Inlet flow angle, α_i	36.3 deg
Stagger angle, Γ	16.3 deg
Diffusion factor	0.52

The compressor blade profile used in this research was a second-generation controlled-diffusion airfoil designed by Gelder et al. [21], which had a design diffusion factor of the blade of 0.52. The chord length and span of the blades are 127.3 mm and 254 mm, respectively. The blades were placed 152.4 mm apart for a solidity of 0.835. Detailed geometry of the cascade is given in Table 1.

Instrumentation. Measurements have been carried out with static taps on blades, a pitot probe, and a five-hole probe with a probe diameter of 3 mm and a port size of 0.1 mm. The pitot probe has been used at 1.0 chord length upstream from the leading edge between the Blades 7 and 8 to obtain the upstream pressure and velocity. For blade loading, static pressure was measured via pressure taps at the midspan of Blade 5. There were 18 and 20 taps on the pressure and suction sides, respectively. Also, there were taps at the leading edge and trailing edge for a total of 40 taps. To avoid static pressure fluctuations due to the local roughness near the taps, the surface of the midspan region near the taps were kept smooth throughout. Finally, a cobra type five-hole probe was traversed at 0.2 chord length downstream from the trailing edge of Blade 3 to obtain total and static pressures, velocity, and flow angles. From the measurements, deviation and loss coefficient were calculated. Measurement locations are shown in Fig. 2.

Table 2 lists the tested blade roughness and coating thickness values. For each case, the centerline averaged roughness (R_a) was measured with a Mitutoyo SurfTest-401 profilometer. For the polished blade surface (polished), the centerline averaged roughness measured was 0.38 μm . In addition, roughness for the “as manufactured” blade surface (baseline) was measured to be 1.70 μm . To roughen the blades, paint or glue was sprayed onto the blades (Rough 1 and Rough 2). Thus, a total of four roughness values were tested as summarized in Table 2.

In addition, six different locations of roughness were tested at a surface roughness of 2.89 μm on the baseline blade. The six cases included roughness on the entire suction side, entire pressure side, leading edge only, and partially roughened suction side cases covering up to 20%, 35%, and 50% chordwise location from the

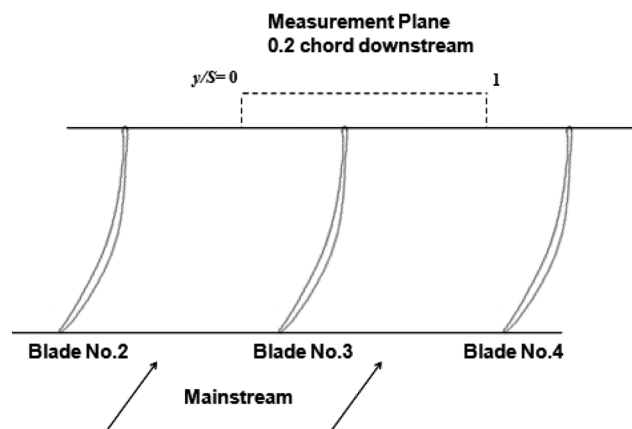


Fig. 2 Measurement locations

Table 2 Tested roughness and thickness

	Centerline averaged roughness (μm)	Std. dev	Thickness (mm)
Polished	0.38	0.109	–
Baseline	1.70	0.036	–
Rough 1	2.03	0.276	0.050–0.076
Rough 2	2.89	0.367	0.025–0.051

leading edge. Figure 3 and Table 3 show the tested roughness locations, and Fig. 4 shows a picture of blades with roughness on the suction surface between the leading edge and the 20% chord-wise location on the baseline blade.

Data Reduction and Uncertainty. Data were nondimensionalized as follows. First, on the blade static pressure P measurements were converted into pressure coefficient C_p defined below (Eq. (1)).

$$C_p = \frac{P - P_1}{P_{01} - P_1} \quad (1)$$

Here, P_1 and P_{01} indicate the upstream static and total pressure, respectively.

Loss coefficient is defined as the normalized difference between the upstream and downstream total pressures.

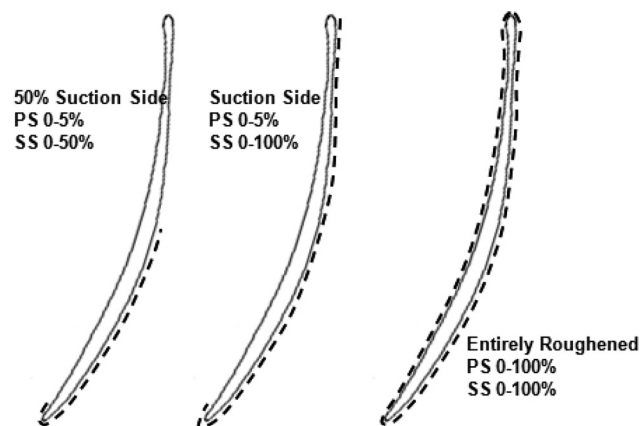
$$Y_p = \frac{P_{01} - P_0}{P_{01} - P_1} \quad (2)$$

where P_0 represents the downstream mass-averaged total pressure obtained via the five-hole probe. See Hansen [22] for more details on the loss measurement.

From the repeatability tests, static pressure coefficient and loss coefficient are repeatable to within 1% and 5%, respectively, for a given condition.

Results

Effects of Roughness Magnitude and Reynolds Number. Figure 5 shows the static pressure distribution for various roughness magnitudes at a Reynolds number of 600,000. A spanwise laminar separation bubble is visible from x/c of about 0.47 with reattachment at $x/c = 0.47$ on the suction side. Little change in loading is visible for R_a values of up to 2.03 μm . However, when the blades were roughened to $R_a = 2.89 \mu\text{m}$, (1) the suction side pressure is increased over most of the blade, significantly reducing the blade loading and hence turning; and (2) suc-

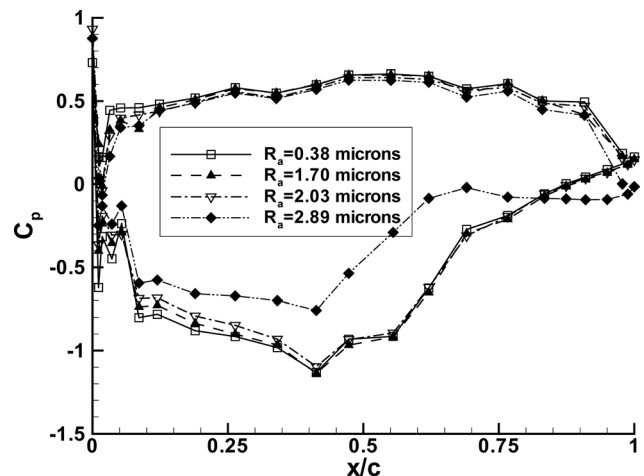
**Fig. 3 Tested roughness locations on the baseline blade****Table 3 Tested roughness locations on the suction side**

	Coverage on suction side	Coverage on pressure side
Leading edge	0–5%	0–5%
20% Suction side	0–20%	0–5%
35% Suction side	0–35%	0–5%
50% Suction side	0–50%	0–5%
100% Suction side	0–100%	0–5%

**Fig. 4 Picture of roughened blades (20% suction side of the baseline blade)**

tion side C_p does not change much between x/c of 0.7 and 1.0, indicating large turbulent separation, with a lower overall static pressure recovery. This turbulent separation increased the loss at a Reynolds number of 600,000 between R_a of 2.03 and 2.89 μm . The two trends found in this highly loaded, low-speed cascade are qualitatively similar to those found in a highly loaded, transonic compressor cascade by Leipold et al. [6]. For a Reynolds number of 600,000, a critical roughness (at which blade loading decreased significantly or large turbulent separation occurs) clearly exists between R_a of 2.03 and 2.89 μm . Furthermore, for the polished surface, increased blade loading due to decreasing C_p on the suction side between x/c of 0.10 and 0.25 can be seen.

Figure 6 shows the loss coefficient plotted versus blade pitch, y/S . Roughness has little impact on loss coefficient distribution for R_a up to 2.03 μm . High loss (or wake) region had a width of approximately 10% of the pitch with a maximum loss coefficient value of 0.4. However, for R_a of 2.89 μm , the wake's width

**Fig. 5 Blade loading for R_a of 0.38, 1.70, 2.03, and 2.89 μm at Reynolds number 600,000**

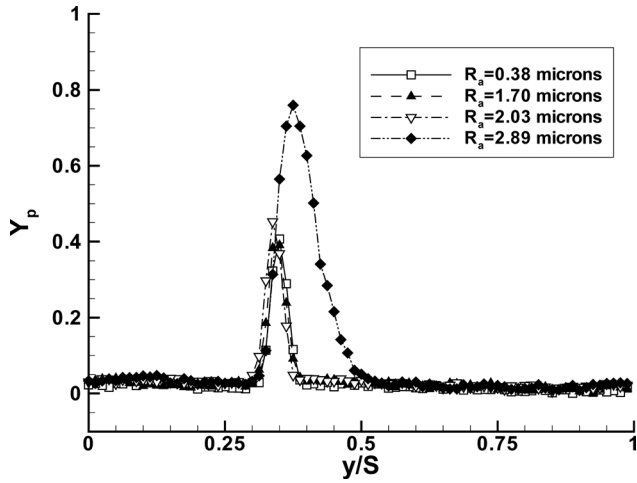


Fig. 6 Loss coefficient versus pitch for R_a of 0.38, 1.70, 2.03, and 2.89 μm at Reynolds number 600,000

increased significantly to 30% of the pitch. Also, the maximum loss coefficient value doubled to 0.8. Roughness mainly affected loss on the suction side; however, the pressure side was hardly affected. The increase in loss mainly on the suction side qualitatively agreed with those found by Leipold et al. [6] and Back et al. [13]. Furthermore, from the perspective of loss (as was found in blade loading), a critical roughness exists between R_a of 2.03 and 2.89 μm for this blade at a Reynolds number of 600,000. This trend also qualitatively agreed with the findings of Back et al. [13].

Figure 7 shows the mass-averaged loss coefficient plotted versus roughness magnitude for Reynolds numbers of 400,000 and 600,000 from the current study. For comparison, the centerline averaged roughness (R_a) has been converted to nondimensionalized equivalent sandgrain roughness (k_s/c) by using Schaffler's [2] correlation ($k_s = 8.9 R_a$). Compared to the polished blade ($k_s/c = 0.000028$; $R_a = 0.38 \mu\text{m}$) increasing roughness leads to increased loss. For a Reynolds number of 600,000, a rapid increase of loss between k_s/c of 0.000142 and 0.000203 was observed. Thus the Reynolds number sensitivity of loss to roughness increased visibly between R_a of 2.03 and 2.89 μm . For comparison, the data from Back et al. [13] for $k_s/c = 0.00006$ at $Re = 400,000$ is also shown.

Figure 8 shows the chordwise static pressure distribution of each magnitude of roughness at a Reynolds number of 400,000. At 0.47 chord from the leading edge, the separation bubble became smaller between R_a of 2.03 and 2.89 μm . This diminished separation bubble can be an evidence for decreasing loss between R_a of 2.03 and 2.89 μm at Reynolds numbers of 300,000 and 400,000.

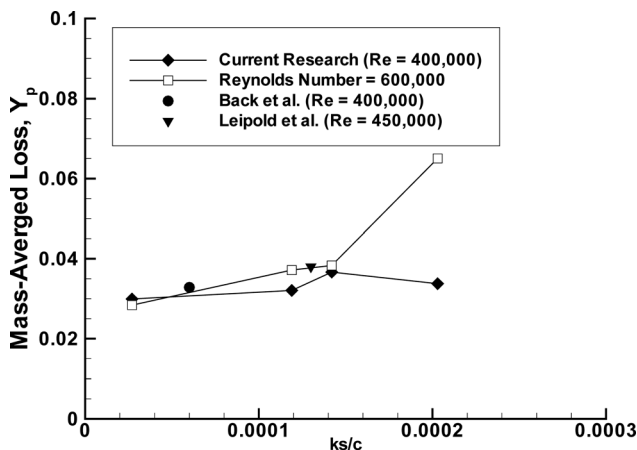


Fig. 7 Mass-averaged loss coefficient versus roughness magnitude for k_s/c

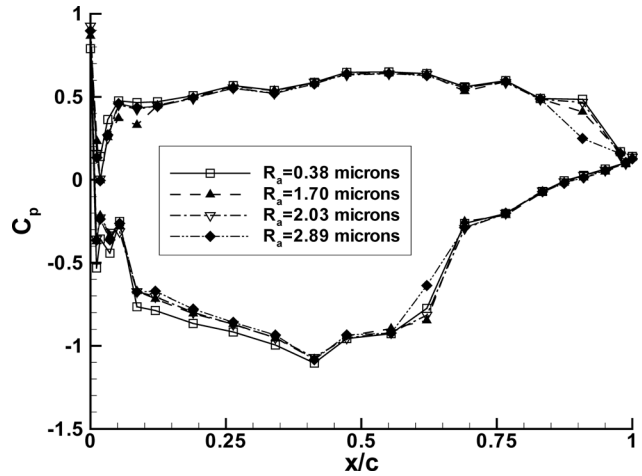


Fig. 8 Blade loading for R_a of 0.38, 1.70, 2.03, and 2.89 μm at Reynolds number 400,000

Figure 9 shows the mass-averaged loss coefficient plotted versus Reynolds number. For the polished surface ($R_a = 0.38 \mu\text{m}$), a slight decrease of loss coefficient between Reynolds numbers of 400,000 and 640,000 can be seen. Averaged loss decreased from 0.030 to 0.027 as the Reynolds number increased from 400,000 to 640,000. However, for R_a of 1.70 μm and higher roughness magnitude, mass averaged loss increased with Reynolds number. The trend is likely due to the turbulent separation occurred for R_a of 2.89 μm at a Reynolds number of 600,000 (Fig. 5). Among three rough blades, for the cases of R_a of 1.70 and 2.03 μm , loss is less sensitive to Reynolds number. However, for $R_a = 2.89 \mu\text{m}$, loss increased rapidly with Reynolds number. Loss coefficient increased from 0.034 for a Reynolds number of 300,000 to 0.065 for a Reynolds number of 600,000. Roughness effect is especially significant at Reynolds numbers above 500,000. Leipold et al. [6] found similar trends for R_a of 11.15 μm between Reynolds numbers of 300,000 and 1×10^6 in a highly loaded, transonic cascade.

Effects of Roughness Location. Figure 10 shows the blade's chordwise static pressure distributions for the baseline case and the cases with various roughness coverage at the Reynolds number of 600,000. According to the C_p distribution, turbulent separation occurred at chordwise location x/c of approximately 0.70 when the suction side was roughened or when the entire blade (pressure side and suction side) was roughened at a Reynolds number of 600,000. Therefore, the increased loss in the aft half chord of the suction side is at least partially due to turbulent separation.

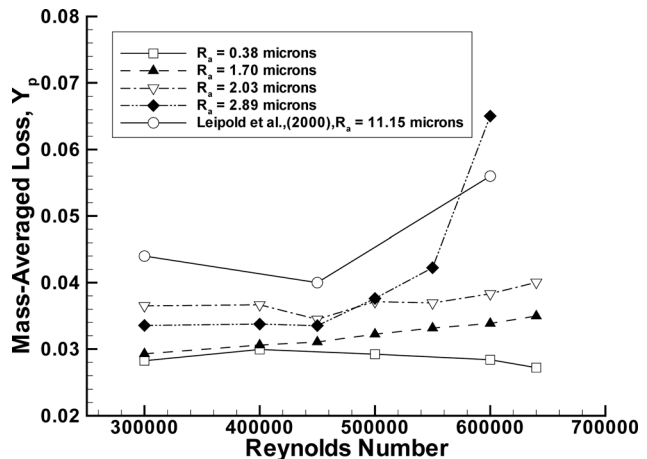


Fig. 9 Mass averaged loss coefficient versus Reynolds number for R_a of 0.38, 1.70, 2.03, and 2.89 μm

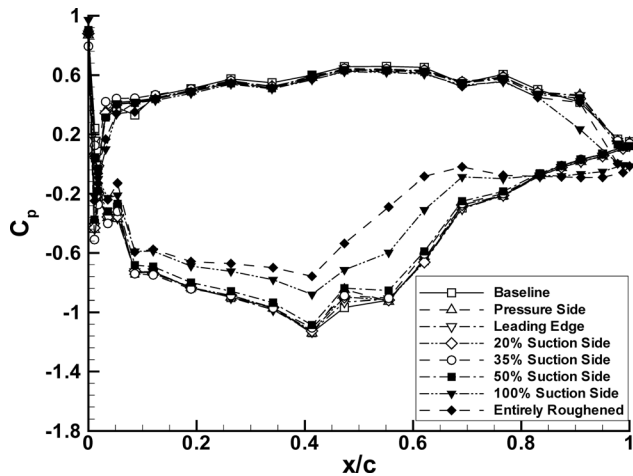


Fig. 10 Static pressure coefficient for Reynolds number of 600,000 for baseline blade ($R_a = 1.70 \mu\text{m}$)

Static pressure on the pressure side was hardly affected except when roughness covers the entire suction side. Static pressure distribution on the suction side decreased only slightly in absolute magnitude as the roughened suction side region was extended from the leading edge. However, turbulent separation occurs at a chordwise location x/c of approximately 0.70 when either the entire suction side or entire blade was roughened. Comparing the blade with a 100% rough suction side and the entirely rough blade, the suction side C_p is further decreased in absolute magnitude for the latter case.

Figure 11 shows the loss coefficient plotted versus nondimensional pitch for a Reynolds number of 600,000. Having roughness only on the pressure side does not affect loss in a measureable manner. On the contrary, both the width and depth of wake increased as the suction side roughened area increased. Partial roughness on the suction side increased the wake deficit more than its width. Until 20% of the suction side is roughened, the wake width was hardly affected. This result is unlike the results found in rotating facilities by Suder et al. [19] in a transonic compressor and by Gbadebo et al. [20] in a low-speed compressor. They found that most of the reduction in total-to-total pressure rise was due to roughening of the leading edge. However, Gbadebo et al. [20] also found the most of the pressure rise reduction is due to the 3D endwall flow. On the contrary, this study focuses on profile loss. Profile loss is not affected by roughness covering from the leading edge to 20% of the suction side in both investigations (Gbadebo et al. [20] and current research). Comparing the

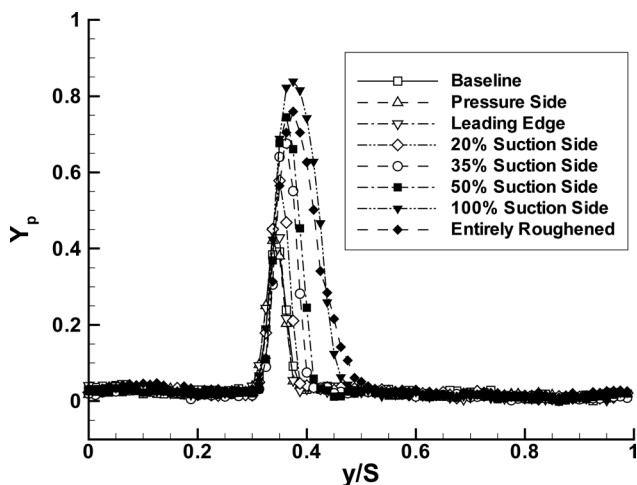


Fig. 11 Loss coefficient versus pitch for Reynolds number of 600,000 for baseline blade ($R_a = 1.70 \mu\text{m}$)

entirely rough blade with 100% suction side roughness, the latter's wake has narrower width but higher maximum loss. Thus, roughness on the pressure side can be thought to enhance turbulent flow mixing and widen the wake. Thus, pressure side roughness by itself does not affect loss but further increases loss when the suction side is roughened as well.

Figure 12 shows the mass-averaged loss coefficient plotted versus the roughened range from the leading edge on the suction side for Reynolds number 600,000. Baseline and entirely roughened cases as well as the pressure side roughness only case are also shown. For the blade with roughness only on the pressure side, the loss was almost identical to that for the baseline and 5% roughened blades. With roughness on the suction side, an almost linear increase in loss can be found between 35% and 100% suction side roughness coverage. On the contrary, relatively little change can be seen between 0% to 35% suction side roughness coverage. Thus, roughness on the downstream 2/3 of the suction side surface has a greater influence on loss. This result could be due to either the cumulative effect of the preceding roughened suction surface reaching a critical threshold or the roughness having greater effect on loss generation in the turbulent flow region. As well known and confirmed by Roberts and Yaras [23], roughness induces earlier transition. In addition, Brzek et al. [24] reported increased turbulent fluctuation and skin friction due to roughness in a turbulent flow region. Therefore, in a turbulent region, larger eddies can be generated due to surface roughness, and that could be the main reason for the rapid increase in loss in the aft half-chord of the blades. Finally, turbulent separation can be another reason for increased loss. Denton [25] stated that the separated region gives larger vorticities, increasing loss.

Effects of Roughness Location and Reynolds Number. Figure 13 shows the mass-averaged loss coefficient plotted versus Reynolds number for the tested roughness locations. Roughness on the pressure side and the leading edge do not affect performance. Also, for up to 20% of the suction side coverage with roughness, the mass-averaged loss was hardly influenced by Reynolds number. Even when the suction side roughness coverage extends beyond 20% from the leading edge, the mass-averaged loss changed little at lower Reynolds numbers but increased with higher Reynolds numbers. As more of the suction surface is roughened, (1) the losses at higher Reynolds numbers increase and (2) the Reynolds number at which loss begins to increase is decreased. As roughness coverage increases from 35% of the suction side to the entire suction side, the Reynolds number at which loss begins to increase is lowered from 600,000 to 450,000. Thus, the loss is more sensitive to roughness in the aft half-chord of the blade. Finally, as more of the suction side is

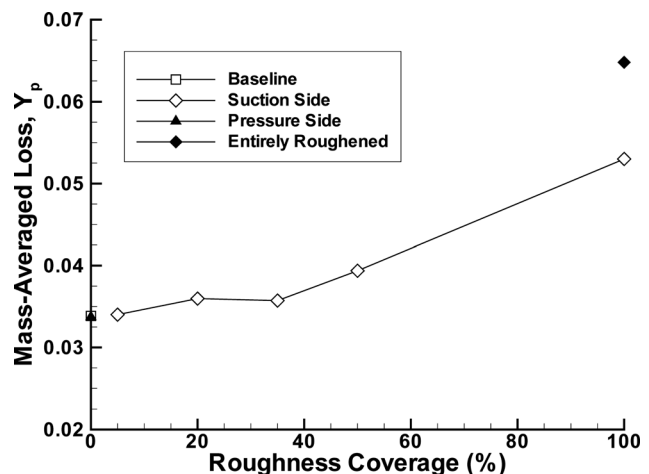


Fig. 12 Mass-averaged loss versus chordwise suction side roughened area for Reynolds number of 600,000

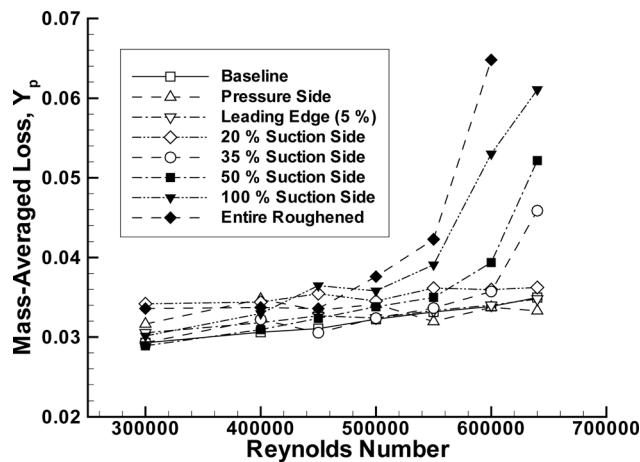


Fig. 13 Mass-averaged loss versus Reynolds number for each roughness location

roughened, the loss increases more rapidly with Reynolds number.

Conclusions

The conclusions of this investigation can be summarized as follows:

- (1) For a given roughness ($R_a = 2.89 \mu\text{m}$), performance begins to deteriorate significantly at a critical Reynolds number.
- (2) For a Reynolds number of 600,000, a critical roughness exists between R_a of 2.03 and $2.89 \mu\text{m}$ from the perspective of loss.
- (3) Performance degradation (in loading and loss) is mainly due to roughness on the suction side. However, for Reynolds numbers above 500,000, roughening the entire blade (or adding roughness to the pressure side in addition to roughness on the suction side) further reduces loading and increases loss.
- (4) The mass-averaged loss is affected mostly by suction side roughness in the aft half-chord ($0.5 < x/c < 1.0$).
- (5) While the loss for a smooth blade is insensitive to Reynolds number, the loss for blades with roughness on the suction side increases with Reynolds number.
- (6) Increasing the area covered by roughness on the suction side from the leading edge reduces the Reynolds number at which the loss begins to increase.

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Nomenclature

c = chord length
 H = span
 S = pitch length
 σ = solidity
 Re = Reynolds number
 R_a = centerline averaged roughness
 k_s = equivalent sandgrain roughness
 C_p = static pressure coefficient
 C_{ps} = static pressure on the suction side

C_{pp} = static pressure on the pressure side
 P = static pressure on the blade taps
 P_0 = downstream total pressure
 P_1 = upstream static pressure
 P_{01} = upstream total pressure
 α_i = inlet flow angle
 Γ = stagger angle
 Y_p = loss coefficient

References

- [1] Bammert, K., and Woelk, G. U., 1979, "Influence of the Blading Surface Roughness on the Aerodynamic Behavior and Characteristic of an Axial Compressor," *ASME J. Eng. Power*, **102**, pp. 283–287.
- [2] Schaffler, A., 1980, "Experimental and Analytical Investigation of the Effects of Reynolds Number and Blade Surface Roughness on Multistage Axial Flow Compressors," *ASME J. Eng. Power*, **102**, pp. 5–13.
- [3] Malt'sev, Y. N., and Shakov, V. G., 1989, "Influence of Roughness of Deposits in Compressor Cascade on Flow Lag Angle," Soviet Aeronautics (English translation of *Izvestiya VUZ, Aviatsionnaya Tekhnika*), **32**, pp. 90–92.
- [4] Taylor, R. P., 1990, "Surface Roughness Measurements on Gas Turbine Blades," *ASME J. Turbomach.*, **112**, pp. 175–180.
- [5] Bons, J. P., Taylor, R. P., McClain, S. T., and Rivir, R. B., 2001, "The Many Faces of Turbine Surface Roughness," *ASME J. Turbomach.*, **123**, pp. 739–748.
- [6] Leipold, R., Boese, M., and Fottner, L., 2000, "The Influence of Technical Surface Roughness Caused by Precision Forging on the Flow Around a Highly Loaded Compressor Cascade," *ASME J. Turbomach.*, **122**, pp. 416–425.
- [7] Schreiber, H.-A., Steinert, W., and Kusters, B., 2002, "Effects of Reynolds Number and Free-Stream Turbulence on Boundary Layer Transition in a Compressor Cascade," *ASME J. Turbomach.*, **124**, pp. 1–9.
- [8] Millsaps, K. T., Baker, L. J., and Patterson, J. S., 2004, "Detection and Localization of Fouling in a Gas Turbine Compressor From Aerodynamic Measurements," ASME Paper No. GT2004-54173.
- [9] Aker, G. F., and Saravanamuttoo, H. I. H., 1989, "Predicting Gas Turbine Performance Degradation due to Compressor Fouling Using Computer Simulation Techniques," *ASME J. Eng. Gas Turbines Power*, **111**, pp. 343–350.
- [10] Massardo, A. F., 1991, "Simulation of Fouled Axial Multistage Compressors," IMechE Conference on Turbomachinery, Paper No. C423/048, pp. 243–252.
- [11] Syverud, E., and Bakken, L. E., 2006, "The Impact of Surface Roughness on Axial Compressor Performance Deterioration," ASME Paper No. GT2006-90004.
- [12] Song, T. W., Kim, T. S., Kim, J. H., and Ro, S. T., 2001, "Performance Prediction of Axial Compressors Using Stage Characteristics and Simultaneous Calculation of Interstage Parameters," *Proc. Inst. Mech. Eng., Part A*, **215**, pp. 89–98.
- [13] Back, S. C., Jeong, I. C., Sohn, J. L., and Song, S. J., 2009, "Influence of Surface Roughness on the Performance of a Compressor Blade in a Linear Cascade—Experiment and Modeling," ASME Paper No. GT2009-59703.
- [14] Tay, G. F. K., Kuhn, D. C. S., and Tachie, M. F., 2009, "Particle Image Velocimetry Study of Rough-Wall Turbulent Flows in Favorable Pressure Gradient," *ASME J. Fluids Eng.*, **131**, p. 061205.
- [15] Pailhas, G., Touvet, Y., and Aupoix, B., 2008, "Effects of Reynolds Number and Adverse Pressure Gradient on a Turbulent Boundary Layer Developing on a Rough Surface," *J. Turbul.*, **9**, pp. 1–24.
- [16] Bammert, K., and Sandstede, H., 1980, "Measurement of the Boundary Layer Development Along a Turbine Blade With Rough Surfaces," *ASME J. Eng. Power*, **102**, pp. 978–983.
- [17] Zhang, Q., and Ligrani, P. M., 2006, "Aerodynamic Losses of a Cambered Turbine Vane: Influences of Surface Roughness and Freestream Turbulence Intensity," *ASME J. Turbomach.*, **128**, pp. 536–546.
- [18] Yun, Y. I., Park, I. Y., and Song, S. J., 2005, "Performance Degradation due to Blade Surface Roughness in a Single-Stage Axial Turbine," *ASME J. Turbomach.*, **127**, pp. 137–143.
- [19] Suder, K. L., Chima, R. V., Strazisar, A. J., and Roberts, W. B., 1994, "The Effect of Adding Roughness and Thickness to a Transonic Axial Compressor Rotor," ASME Paper No. 94-GT-339.
- [20] Gbadebo, S. A., Hynes, T. P., and Cumpsty, N. A., 2004, "Influence of Surface Roughness on Three-Dimensional Separation in Axial Compressors," ASME Paper No. GT2004-53619.
- [21] Gelder, T. F., Schmidt, J. F., Suder, K. L., and Hathaway, M. D., 1989, "Design and Performance of Controlled-Diffusion Stator Compared With Original Double-Circular-Arc Stator," NASA Technical Paper No. 2852.
- [22] Hansen D. J., 1995, "Investigation of Second Generation Controlled-Diffusion Compressor Blades in Cascade," M.S. thesis, Naval Postgraduate School, Monterey, CA.
- [23] Roberts, S. K., and Yaras, M. I., 2004, "Boundary-Layer Transition Over Rough Surfaces With Elevated Free-Stream Turbulence," ASME Paper No. GT2004-53668.
- [24] Brzek, B. G., Cal, R. B., Johansson, G., and Castillo, L., 2008, "Transitionally Rough Zero Pressure Gradient Turbulent Boundary Layers," *Exp. Fluids*, **44**, pp. 115–124.
- [25] Denton, J. D., 1993, "Loss Mechanisms in Turbomachines," *ASME J. Turbomach.*, **115**, pp. 621–656.