Numerical Study of Stage Roughness Variations in a High Pressure Compressor

Hendrik Seehausen1,2, Philipp Gilge1, Andreas Kellersmann2, Jens Friedrichs2 and Florian Herbst1

1 Junior Research Group Multiphysics of Turbulent Flows, Institute of Turbomachinery and Fluid Dynamics, Leibniz Universität Hannover
Appelstr. 9, 30167 Hannover, GERMANY
2 Institute of Jet Propulsion and Turbomachinery, Technische Universität Braunschweig
Hermann-Blenk-Str. 37, 38108 Braunschweig, GERMANY
3 E-mail: seehausen@tfd.uni-hannover.de

ABSTRACT
The objective of this study is to quantify the sensitivity of blade roughness on the overall performance of a 10-stage high-pressure compressor of the jet engine type V2500-A1. The Reynolds-Averaged Navier-Stokes flow solver TRACE is used to study the multi-stage compressor. The three-dimensional numerical setup contains all geometric and aerodynamic features such as bleed ports and the variable stator vanes system. In order to estimate the effect of stage roughness on overall compressor performance, compressor maps of the CFD-model are created by modeling the surface roughness separately for a single stage and combinations of stages. The surface roughness values are applied to the blade’s suction side of the first, center and last stage in the CFD-model by setting an equivalent sand-grain value. This equivalent sand-grain roughness is determined from non-intrusive measurements of blade surfaces of an equivalent real aircraft engine for the first, center and last stage. In addition, further simulations are conducted to analyze the performance drop of a fully rough HPC due to surface roughness. The studies are performed at the operating conditions ‘cruise’ and ‘take-off’ to cover two different Reynolds number regimes. The results show that the models with roughness in a single stage already lead to significantly lower mass flow rates because of higher blockage compared to the smooth compressor. In fact, roughness at the first stage has the biggest effect on the overall performance with a drop in performance of about 0.1% while the effect of the last stage is the smallest. This behavior is mainly caused by enhanced instabilities through the compressor changing the stage-by-stage matching of the stages downstream. In addition to the displacement of the compressor maps to a lower mass flow, a reduction of stall and choke margins is noticeable.

NOMENCLATURE
Abbreviations

| Abbreviation | Description                                      |
|--------------|--------------------------------------------------|
| B            | Blade                                            |
| CAD          | Computer Aided Design                           |
| CFD          | Computational Fluid Dynamics                    |
| DLR          | Deutsches Zentrum für Luft- und Raumfahrttechnik (German Aerospace Center) |
| FEM          | Finite Element Method                           |
| HPC          | High Pressure Compressor                        |
| IGV          | Inlet Guide Vane                                |
| LE           | Leading Edge                                    |
| PS           | Pressure Side                                    |
| RANS         | Reynolds-Averaged Navier-Stokes                  |

Presented at International Gas Turbine Congress 2019 Tokyo, November 17-22, Tokyo, Japan
Review Completed on June 11, 2020

SFC Specific Fuel Consumption
SS Suction Side
TLM Turbulent Length Scale
TRACE Turbomachinery Research Aerodynamics Computational Environment
VSV Variable Stator Vane
V Vane

Latin Symbols

| Symbol | Description                           |
|--------|---------------------------------------|
| c      | Chord Length                          |
| H      | Absolute Duct Height                  |
| h      | Height                                |
| k      | Turbulent Kinetic Energy              |
| k_s   | Sand-grain Roughness                  |
| k_s^+ | Non-dimensional Sand-grain Roughness |
| m_corr| Corrected Mass Flow                   |
| Ma     | Mach Number                           |
| n      | Rotational Speed                      |
| p      | Pressure                              |
| Ra     | Average Roughness                     |
| Re     | Reynolds Number                       |
| Tu    | Turbulent Intensity                   |
| T     | Temperature                           |
| U     | Velocity Magnitude                    |
| u_t   | Shear Velocity                        |
| u^+   | Non-dimensional Velocity             |
| y^+   | Non-dimensional Wall Distance         |

Greek Symbols

| Symbol | Description                           |
|--------|---------------------------------------|
| α      | Absolute Flow Angle                   |
| β      | Relative Flow Angle                   |
| η_{pol} | Polytropic Efficiency                |
| κ      | Isentropic Exponent                   |
| κ_{le} | Angle of Tangent to Mean Camber Line |
| μ      | Dynamic Viscosity                     |
| μ_e   | Eddy-Viscosity                        |
| ν      | Kinematic Viscosity                   |
| π_{tt} | Total Pressure Ratio                  |
| ρ      | Density                               |
| τ_w   | Wall Shear Stress                     |
| τ_{tt} | Total Temperature Ratio               |
| ω      | Specific Dissipation Rate             |

Subscripts

| Subscript | Description |
|-----------|-------------|
| in        | Inlet       |
| out       | Outlet      |
| t         | Total       |

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INTRODUCTION

A compressor increases the total pressure resulting in an adverse pressure gradient and unstable boundary layers. Unstable boundary layers tend to increase losses through the flow domain. This causes the compressor blade to be highly sensitive to boundary layer separation and a smaller operating area when e.g. interacting with surface roughness. Surface roughness on compressor blades leads to a reduction in the pressure rise, and hence a decrease in efficiency of the whole engine [1]. During operation, a turbomachine such as an aircraft engine experiences many different conditions, causing deterioration and fouling. The operational conditions are described by environmental factors such as aerosols, dust and other particles [2]. Among other effects such as geometry deterioration, this results in complex surface roughness on the blading of these engines [3,4]. In general, surface roughness leads to a drop in performance and, thus, an increased specific fuel consumption (SFC). The significance of rough blading on the aerodynamic behavior also depends on the component of the engine.

Most of the investigations into the aerodynamic effects of surface roughness are based on findings of Nikuradse [5]. Nikuradse observed different flow regimes depending on Reynolds number and surface roughness. Further investigations of Schlichting [6] allowed the transformation of the technical roughness used by Nikuradse into an equivalent sand-grain roughness parameter $k_s$ to describe the aerodynamic drag. Consequently, this resulted in a dimensional roughness parameter $k_s$ based on the sand-grain value $k_s$, shear velocity $u_s$ and kinematic viscosity $\nu$ to quantitatively characterize the effect of surface roughness on the flow:

$$k_s = \frac{k_s u_s}{\nu}.$$  \hspace{1cm} (1)

To date, investigations on the effect of blade surface roughness have mainly concentrated on a single blade or single stage [7,8]. Also effects of non-uniformly distributed surface roughness are well documented. Studies have shown that the location of roughness is significant for the losses and can be correlated with the flow regime predominating at these locations. In addition, it was reported that the performance reduction due to local roughness cannot be used to predict the change in performance of combined roughness [9,10].

Investigations of multistage compressors have mainly concentrated on the relation between surface roughness and Reynolds number. Operation conditions with high Reynolds numbers, such as ‘take-off’, have a higher sensitivity to surface roughness, than, e.g. ‘cruise’. This results in a greater proportion of operation in hydraulically smooth regimes due to the lower Reynolds number. Nevertheless, even for low surface roughness caused by manufacturing, the surface roughness can behave hydraulically rough when a critical Reynolds number is reached [11].

Furthermore, increased surface roughness leads to a thicker boundary layer implying an increased displacement thickness and a broader wake downstream of the blade [12]. This, in turn, results in lower mass flow rate passage flows implying higher blockage and an increase in velocity magnitude in the mean flow [7]. In addition, higher flow losses are caused by the enlargement of three-dimensional separation due to surface roughness, especially in hub regions [13]. As a result, compressor maps are narrowed and shifted to lower mass flows because of surface roughness induced higher flow losses [14,15].

Several studies on compressor stages and multistage compressors were conducted to consider whether the blade or vane is responsible for the performance drop induced by surface roughness. Morini et al. [15] performed a study of non-uniform surface roughness effects on the compressor stage performance of the transonic NASA Stage 37 test case. They used a roughness parameter of $k_s = 40\mu m$ ($R_a = 6.45\mu m$). By comparing rough blades combined with either rough or smooth vanes, they concluded that the blade was responsible for almost the entire performance loss. Furthermore, roughness on the suction side of the blades contributed most to the performance loss.

An investigation by Teng et al. [16] showed similar result for a 5-stage low pressure compressor with uniformly distributed equivalent sand-grain roughness $k_s$. The $k_s$ values they used were in a range of $8.75\mu m$ to $14.35\mu m$ on the suction side, and $7.26\mu m$ to $10.39\mu m$ on the pressure side. The Reynolds number was $1.4\times10^6$ at the inlet guide vane. Simulations showed that surface roughness had smaller effects on the performance on vanes than on blades. In addition, the last stage was responsible for most of the efficiency drop with a share of 0.32%. This behavior could be explained by an extending separation into the mean span on the rotor’s suction side of stage five. Unfortunately, none of the presented studies offered non-dimensional equivalent sand-grain roughness values $k_s''$, which would result in more meaningful comparability.

Even though there have been investigations on the effect of surface roughness on the performance of multistage compressors, it is necessary to understand these effects more precisely. Most of the investigations concentrated on uniformly distributed surface roughness over the stages and not studied the loss-sources in detail. Therefore, this study will go into the stage-by-stage design changed by surface roughness and identify significant stages for the overall compressor performance at different operation points. This can lead to a better prediction of roughness-induced losses, and a reduction in frequency of servicing. In this paper, the significance of non-uniformly distributed surface roughness on the performance reduction is numerically determined by setting equivalent sand-grain values $k_s$ in the front, center and last stage of a V2500 HPC CFD-model separately and in combination. The roughness values are measured on real operationally used blades of high-pressure compressors from medium sized high-bypass aircraft engines [3,4]. In addition, the obtained roughness values were interpolated to gain numerical results of an operationally used V2500 HPC with all blades and vanes rough. This makes it possible to describe the dimension of performance reduction due to roughness, and to compare them with losses caused by geometry deterioration.

HPC GEOMETRY

The HPC geometry used in this study was already presented by Reitz et al. [17]. A digital model of the V2500 HPC consisting of 10 compressor stages and an inlet guide vane (IGV) was built (Fig. 1). The inflow was provided by the low-pressure compressor through an S-duct to provide the correct flow towards the IGV. The complete annulus was modeled as symmetric. The shroud geometry was generated using technical drawings and the hub geometry using digitized blades aligned on the digitized drums of the HPC.

To adjust the flow conditions at different rotational speeds, the first five stages are designed as variable stator vanes (VSV). Thus, for every operation condition simulated the VSV position had to be adjusted. The first four VSVs are pivot mounted at the shroud, as well as at the hub. Thus, cavities are necessary and have an influence on the flow characteristics. The fifth VSV is only pivot mounted at the shroud, and is cantilevered at the hub. Therefore, the model was equipped with cavities, which were built to be rotationally symmetrical. The flow in cavities is difficult to predict caused by its unsteady behavior and significantly depends on the domain size [18]. However, to provide a good approximation of the influence of cavity flows on the main flow, the flow within the cavities labyrinth seals were modeled in their number and height. Bleed ports were also modeled as rotationally symmetrical, even though the bleed ports were originally designed as cylindrical holes. This rough estimation was necessary to provide a symmetric model with one blade per row, and to be able to use periodic boundary conditions.

For the digital model of the HPC, newly manufactured blades and vanes were digitized by a structured light 3D scanner in conjunction with a photogrammetric system. The resulting geometries were used to model CAD datasets. Subsequently, realistic blade geometries were generated by FEM geometry transformation using
the rotational speed, temperature and pressure field of operating point ‘climb’. The radial clearance of blades and vanes was set to approximately 1% of operational blade height using engine manual information, the airfoil lengthening and the annulus dilation. Additionally, the first rotor was equipped with snubbers, which have strong influence on the aerodynamics and the blockage of the HPC. Also the outlet guide vane (OGV) was designed and modeled as a tandem blade row.

**MEASURED SURFACE ROUGHNESS**

Since the implementation of the equivalent sand-grain surface roughness $k_s$, many correlations with technical roughness were introduced [1]. The correlation of Sigal and Danberg [19] was used by Gilge et al. [3, 4]. Gilge et al. measured the surface roughness of real operationally used blades of HPCs with a confocal laser microscope. The obtained surface roughness is given as typical height for engines, which had completed 20,000 cycles in geographic regions with mainly low environmental influences. Figure 2 shows some of these typically arising surface structures on blades of midsize bypass aircraft engines. On the pressure side, homogeneous structures caused by impacts, erosion and depositions were identifiable and resulted in isotropic distributed roughness elements.

Moreover, the roughness-height decreased along the chord, while maintaining the isotropic structures. On the suction side, the structure of the roughness elements differed extremely along the chord. At the leading edge, anisotropic structures occurred, while downstream more homogeneous and isotropic structures were detectable. The anisotropic structures are characteristic for oil leakages of the observed surface roughness can be found in Gilge et al. [3,4]. Additionally, they present the equivalent sand-grain roughness values $k_s$ on the blades of the stages one, five and ten as marked in Fig. 1. These values could be used to approximate a typically deteriorated surface of each blade by an averaging of the $k_s$ values.

**NUMERICAL SETUP**

**Meshing**

The meshing was performed by an automated routine using Python-scripts controlling Numeca Autogrid and the HPC geometry was provided by Reitz et al. [17] for this study. The model was built as one pitch periodic with a structured mesh. The blades were modeled with an O-grid within an H-grid. The non-dimensional wall distance at the blades and vanes is $y^+=3$ and at hub and shroud $y^+=10$. The resulting mesh has a grid consisting of approximately 26 million cells. A mesh study can be found in Reitz et al. [17].

**Simulation Setup**

The non-commercial flow solver TRACE 9.1 [21–24] from the Institute of Propulsion Technology at the German Aerospace Center (DLR) was used to conduct the numerical simulations. The solver uses a finite-volume method with structured multi-block meshes to solve the three-dimensional Favre-averaged Navier-Stokes equations. Convective fluxes are discretized by Roe’s second-order upwind scheme, diffusive fluxes by a 2nd order central differencing scheme. The inlet boundary conditions and other solver settings are presented in Tab. 1.

In the current study, the investigated throttle lines were reduced to the operating points at ‘cruise’ and ‘take-off’ to limit the computational effort of the numerical simulations. Hence, this resulted in two different averaged Reynolds numbers of about $5 \times 10^7$ and $1.5 \times 10^8$, calculated with the mass averaged flow values and referred to the inlet condition in the reference frame of each stage:

$$R_e = \frac{\rho_{in} \, c \, U_{in}}{\mu_{in}}$$

(2)

The averaged Reynolds number was obtained by an averaging over all stages with constant Reynolds number except the IGV.
Based on Schaffler [11], the operating points at ‘cruise’ and ‘take-off’ cover two significant Reynolds numbers.

The generation of the throttle lines was performed by increasing the static pressure at the HPC outlet. An implemented convergence criterion was requested after each simulation to ensure the mass conservation. Consequently, a mass flow difference between inlet and outlet including the bleed ports smaller than 0.1% satisfied the stopping criterion. A converged mass flow was accompanied by the asymptotic convergence of efficiency and pressure ratio. The last converged simulation of the throttle line determined the numerical point of stall.

In the calculations, the $k-\omega$ turbulence model of Wilcox [25] was used while near-wall effects are modeled by the wall function. Thus, no resolving of the viscous sublayer is performed. The wall function is based on the friction velocity $u_t$ which describes a turbulent velocity scale near the wall:

$$u_t = \frac{\tau_0}{\rho} \left( \frac{\nu}{\kappa} \right)^{1/2} \left( \frac{1}{k_s} \right)$$  \hspace{1cm} (3)

As a result of rough surfaces arising from deterioration the wall function has to be modified. The logarithmic profile:

$$\frac{U}{u_t} = \left( \frac{1}{\kappa} \right) \ln \left( \frac{1}{k_s} \right)$$  \hspace{1cm} (4)

still exists, but is adapted by the equivalent sand-grain roughness $k_c$. For this, the value $C^+$ of the logarithmic profile is expressed as a function of $k_c$ for completely rough walls resulting in [25]:

$$C^+ = \frac{1}{\kappa} \ln \left( \frac{1}{k_s} \right) + 8.4.$$  \hspace{1cm} (5)

**Validation**

To analyze the CFD simulations, a post-processing was applied to calculate the pressure ratio $\pi_t$ and polytropic efficiency $\eta_{poly}$ of each operating point. The polytropic efficiency $\eta_{poly}$ is calculated from the total pressure ratio $\pi_t$:

$$\pi_t = \frac{p_{t,\text{out}}}{p_{t,\text{in}}}$$  \hspace{1cm} (6)

and total temperature ratio $\tau_t$:

$$\tau_t = \frac{T_{t,\text{out}}}{T_{t,\text{in}}}$$  \hspace{1cm} (7)

as follows

$$\eta_{poly} = \frac{\kappa - 1}{\kappa} \log(\pi_t) - \frac{\kappa - 1}{\kappa} \log(\tau_t).$$  \hspace{1cm} (8)

| Setting | Comment |
|---------|---------|
| $p_{t,\text{in}}, T_{t,\text{in}}, \Delta u_{\text{in}}(b/H)$ | Extrakte from |
| $M_{\infty}, T_{\infty}, \text{TLMs}$ | full performance synthesis calculations |
| Rotational speed | $n = n_{\text{cruise-Bake-off}}$ |
| Walls | No slip walls (hydraulically smooth) |
| Blade to vane interface | Mixing plane |
| Turbulence model | Wilcox $k-\omega$ [25] |
| Stagnation point anomaly fix | Kato-Lauder [26] |
| Analysis type | Steady state |
| Blade/vane meshes | One pitch periodic |

In Fig. 3, the compressor map of the CFD simulations obtained for the hydraulically smooth setup is shown (black squares). The corrected compressor inlet mass flow $\dot{m}_{\text{corr}}$

$$\dot{m}_{\text{corr}} = \dot{m}_{\text{in}} \frac{101325 \, Pa}{P_{\text{in}}} \sqrt{\frac{T_{\text{in}}}{288.15 \, K}}$$  \hspace{1cm} (9)

is normalized with respect to the mass flow of the aerodynamic design point at ‘take-off’. The right throttle line represents the operating points of ‘cruise’, while the left throttle line shows the operating points of ‘take-off’.

The smooth compressor map of Fig. 3 was already validated in Reitz et al. [17]. To ensure the validity of the newly simulated throttle lines, a validation for the study of this paper was necessary. Based on the validation of Reitz et al., the available OEM data of Tubbs [27] with the given aerodynamic design point of ‘cruise’ was used. Tubbs set up a test rig with newly manufactured blades
and experimentally generated a compressor map for the V2500-A1 HPC. The validation point (green triangle) is shown in Fig. 3 and the deviations can be found in Tab. 2. The numerical results slightly overestimate the losses, leading to a deviation in mass flow $\dot{m}$ of −0.76%, in polytropic efficiency $\eta_{pol}$ of −0.6% and in total pressure ratio $\pi$ of −0.24%. As mentioned by Reitz et al. [17], the deviations are caused by the reference condition of ‘cruise’. An early stage of operation ‘cruise’ results in a highly-loaded compressor with decreased efficiency as a consequence of the high aircraft weight and the low flight height. For a later stage of operation ‘cruise’ the reduced aircraft weight would result in an increased flight height with reduced rotational speed and, thus, slightly lower mass flow with higher efficiency and nearly constant pressure ratio. As there is little available reference data for a newly manufactured HPC, a further validation for the throttle line of ‘take-off’ was performed using the scaling factors obtained by Reitz et al. [17]. The factors were identified by scaling the aerodynamic design point of ‘take-off’ from the CFD-generated map on the test-cell data of a deteriorated jet engine V2500-A1. This resulted in a constant shifting value for the mass flow, polytropic efficiency and total pressure ratio. These constant values were afterwards multiplied with each operating point. Thus, a compressor map of a deteriorated HPC was approximated. The result is shown in Fig. 3 indicated by the blue crosses. Subsequently, a comparison of the test-cell data point for ‘take-off’ (red triangle) with the scaled aerodynamic design point for ‘take-off’ (blue square) could be performed. As can be seen, a good approximation exists. The deviations are summarized in Tab. 2. To conclude, the setup’s validation indicates a map with a good prediction quality.

FULLY ROUGH HPC

In the previous study of Reitz et al. [17], the effect of a macroscale geometry change by means of doubled tip gaps on the performance of a HPC was investigated. It was shown that doubled tip gaps could not describe the total changes of the worn physical engine. Thus, further effects such as increased surface roughness must occur, affecting the performance reduction. To quantify the relevance of surface roughness, a method had to be developed to extrapolate the equivalent sand-grain values $k_s$ of the blades measured by Gilge et al. [3,4] across all the blades. Therefore, the development of surface roughness induced by erosion and deposition needed to be considered and is presented below.

Modeling of Stage Roughness

A study on the distribution of deposits along the axial compressor flow path was conducted by Tarabrin et al. [29] and was carried out with the 16-stage compressor MS5322R of Nuovo Pignone in Russia. Considering the location in Russia, which is minimally influenced by natural effects [2], a similar action of deposits compared to the measured engine by Gilge et al. [3,4] could be assumed. The results of the deposit measurements are shown in Fig. 4. The most noticeable finding was that the deposits decreased from the IGV to the sixth stage on both blades and vanes. Especially in the front stages, the deposits on vanes were bigger than on blades. This could be explained by the centrifugal forces affecting the particles on the blades [29]. Similar results were observed by Syverud et al. [30], who analyzed the axial compressor deterioration by spraying droplets of saltwater into the engine. The engine was a GE J8513 with eight stages and a total pressure ratio $\pi_t$ of about 6.5. Unfortunately, it was only possible to measure the deposits and surface roughness of the vanes. Results showed that smaller and more homogeneous grain sizes were found on the suction side of the vanes than on the pressure side. This is consistent with the findings of Gilge et al. [3,4] for blades, showing larger and more widely distributed roughness peaks on the pressure side.

Anyway, deposits cannot be used directly as surface roughness, because they have to be transferred into an equivalent sand-grain roughness $k_s$. Thus, a correlation of deposits and surface roughness had to be established. In a computational approach, Melino et al. [31] described the relationship between compressor operating hours and the increase in blade surface roughness. This approach combined CFD simulations of NASA Stage 37 with different values of roughness uniformly distributed across the blade’s surface, and a model to estimate the fouling in axial compressors [32]. Within the model, a stage-scaling method was used to simulate the compressor behavior. To consider the effects of fouling, a modification of the compressor map based on scaling functions was applied. Connecting both approaches, a correlation of surface roughness over time was drawn. Finally, Melino et al. [31] applied the developed method to a 17-stage axial compressor with a pressure ratio $\pi_t$ equal to 12.5. The results show a nonlinear decrease in roughness from the first to fifth stage after 2000 operation hours, which agrees with the deposits located along the flow path ([29][30]).

These studies were used as an orientation to describe the roughness characteristic of the HPC simulation setup used in this study because of a lack of quantitative information about the surface roughness of the blading in multistage engines. Figure 5 shows the selected equivalent sand-grain values $k_s$, which were uniformly distributed over the suction and pressure sides of blades and vanes. The reference points represent the measured roughness values of Gilge et al. [3,4]. These points were used to scale the roughness values to the right roughness level. As stated before, the strongly nonlinear behavior from the IGV to fifth stage is noticeable. Further downstream, the roughness value on the suction side behaves constant, while it slight linear decreases on the pressure side. The roughness parameters on the vanes were set to a higher value oriented on measured deposits of the first stage (see Fig. 4). Similar correlations were reported by Kellersmann et al. [33], who analyzed the effect of surface roughness on the performance of a low-pressure turbine. It is, therefore, supposed that fouling mechanisms cause related effects on the surface roughness of vanes in the HPC.
The Interaction of Deterioration Modes

To analyze the overall compressor performance reduction induced by surface roughness of an operationally worn HPC, a map with a fully rough blading was investigated. The equivalent sand-grain roughness \( k_s \) was obtained by the previously described method. Moreover, a map with fully rough blading and doubled tip gaps was simulated to analyze the interaction of micro-scaled geometry changes such as surface roughness, and macro-scaled geometry changes.

Figure 6 shows the generated compressor maps. Dots and squares represent numerically estimated operating points and maps obtained by the scaling factors of Reitz et al. [17] are marked with crosses. The fully rough map (green dots) is shifted towards lower mass flows. Furthermore, a narrowing of the map with reduced surge and choke margins is obvious, especially for ‘take-off’. The aerodynamic design point of ‘take-off’ is approximately 4% reduced in total pressure ratio \( \pi_{tr} \), approximately 2.36% in polytropic efficiency \( \eta_{p,\text{pol}} \), and approximately 5.73% in mass flow \( m_{\text{corr}} \). The corresponding deviations for ‘cruise’ are approximately 12.3% reduction in total pressure ratio \( \pi_{tr} \), approximately 2.68% in polytropic efficiency \( \eta_{p,\text{pol}} \), and approximately 4% in mass flow \( m_{\text{corr}} \). The effect of surface roughness on the performance reduction is about 1% bigger than the effect of doubled tip gaps [17].

Especially at ‘cruise’, a rough blading leads to a high deficit in pressure rise. It is thought that this phenomenon is based on the higher Reynolds number at ‘take-off’ accompanied by less distinct variations of the nondimensional equivalent sand-grain values \( k_s' \). At ‘cruise’, a smooth blading lies in the hydrodynamically smooth regime, while supposedly a smooth blading already represents a hydrodynamically rough surface at ‘take-off’ [11]. If the surface roughness increases, the aerodynamic behavior of surface roughness becomes hydrodynamically rough at ‘cruise’. This results in higher pressure losses compared to the smooth blading at ‘cruise’ than ‘take-off’. Nevertheless, the mass flow reduction is about 1.73% higher at ‘take-off’ than ‘cruise’. This implies higher blockage and reduced stall margin at ‘take-off’.

To investigate the effect of interacting deterioration modes, the scaling factors for doubled tip gaps of Reitz et al. [17] are applied to the fully rough map. These scaling factors were obtained by calculating the constant shift between the aerodynamic design point of the smooth blading and the blading with doubled tips gaps. The resulting compressor map is illustrated by the magenta crosses in Fig. 6. To verify the applicability of the scaling factors to a fully rough map, the resulting throttle line at ‘take-off’ of the fully rough HPC CFD model with doubled tip gaps is shown by the magenta colored dots. As can be seen, the throttle lines at ‘take-off’ of the scaled and simulated map differ slightly from each other. The aerodynamic design point of the simulated throttle line at ‘take-off’ shows a 0.4% higher mass flow \( m_{\text{tr}} \), 1.72% lower polytropic efficiency \( \eta_{p,\text{pol}} \), and 10.43% lower total pressure ratio \( \pi_{t} \) compared with the scaled throttle line of ‘take-off’. The scaling factor for the pressure ratio is quite small due to small changes in outlet Mach number occurring when doubled tip gaps are present [17]. This leads to the conclusion that there are interactions of the deterioration modes which are not covered by the constant scaling. Nevertheless, the scaled map represents a good approximation of the fully rough HPC CFD model with doubled tip gaps and can be used to predict the effect of additional deterioration modes.

Therefore, a scaled compressor map (blue crosses) of a fully deteriorated V2500-A1 HPC is illustrated in Fig. 6. This map is generated by applying the scaling factors for test-cell validation of Reitz et al. [17] to the smooth map. As can be seen, the map scaled on test-cell data is not narrowed as the fully rough map would predict it. The test-cell data only provide information about the aerodynamic design points. Thus, the scaling factors are based on a linear shifting without the effect of narrowing and cannot be used to precisely predict the compressor map of a fully deteriorated HPC. Nevertheless, it is possible to quantify the difference between the map scaled on test-cell data and the fully rough map scaled on doubled tip gaps. Consequently, the engine investigated in the test-cell needs to have more deterioration modes than surface roughness and doubled tip gaps. These additional modes could include further geometry changes, such as stagger angle and blade geometry variations [34,35].

Even if the increased surface roughness in conjunction with doubled tip gaps cannot comprehensively describe the changes to the compressor map, it has been shown that a fully rough blading has a significant effect on the performance reduction of a multistage HPC. The investigation with applied surface roughness of real operationally worn blades indicates that surface roughness produces higher losses than doubled tip gaps.

Sensitivity to Stage Roughness

Based on the significant effect of a fully rough blading on the performance of a multistage HPC, the question arises where the loss-sources are located. Previous studies already showed that roughness on the suction side has the biggest effect on the losses and shifting of compressor maps [15]. Thus, a further study with uniformly distributed surface roughness over the blade’s suction side of single stages was conducted. The equivalent sand-grain values \( k_s' \) measured by Gilge et al. [3, 4] were applied to the first, fifth and last stage of the simulation model separately. Additionally, simulations with surface roughness on all three stages at once were performed.

Thereby, the stage effect on the compressor performance using surface roughness of operationally used HPCs could be analyzed. Table 3 summarizes the different setups used within the numerical study of stage roughness effects on the overall HPC performance. The setups 1-3 represent investigations with single rough stages,
Table 3: Equivalent sand-grain values $k_s$ set to the surface of blade’s suction side in the simulation model (corresponding $k_s'$ values are shown in Fig. 8)

| Setup | Blade position | first stage [µm] | center stage [µm] | last stage [µm] |
|-------|----------------|------------------|------------------|-----------------|
| 1     | smooth         | 15.26            | smooth           | smooth          |
| 2     | smooth         | 8.21             | smooth           | smooth          |
| 3     | smooth         | smooth           | 8.03             | smooth          |
| 4     | 15.26          | 8.21             | smooth           | 8.03            |

while the last setup contains the combination of surface roughness at all three blades. The resulting compressor maps based on the operation conditions ‘cruise’ and ‘take-off’ are shown in Fig. 7. The corrected mass flow $\dot{m}_{corr}$ of each operating point is normalized by the mass flow of the aerodynamic design point of ‘take-off’. As can be seen, surface roughness has less effect on the performance at operation condition ‘cruise’ than ‘take-off’. At ‘cruise’, the change between the smooth map (black squares), and all blades rough map (magenta diamonds) for the approximated aerodynamic design points corresponds to a 1.74% increase in total pressure ratio $\pi_t$, 0.2% decrease in polytropic efficiency $\eta_{pol}$, and a 1.94% decrease in mass flow $\dot{m}_{corr}$. Compared to this, surface roughness on all three blades results in 2.63% increase in total pressure ratio $\pi_t$, 0.03% reduction in polytropic efficiency $\eta_{pol}$, and 0.87% reduction in mass flow $\dot{m}_{corr}$ at ‘take-off’. In both operation conditions, surface roughness results in increased total pressure ratio at maximum efficiency. This is in conjunction with the mass flow reduction caused by surface roughness. The reason for this will be discussed later in this section. Moreover, it is obvious that most of the performance reduction is caused by surface roughness at the first stage (light green), while surface roughness on the last stage (dark green dots) has a negligible effect (see Fig. 7).

To explain this behavior, the non-dimensional equivalent sand-grain roughness $k_s'$ of blades with rough suction sides is taken into account and illustrated in Fig. 8. The top row (a) contains the $k_s'$ values obtained by the simulations with rough blades in a single stage. The light green lines represent the $k_s'$ values of the first, green dashes the fifth and dark green dots the last stage. In the bottom row (b), the $k_s'$ values of the corresponding blades appearing by the simulations with combined rough stages are shown. The dashed black lines correspond to a $k_s' = 5$, which traditionally defines the change of hydraulically smooth to transitionally rough regimes [36]. In detail, this limit is a function of the Reynolds number itself and therefore a not-fixed indicative value [37]. Nevertheless, most of the $k_s'$ values lie in the transitionally rough regime for both operating points and, thus, indicate that the losses are dependent on the $k_s$ value and the Reynolds number. This results in significantly lower $k_s'$ values for operating points of ‘cruise’ than ‘take-off’. Moreover, this indicates a higher sensitivity to surface roughness for ‘take-off’ (see Fig. 7). Although surface roughness at the first stage affects the performance reduction most, the $k_s'$ values on this stage are the smallest compared to the stages five and ten. Possibly, this significant influence is also caused by an existing shock on the blade’s suction side of the first stage at approximately 0.3 relative chord for the aerodynamic design point of ‘cruise’ and approximately 0.17 relative chord for ‘take-off’ [11]. However, the non-dimensional equivalent sand-grain roughness $k_s'$ increases over the stages, while the effect on the performance decreases. This contradicts the proper definition of the non-dimensional equivalent sand-grain roughness $k_s'$. When comparing the $k_s'$ values of the investigations with single rough blades (a) and the investigation of the combined rough stages (b), no difference in the $k_s'$ values is detectable (see Fig. 8). Hence it can be assumed that surface roughness at the first stage has no significant effect on the $k_s'$ values and, thus, the Reynolds number of stages five and ten. Moreover, no displacement of the loss loca-
tions can be assumed. Consequently, there must be other factors causing the significant shifting of the map by roughness at the first stage.

One potential factors is the location of rough blading in a multi-stage compressor. To verify this hypothesis, the effect of surface roughness at 'take-off' is investigated more precisely. In a first step, the deficits in total pressure rise $\pi_t$ and polytropic efficiency $\eta_{poly}$ of each stage are analyzed to study the forming of instabilities caused by surface roughness at the first and fifth stage on the overall compressor performance. Figure 9 shows the differences in total pressure ratio $\Delta \pi_t = \pi_t_{\text{rough}} - \pi_t_{\text{smooth}}$ and polytropic efficiency $\Delta \eta_{poly} = \eta_{poly_{\text{rough}}} - \eta_{poly_{\text{smooth}}}$ for identical pressure ratio (continuous lines) as well as identical mass flow (dashed lines) (compare Fig. 7). As can be seen, surface roughness at the first stage causes a significantly decreased polytropic efficiency of about 0.8% and a slightly smaller pressure rise compared to smooth for an identical mass flow. To keep the total pressure ratio constant, the loading of each stage has to raise. Considering an equal mass flow rate, the performance rapidly drops downstream of the fifth stage. In contrast, surface roughness at the fifth stage is not affecting the performance of the stage itself, rather it reduces the performance downstream of stage seven for an identical mass flow. The loss in pressure rise and polytropic efficiency is the same at stages eight and nine, as for surface roughness at the first stage. Nevertheless, the overall performance reduction is higher for surface roughness at the first stage due to higher losses at the other stages.

To investigate the reasons of this behavior, the differences in incidence $i$ occurring in each row of the HPC are shown in Fig. 10. A value $\Delta i = i_{\text{rough}} - i_{\text{smooth}}$ higher zero corresponds to an incidence shifted towards the suction side, and a value lower zero towards the pressure side. The changes in incidence are significantly higher for operating points with identical mass flow rates (b) than identical pressure ratios (a). To achieve an identical total pressure rise (a) with rough blades, the mass flow has to be reduced. This leads to a shift of the blade’s inlet flow angles towards the pressure side, accompanied by an increased blade loading. Considering roughness at the fifth stage, the blade loading of the front stages is increased caused by higher velocities and ensures the pressure rise. At the fifth stage, the incidence equals the performance losses induced by the rough surface. In contrast, if keeping an identical mass flow (b), the inlet flow angles are shifted towards the suction side, reducing the blade loading and the consequent pressure rise. Moreover, it is detectable that the incidences are equal to the smooth setup until the instability occurs. In both roughness setups, the incidences are amplified downstream of the blade with roughness, which is consistent with the total pressure rise and polytropic efficiency characteristics (see Fig. 9). To conclude, the surface roughness at the blade of the first and fifth stage leads to instabilities in the flow changing the incidence of blades and vanes downstream. Thereby, the stage-by-stage interface no longer match the original design. This phenomenon is higher for roughness at the first than fifth stage.

The boundary-layer development is studied in order to investigate the forming of instabilities caused by the rough surface at the first stage. Figure 11 represents the displacement thickness $\delta_i$ (dark green) and the momentum thickness $\delta_m$ (light green) over the blade’s suction side of the first stage for the smooth and rough surface. As can be seen, surface roughness slightly increases the boundary-layer parameters, especially at the shock regions. Thus, the flow cross-section is minimized resulting in lower mass flows [7]. This phenomenon is more distinctive for identical pressure ratio (a) than identical mass flow (b) caused by the shifted incidence of about $-0.2^\circ$. Even if these differences in the boundary-layer parameters are quite small, the effect on the downstream blades is significantly different. Considering an identical mass flow (b), the incidence of the downstream vane is shifted towards the pressure side of about $0.5^\circ$. This, in turn, results in thicker boundary layer along the vane compared with identical pressure ratio (a). Fig. 12 illustrates the difference in flow turbulence between rough and smooth blade’s suction side at the first stage. The rough surface results in a broader wake downstream of the blade with higher turbulent kinetic energy $k$ compared with a smooth blade. The location of the developing turbulence in the flow regime is observable in the colorplots. As can
In both cases, the higher turbulent flow results in a higher turbulent kinetic energy over the pitch at the interface between blade and vane of the first stage at mid-span. Compared for operating points with identical pressure ratio (a) and mass flow (b) at ‘take-off’ (see Fig. 7).

be seen, the increase of the eddy-viscosity ratio \( \mu_e/\mu \) calculated with the turbulent kinetic energy \( k \) and specific dissipation rate \( \omega \)

\[
\mu_e = \frac{\rho k}{\omega} \tag{11}
\]

is higher for identical pressure ratio (a) than identical mass flow (b). In both cases, the higher turbulent flow results in a higher turbulence at the following vane induced by the mixing plane. Nevertheless, this higher turbulent flow is not affecting the influence of surface roughness at stages five and ten, as could be seen by comparing the \( k^* \) values of single rough stage and combined rough stages investigations (see Fig. 8).

To conclude, it is not possible to predict the effect of surface roughness in a multistage compressor only with the \( k^* \) values. The location of surface roughness in a multistage compressor is also an important factor for the performance reduction due to the forming of instabilities. Small changes in incidence at the first rows caused by roughness-induced lower mass flow rates and velocity deficits are increasing the incidences downstream, resulting in a sum of losses. Additionally, the amplification of instabilities is dependent on the Reynolds number. More generally, it is important to consider the location in the multistage compressor in addition to the \( k^* \) values and the location of roughness at the blade as shown by previous studies [8,9].

CONCLUSIONS

In this paper, the effect of operational surface roughness on the overall compressor performance was studied numerically. The investigated HPC is part of a V2500-A1 aircraft engine.

An investigation of a fully rough HPC blading and in combination with doubled tip gaps representing geometry deterioration revealed that a super-positioning of induced geometry losses, and surface roughness is not possible. Further, the micro-scaled effects of surface roughness reduce the performance of about 2.5% and, thus, produce higher losses as are induced by macro-scaled geometry changes such as doubled tip gaps. The gap between the compressor map with the newly manufactured blading and the deteriorated HPC cannot be fully described. The significance of surface roughness could nevertheless be presented, but still should be further investigated.

Simulations conducted with stage roughness variations show that the first stage has the biggest effect on the compressor performance at both operation conditions. In contrast, the stall and choke margins are only reduced at ‘take-off’ due to the higher blockages based on the higher Reynolds numbers. Moreover, it is not possible to sum up the losses of each separately simulated stage in order to predict the losses of the combined rough stages.

Basically two important conclusions can be drawn:

1. Surface roughness in front stages enhances instabilities in the flow by changing the stage-by-stage matching, resulting in bigger overall performance losses than in rear stages.

2. Surface roughness influences the narrowing and shifting of compressor maps to a higher extent as geometry deterioration of the blading represented by doubled tip gaps.

ACKNOWLEDGMENT

The present work has been carried out in the subproject B3 within the Collaborative Research Center (CRC) 871 “Regeneration of Complex Capital Goods” which is funded by the DFG (Deutsche Forschungsgemeinschaft) under grant SFB 871. The authors would like to thank the DFG for the support. Moreover, the authors would like to acknowledge the substantial contribution of the DLR Institute of Propulsion Technology and MTU Aero Engines AG for providing TRACE. The results presented here were partially carried out on the cluster system at the Leibniz University IT Service (LUIS). Thus, the authors acknowledge the support of the cluster system team in the production of this work.

REFERENCES

[1] Bons, J. P., 2010, “A Review of Surface Roughness Effects in Gas Turbines”, Journal of Turbomachinery (132), p. 021004.

[2] Wensky, T., Winkler, L., and Friedrichs, J., 2010, “Environmental influences on engine performance degradation”, ASME Turbo Expo 2010: Power for Land, Sea, and Air, pp. 249–254.
[3] Gilge, P., Kellersmann, A., Friedrichs, J., and Seume, J. R., 2019, “Surface roughness of real operationally used compressor blade and blisk”, Proceedings of the Institution of Mechanical Engineers, Part G: Journal of Aerospace Engineering.

[4] Gilge, P., Kellersmann, A., Kurth, S., Herbst, F., Friedrichs, J., and Seume, J., 2019, “Dataset: Surface roughness of real operationally used compressor blade and blisk”, Research Data Repository of the Leibniz University of Hannover. Available at: https://doi.org/10.25835/0084372.

[5] Nikuradse, 1933, “Law of Flows in Rough Pipes”, NACA Technical Memorandum 1292.

[6] Schlichting, H., 1936, “Experimentelle Untersuchungen zum Rauhigkeitsproblem”, Archive of Applied Mechanics, Vol. 7(1), pp. 1–34.

[7] Morini, M., Pinelli, M., Spina, P. R., and Venturini, M., 2009, “CFD simulation of fouling on axial compressor stages”, Proceedings, ASME Turbo Expo 2009: Power for Land, Sea, and Air, American Society of Mechanical Engineers, pp. 331–342.

[8] Aldi, N., Morini, M., Pinelli, M., Spina, P. R., Suman, A., and Venturini, M., 2014, “Numerical analysis of the effects of surface roughness localization on the performance of an axial compressor stage”, Energy Procedia, Vol. 45, pp. 1057–1066.

[9] Gilge, P., and Mulleners, K., 2016, “Resulting Aerodynamic Losses of Combinations of Localized Roughness Patches on Turbine Blades”, AIAA Journal, Vol. 54(8), pp. 2552–2555.

[10] Gilge, P., Seume, J. R., and Mulleners, K., 2018, “Analysis of Local Roughness Combinations on the Aerodynamic Properties of a Compressor Blade”, Proceedings, 2018 AIAA Aerospace Sciences Meeting, American Institute of Aeronautics and Astronautics, p. 12.

[11] Schaffler, A., 1980, “Experimental and analytical investigation of the effects of Reynolds Number and blade surface roughness on multistage axial flow compressors”, Journal of Engineering for Power, Vol. 102(1), pp. 5–12.

[12] Leipoldt, 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”, Journal of turbomachinery, Vol. 122(3), pp. 416–424.

[13] Gbadebo, S. A., Hynes, T. P., and Cumpsty, N. A., 2004, “Influence of Surface Roughness on Three-Dimensional Separation in Axial Compressors”, Journal of Engineering for Gas Turbines and Power, Vol. 126(4), p. 455.

[14] K. Bammttert, G. W., 1980, “The Influence of the Blading Surface Roughness on the Aerodynamic Behavior and Characteristic of an Axial Compressor”, Journal of Engineering for Gas Turbines and Power, Vol. 102(2), pp. 283–287.

[15] Morini, M., Pinelli, M., Spina, P. R., and Venturini, M., 2011, “Numerical analysis of the effects of nonuniform surface roughness on compressor stage performance”, Journal of Engineering for Gas Turbines and Power, Vol. 133(7), p. 072402.

[16] Teng, X., Chu, W., Zhang, H., Liu, K., and Li, J., 2018, “The Influence of Geometry Deformation on a Multistage Compressor”, ASME Turbo Expo: Power for Land, Sea, and Air, Vol. Volume 2A: Turbomachinery.

[17] Reitz, G., Kellersmann, A., and Friedrichs, J., 2018, “Full High Pressure Compressor Investigations to Determine Aerodynamic Changes due to Deterioration”, Proceedings, ASME Turbo Expo 2018: Turbomachinery Technical Conference and Exposition, American Society of Mechanical Engineers. V02AT39A035.

[18] Wein, L., Seume, J. R., and Herbst, F., 2018, “Unsteady Flow in a Labyrinth Seal”, Proceedings, GPSS Forum Montreal 2018, Global Power and Propulsion Society.

[19] Sigal, A., and Danberg, J. E., 1990, “New correlation of roughness density effect on the turbulent boundary layer”, AIAA journal, Vol. 28(3), pp. 554–556.

[20] Kurz, R., and Brun, K., 2012, “ Fouling mechanisms in axial compressors”, Journal of engineering for gas turbines and power, Vol. 134(3), p. 032401.

[21] Franke, M., Kuegeler, E., and Nurnberger, D., 2005, “Das-DLR-Verfahren TRACE: Moderne Simulationstechniken für Turbosystemströmungen”, Proceedings, DGLR Congress.

[22] Nurnberger, D., 2004, “Implizite Zeitintegration für die Simulation von Turbosystemströmungen”, PhD thesis, Ruhr-Universität Bochum.

[23] Kuegeler, E., Nurnberger, D., Weber, A., and Engel, K., 2008, “Influence of blade fillets on the performance of a 15 stage gas turbine compressor”, Proceedings, ASME Turbo Expo 2008: Power for Land, Sea, and Air, American Society of Mechanical Engineers, pp. 415–424.

[24] TRACE User Guide, 2018 (accessed May 7, 2019), http://www.trace-portal.de/userguide/trace/index.html.

[25] Wilcox, D. C., 1988, “Reassessment of the scale-determining equation for advanced turbulence models”, AIAA journal, Vol. 26(11), pp. 1299–1310.

[26] Kato, M., and Lauder, B., 1993, “The modelling of turbulent flow around stationary and vibrating square cylinders”, In: Proceedings 9th Symposium on Turbulent Shear Flow, Kyoto, Japan, pp. 10.4.1–10.4.6.

[27] H. Tubbs, 1991, “Aerodynamic development of the high pressure compressor for the IAE V2500 aero engine”, In: Proceedings of the I Mech E European Conference.

[28] Reitz, G., Kellersmann, A., Schlange, S., and Friedrichs, J., 2018, “Comparison of sensitivities to geometrical properties of front and aft high pressure compressor stages”, CEAS Aeronautical Journal, Vol. 9(1), pp. 135–146.

[29] Tarabrin, A. P., Schurovsky, V. A., Bodrov, A. I., and Stalder, J.-P., Tuesday 2 June 1998, “Influence of Axial Compressor Fouling on Gas Turbine Unit Performance Based on Different Schemes and With Different Initial Parameters”, Proceedings, Volume 4: Heat Transfer; Electric Power; Industrial and Co-generation, ASME, p. V004T11A006.

[30] Syverud, E., Brekke, O., and Bakken, L. E., 2007, “Axial Compressor Deterioration Caused by Saltwater Ingestion”, Journal of Engineering for Gas Turbines and Power, Vol. 129(1), p. 119.

[31] Melino, F., Morini, M., Peretto, A., Pinelli, M., and Spina, P. R., 2012, “Compressor fouling modeling: relationship between computational roughness and gas turbine operation time”, Journal of Engineering for Gas Turbines and Power; Vol. 134(5), p. 052401.

[32] Melino, F., Peretto, A., and Spina, P. R., 2010, “Development and validation of a model for axial compressor fouling simulation”, Proceedings, ASME Turbo Expo 2010: Power for Land, Sea, and Air, American Society of Mechanical Engineers, pp. 87–98.

[33] Kellersmann, A., Weiler, S., Bode, C., Friedrichs, J., Stading, J., and Ramam, G., 2018, “Surface roughness impact on lowpressure turbine performance due to operational deterioration”, Journal of Engineering for Gas Turbines and Power, Vol. 140(6), p. 062601.

[34] Marx, J., Stading, J., Reitz, G., and Friedrichs, J., 2014, “Investigation and analysis of deterioration in high pressure compressors due to operation”, CEAS Aeronautical Journal, Vol. 5(4), pp. 515–525.