Experimental investigations on steady wake effects in a high-lift turbine cascade

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Bernd Stoffel, Andreas Fiala, Klaus Heinig

Abstract This paper reports on the investigation of steady wake effects in cascades. An annular cascade rig, where two stator rows having the same blade pitch can be circumferentially traversed relatively to each other, is used to analyse the profile losses and the boundary layer development of the downstream stator for different circumferential positions of the upstream stator ("clocking positions"). Different measurement techniques are used such as three-hole pressure probes, and hot-wire- and surface-mounted hot-film probes. The results show a varying pressure loss coefficient of the downstream cascade (S2) for different clocking positions of the upstream cascade (S1, SP).

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>t</td>
<td>Pitch, time</td>
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<tr>
<td>T</td>
<td>Measuring time</td>
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<tr>
<td>S</td>
<td>Surface length coordinate</td>
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<tr>
<td>S(f)</td>
<td>Detector function</td>
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<tr>
<td>S(2)f</td>
<td>Smoothed detector function</td>
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<tr>
<td>Sps</td>
<td>Total surface length</td>
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<tr>
<td>SS</td>
<td>Suction side</td>
</tr>
<tr>
<td>x</td>
<td>Circumferential coordinate</td>
</tr>
<tr>
<td>a</td>
<td>Flow angle</td>
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<tr>
<td>y</td>
<td>Intermittency</td>
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<td>œ</td>
<td>Loss coefficient</td>
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1 Introduction

The need to reduce weight and manufacturing costs of gas turbines and turbo engines leads to the demand for a reduction in the number of blades and therefore an increased loading of the individual blade. Today's state of the art turbine blades, so-called controlled diffusion airfoils, allow a small separation bubble on the blade surface. The understanding of the physics of such a separation bubble under the influence of incoming wakes is crucial to determine the profile losses and aerodynamic performance of the blades. In turbomachinery applications, wakes of upstream blade rows interact with the boundary layer of downstream blade rows, and wake-induced transition is the dominant transition mechanism besides bypass transition and transition in a separation bubble.

This paper is concerned with the experimental investigation of the loss mechanisms of high-lift turbine blades with a small separation bubble on the suction surface under the influence of incoming steady wakes. To investigate the influence of the upstream stator wake on the boundary layer of the downstream stator for different impingement points of the wake, two stator rows of an annular cascade test rig can be traversed circumferentially. Various authors have shown that this change in relative circumferential position of stator rows, called "clocking" or indexing", may have an impact on aerodynamic performance.

The first clocking investigations were carried out by Huber and Sharma (1995) as they investigated the loss behaviour of a three-stage low-pressure turbine with the same number of blades for all stator rows. For different circumferential positions of the stator blades in adjacent stages they measured a change in turbine efficiency of about ±0.5%. This efficiency variation by indexing the
stator rows is called the clocking effect. Griffin et al. (1995) performed numerical investigations and showed similar results.

In earlier experiments Binder et al. (1989) observed separations on some rotor blades in a five-stage low-pressure turbine. They showed that separation occurred when the choked wakes of the upstream rotor hit the leading edge of the second rotor blades and that separation was suppressed when the wakes convected through the rotor passage.

Ladwig (1991) used a cascade wind tunnel to investigate low-pressure turbine blades under steady state flow conditions. With the use of cylindrical bars steady wakes were generated, impinging on the turbine blades. Ladwig showed that by changing the point of impact of the cylinder wakes, the profile losses of the turbine blades changed. He proposed to use this effect to increase turbine efficiency. Fottner and Engel (1995) and Acton (1997) were able to confirm the results of Ladwig using a different turbine blade design.

Hailstead et al. (1997) showed in experimental and numerical investigations of a one and one half-stage turbine that the boundary layer development of the suction side of the second stator was strongly influenced by the relative position of the first stator. The wake-induced periodic transition caused by the rotor wakes was the dominant mechanism. If the periodically incoming wakes of the first stator hit the leading edge of the second stator blades, additional turbulence was transported into the boundary layer of the second stator. The velocity defect and unsteady pressure field also influenced the transition process.

Reimüller and Nihlén (2002) investigated the clocking effect in a one and one half-stage low-pressure turbine with blades of low aspect ratio. Stadtmüller et al. (2000) also showed positive boundary layer effects on high-lift turbine profiles.

Haldeman et al. (2003) performed steady and unsteady pressure measurements in a one and one half-stage turbine to investigate the influence of vane clocking and vane/blade spacing. They showed that the clocking position of the HPT vane has some effect on the time-averaged surface pressure distributions for the transition duct and LPT vane. For the time-accurate pressure distributions, the effect was more apparent.

Due to improved calculation models, numerical simulations of the clocking effect have gained increasing significance. Besides previous work by Dorney and Sharma (1996) and Eulitz and Engel (1998), Dorney et al. (2001) give a good overview of numerical clocking investigations. Some other numerical works with detailed information about the turbulence modelling is given by Höhn and Heinig (2000) and Breitbach (2001). The clocking effect is also observed in axial compressors. A good overview of compressor clocking investigations is given by Dorney et al. (1999), Dorney and Sharma (1996) and Walker et al. (1998).

A quasi-three-dimensional, blade-to-blade, time-accurate, viscous solver was used by Marconcin et al. (2003) for the clocking optimization of a modern transonic two-stage gas turbine. They found efficiency gain contributions by axial gap optimization and clocking, both comparable for the analysed configuration.

Arnold et al. (2003) investigated a one and one half-stage low-pressure turbine using a three-dimensional, time-accurate, viscous solver. They studied the influence of clocking on efficiency variations, wake interaction patterns and flow unsteadiness. The three-dimensional analysis showed that the greatest benefits were achieved when the wake impinged on the second stator leading edge, while an efficiency drop was observed if the wake path entered the mid-channel.

To provide a better understanding of the wake-boundary layer interaction, the physics of the clocking effect and its benefit on turbomachinery performance, a detailed experimental investigation in a high-lift turbine cascade was carried out in this study. The results of this basic investigation can help to understand the unsteady complex flows occurring in turbomachinery.

2 Experimental investigation

2.1 Description of the test facility

Figure 1 shows the experimental set-up with its aerodynamic details. The second stator is the object to

![Fig. 1. Aerodynamic design](image-url)
investigate. This profile is a state-of-the-art high-lift low-pressure turbine airfoil which, under design flow conditions, has a laminar separation bubble on the suction side blade. To investigate the influence of clocking on the second stator, the first stator can be moved circumferentially. The blade design leads to a nearly two-dimensional flow and felt seals are applied between the parts that are movable relative to each other to avoid leakage flows. Table 1 gives a summary of all the test rig parameters. For the design flow conditions the flow can be considered incompressible.

Figure 2 shows the vertical test section of the experimental facility. The flow enters the test section from below after passing a settling chamber. The whole test rig consists of several rings, one on top of the other. The stator 2 blade row (16) is fixed in the inner ring (17), while the stator 1 vanes (14) are fixed in the outer ring (6), which can be circumferentially traversed by a motor (13).

To move the probes (8), (9) in a circumferential direction, the ring (10) can be traversed by the motor (15).

The 55 kW water-cooled four-quadrant electromotor (12) and the emergency break (18) are needed in case there is a rotor between the two stators which will be added and used for further investigations. The details about the experimental set-up and the installed equipment are described in great detail by Heinke (2002).

2.2 Measurement techniques

Figure 3 gives an overview of the measurement planes and techniques applied in each plane. Three-hole pressure probes were used to measure the flow angle and the total pressure behind the second stator. To analyse the static pressure distribution, 21 static pressure tappings were evenly spaced at a distance of 2 mm on the suction side of a blade and 19 static pressure tappings on the pressure side of the adjoining blade. Hot wire probes were used to measure the unsteadiness in the flow field and surface-mounted hot films to investigate the boundary layer behaviour along the stator 2 vanes. Four different blades were equipped with the surface-mounted hot-film probes: two on the suction side and two on the pressure side. On the suction side, 19 sensors were applied on each blade, and on the pressure side the two blades were equipped with 17 and 16 sensors, respectively. To get a higher spatial resolution than the axial spacing of 2.5 mm, the sensors on the two different suction side blades were shifted by half their axial spacing. The same procedure was used for the pressure sides to get an overall spatial resolution of 1.25 mm.

All the pressure measurements were performed using ScaniValve equipment. The pressure sensor was a PDCR22 with a linearity of 0.04% and a pressure range of 70 mbar. A 12-bit A/D converter and a computer with a Pentium II processor were used to carry out the data acquisition.

The hot-wire measurements were carried out using Dantec Streamline equipment and a 55P13 single hot-wire

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Parameter</th>
<th>Value</th>
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</thead>
<tbody>
<tr>
<td>Number of blades vane 1 (S2)</td>
<td>66</td>
<td>Deflection angle vane 2</td>
<td>69.3°</td>
</tr>
<tr>
<td>Number of blades vane 2 (S1_sp)</td>
<td>66</td>
<td>Inlet velocity c_in</td>
<td>47.1 m/s</td>
</tr>
<tr>
<td>Angle of attack α1, st</td>
<td>90°</td>
<td>Exit Mach number vane 2</td>
<td>0.271</td>
</tr>
<tr>
<td>Exit angle α1, st</td>
<td>80.7°</td>
<td>Mass flow</td>
<td>13 kg/s</td>
</tr>
<tr>
<td>Angle of attack α2, st</td>
<td>90°</td>
<td>Inner diameter D1</td>
<td>677.5 mm</td>
</tr>
<tr>
<td>Exit angle α2, st</td>
<td>150°</td>
<td>Outer diameter D2</td>
<td>881 mm</td>
</tr>
<tr>
<td>Deflection angle vane 1</td>
<td>9.3°</td>
<td>Re-number vane 2</td>
<td>2.1×10⁶</td>
</tr>
<tr>
<td>Turbulence intensity at inlet</td>
<td>2.5%</td>
<td></td>
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probe. An external TC6 temperature probe was used to compensate for temperature variations. The voltage signal was converted using a 12-bit A/D converter. The calibration was carried out in a separate calibration tunnel.

To determine the total pressure losses, the pressure probes were traversed in plane 1.1 and 2.4 to measure the losses of the complete configuration and in plane 1.1 and 2.1 to measure the losses of the stator 1 vanes alone. To eliminate any influence of mass flow deviations, all pressure measurements were carried out simultaneously. The inlet total pressure probe was traversed at the same time on the streamline assumed to cross the outlet total pressure probe but shifted parallel by one blade pitch in order to avoid effects of the wake of the upstream probe on the downstream probe. The pressure difference was measured directly by the pressure sensor, whereby the inlet total pressure was connected to the reference input and the outlet total pressure was connected to the ScaniValve revoler.

The same procedure was used to determine the dimensionless Cp value of the stator 2 vanes (Cp=P(x)/Ptot).

Equations 1 and 2 show the evaluation of the total pressure measurements (the bar indicates the mean value over one airfoil pitch) and Eq. 3 defines the dimensionless static pressure distribution on the vane surface. Equations 4, 5, and 6 show the evaluation of the hot-wire raw data. The surface-mounted hot-film probes were used to determine a measure of the time-averaged wall shear stress and the part of the fluctuating wall shear stress. \( \bar{E}_0 \) indicates the time-averaged anemometer output voltage measured in the absence of flow and is needed to calculate the relative wall shear stress \( (E-E_0)/E_0 \) and the fluctuating wall shear stress.

The surface-mounted hot films were also used to determine the intermittency \( \gamma \). The intermittency gives the probability of whether the flow at a certain position is laminar or turbulent, where an intermittency value of 1.0 defines a fully turbulent state and an intermittency value of 0.0 a fully laminar state. To obtain the intermittency values presented in this study, the second derivation of the anemometer output voltage is squared to obtain the detector function \( S(t) \) given in Eq. 7. To eliminate the disturbing effects of voltage peaks, a smoothing procedure is applied to the \( S(t) \) signal: The mean value of ten consecutive \( S(t) \) values is calculated and the ten values are substituted by their mean value \( S^*(t) \). By means of a threshold \( G \), Eq. 8 determines whether the calculated \( S^*(t) \) values indicate turbulent fluctuations or signal noise. Equation 8 leads to a 0/1-distribution with 0-values presenting laminar and 1-values presenting turbulent portions of the flow. The integration of the function \( I(t) \) over the measuring period \( T \) leads to the intermittency \( \gamma \) (Eq. 9).

\[
\begin{align*}
\bar{c} &= \frac{1}{n} \sum_{i=1}^{n} c_i \\
k_1 &= \frac{3}{2n} \sum_{i=1}^{n} (\bar{c} - c_i)^2 \\
k_1 \text{ges} &= \frac{1}{2n} \int \frac{k_1 \text{ dx}}{k_1} \\
S(t) &= \frac{\partial^2 E(t)}{\partial t^2} \\
l(t) &= \begin{cases} 1, & S^*(t) \geq G \\ 0, & S^*(t) < G \end{cases} \\
\gamma &= \frac{1}{T} \int_{t=0}^{T} l(t) \text{dt}
\end{align*}
\]

Detailed information about the intermittency calculation is given by Chakka and Schobeiri (1999), and further details about the smoothing procedure are presented in Heinke (2002).

3 Experimental results

In this section, the influence of different circumferential positions (clocking positions, clp) of the two stators on the flow field and the boundary layer of the second stator are investigated experimentally. The clocking position “0” indicates the relative position where the S1 SP wake hits the leading edge of the S2 blade (cf. Fig. 4). Impingement points where the S1 SP wake is shifted toward the suction side of S2 are labelled with positive values and impingement points shifted toward the pressure side of S2 blade with negative values. Nine different clocking positions are evenly spaced over one airfoil pitch, and one additional

\[
\begin{align*}
\omega_{\text{ok ges}} &= \frac{P_{\text{tot 2.1}}(x) - P_{\text{tot 2.4}}(x)}{P_{\text{tot 2.4}} - P_{\text{tot 2.4}}} \\
\omega_{\text{gesc}} &= \frac{1}{T} \int \omega_{\text{ok ges}} \text{ dx} \\
C_p(S/S_{\text{ges}}) &= \frac{P(S/S_{\text{ges}})}{P_{\text{tot 1.1}}}
\end{align*}
\]

Fig. 4. Definition of the clocking position
clocking position is introduced for the case when the impingement point of the stator 1 wake is shifted 6.6% toward the suction side of stator 2 (clp 0.066).

The wake characteristics of the inlet guide vane represent the primary parameter of the problem and are shown in Figs. 5, 6 and 7 for two different clocking positions. Since no potential effect of the stator 2 vane on the stator 1 wake is expected, the wake characteristics of the inlet guide vane should remain similar for all clocking positions. Deviations are due to measurement inaccuracy. In Figs. 5, 6 and 7, the probes have been traversed at midspan over one airfoil pitch from left to right with respect to Fig. 4. Negative values of x/t indicate probe positions shifted toward the suction side and positive values indicate probe positions shifted toward the pressure side, respectively. The measuring plane is located 60 mm downstream of the stator 1 trailing edge and indicated in Fig. 3 as station 2.1. One parameter which influences the wake behaviour is the relative trailing edge thickness to chord ratio, which has a value of about 0.011. In Fig. 5 the non-dimensionalized velocity deficit downstream of stator 1 can be seen. The small asymmetry of the curve is caused by the slightly thicker suction side boundary layer. Figure 6 shows the typical distribution of the turbulent kinetic energy with a dent at the position of maximum velocity deficit. Higher turbulence values are observed on the suction side. The flow angle downstream of stator 1 is shown in Fig. 7, where a mean flow angle of about 80° can be observed. The changes due to the clocking positions are caused by measurement inaccuracy. The reason for the higher deflection angle (smaller value of $\alpha_{SS}$) on the suction side are transport processes behind the profile, which is typical for airfoils. For different clocking positions, the stator 1 wake has the same wake characteristics but different impingement points on the stator 2 wake. This influence will be discussed in the following discussion.

In Figs. 8 and 9, the local pressure loss coefficient $c_{\text{p,edges}}$ is plotted for different clocking positions against x/t. x refers to the circumferential probe position downstream of stator 2, and t indicates the airfoil pitch at midspan. The probe traverse was carried out over one airfoil pitch from x/t = -0.5 to x/t=0.5. x/t=0.0 indicates the position where the probe is positioned in the wake of stator 2 for clp 0.0. At clp 0.5 the wake of S1_SP passes in

![Fig. 5. Stator 1 wake characteristics: flow velocity](image1)

![Fig. 6. Stator 1 wake characteristics: turbulent kinetic energy](image2)

![Fig. 7. Stator 1 wake characteristics: flow angle](image3)

![Fig. 8. Local pressure loss coefficient depending on the relative position of stator 1 (positive clp)](image4)
Fig. 9. Local pressure loss coefficient depending on the relative position of stator 1 (negative clp)

the middle between two S2 vanes, resulting in a higher pressure loss for the traverse positions $x/t = \pm 0.5$. In Fig. 8, the clocking positions where the stator 1 wake impinges on the suction side of stator 2 (positive clp) are shown and a decrease in pressure loss relative to clp 0.5 can be observed in the range $x/t = 0.2$ to $x/t = 0.5$, whereas the maximum pressure loss occurring around $x/t = 0.0$ increases. At clp 0.06 and clp 0.0 the interaction of the S1 SP wake with the pressure side of S2 is clearly visible between $x/t = -0.3$ and $x/t = -0.1$. In Fig. 9 (negative clp), the strongest interaction between the V1 SP wake and the V2 pressure side can be seen for clp -0.125. The change in V2 outlet deflection angle (see below) results in a shift of the maximum pressure loss coefficient toward the suction side.

An integration of each local pressure loss coefficient curve in Figs. 8 and 9 leads to the integral total pressure $\omega_{ps}$. In Fig. 10, the integral total pressure loss coefficient of stator 2 (S2) $\omega_{ps} \cdot S2$ is plotted against the clocking position. To evaluate the influence of mass flow deviations and measurement inaccuracy, four measurements were carried out. The thin lines indicate the maximum and minimum deviation from the mean value for the four measurements. The greatest deviation occurs for clp 0.25 and has a value of 1.6% with reference to the mean value. Considering calibration errors, nonlinearities of the pressure scanner and errors of the A/D-board, the absolute error of the total pressure loss of the stator 2 blade has a value of about 1.7%. A minimum of pressure loss occurs when the impingement point of the stator 1 wake is shifted by 6% of the pitch towards the suction side of stator 2 (clocking position clp 0.06). The maximum pressure loss is observed at the clocking position clp -0.25, when the impingement point of the S1 SP wake is shifted by 25% to the pressure side of the stator 2 blade.

Figure 11 shows the pitch-averaged exit angle of stator 2 plotted against the clocking position. Deviations of the four measurements are indicated by the thin lines around the mean value. For clp -0.375, the greatest error bar can be seen and the deviation with respect to the mean value is about 0.03%. With respect to the change in flow angle for different clocking positions, the relative error is in the order of 34%. Due to errors in calibration and installation of the probe the inaccuracy of the absolute flow angle has a value of about ±0.5°. Since the changes in flow angle caused by clocking are of interest, the absolute flow angle is not the main focus. The differences in mean value for clp 0.5 and clp -0.5 are due to measurement inaccuracy, as can be seen in the error bar. The minimum and maximum values of the exit angle correlate with the minimum and maximum total pressure loss in Fig. 10. It can be seen that the variation in stator 2 deflection angle for different clocking positions is about 0.2°. The position of maximum deflection angle equals the position of minimum pressure loss and vice versa.

Figure 12 shows the pitch-averaged turbulent kinetic energy downstream stator 2 plotted against the clocking position. Since the maximum of turbulent kinetic energy indicates higher unsteadiness, higher losses would be assumed. However, the maxima/minima of k1 and losses do not coincide but are close to each other. The reason could be the use of a single hot wire, but there could also be a physical reason. The value k1 only represents the fluctuations in streamwise direction and is therefore not the...
exact value for the turbulent kinetic energy. The relative deviation between different measurements is of the order 1.3%.

Figure 13 shows the static pressure distribution of the stator 2 blade for the two clocking positions of minimum and loss maximum plotted against the dimensionless surface coordinate $S/S_{ges}$. It is clear that the pressure side distribution (top) shows no significant change with varying clocking position except near the leading edge resulting from a small variation of the incidence angle with the wake impingement position.

On the suction side a laminar separation bubble exists at about $S/S_{ges}=0.5$. It can be seen that at clocking position $clp=0.06$, the position of minimum total pressure loss, the separation bubble decreases in size. This is caused by the interaction of the stator 1 wake with the stator 2 suction side whereby the wake turbulence forces the separated shear layer to reattach earlier. At clocking position $clp=-0.25$, the position of maximum total pressure loss, no interaction between the stator 1 wake and the stator 2 suction side takes place. At this position, because of the relatively low turbulence level affecting the suction side, the separated shear layer reattaches somewhat downstream, resulting in a bubble increase. This change in bubble size is a relevant contribution to the change in total pressure loss. Another important source of losses is the development of the boundary layer which is influenced by the change in incidence angle. The measurement inaccuracies of the absolute $C_p$ value is about 0.1%.

Figure 14 shows the relative wall shear stress on the suction side of the stator 2 blade for the clocking positions $clp=0.06$ and $clp=0.25$ corresponding to the extreme loss values. Minimum values of quasi-shear stress correspond to zero values of the wall shear stress since the measured voltages are always positive. The strong drop in shear stress indicates the beginning of separation, and the increase after the separation bubble shows reattachment of the flow. The position of the separation bubble agrees well with the position assumed in the design process. It is obvious that no change in transition behaviour takes place. For both clocking positions, laminar to turbulent transition takes place via a laminar separation bubble. The change in bubble size is visible at $S/S_{ges}=0.5$. Deviations in quasi-shear stress caused by measurement inaccuracies are of the order 1.5%.

The intermittency calculations shown in Fig. 15 indicate that the boundary layer on the pressure side stays laminar for all clocking positions.

Derived from the surface-mounted hot-film measurements and the static pressure distribution, the length of the bubble changes by about 30% between the minimum and the maximum value. As already mentioned, a decreasing pressure loss coefficient corresponds to an increase in deflection angle of stator 2 (Figs. 10 and 11).

The question that remains to be answered is: what is the reason for the change in turbulence intensity? The answer can be found in the intermittency calculations for the stator 2 suction side, shown in Fig. 16 for the two clocking positions.
4 Conclusions

Experimental investigations of steady stator-stator interactions were carried out to analyse the boundary layer behaviour of a high-lift turbine blade subjected to incoming wakes for different circumferential positions of the two stators (clocking positions). The experimental results show a change in total pressure loss coefficient of the stator 2 blade row of about 11% between the minimum and the maximum value, depending on the clocking position. The minimum occurs if the impingement point of the stator 1 wake on the stator 2 blade is shifted by 6% of the pitch toward the suction side of S2. The loss maximum occurs if the impingement point is shifted by 25% of the pitch toward the pressure side of the stator 2 blade. The development of the total pressure loss minimum could be explained by the interaction of the S2 suction side boundary layer with the stator 1 wake. This interaction results in:

- A decrease in the size of the laminar separation bubble of about 30%,
- A reduction of the turbulence intensity downstream stator 2 of nearly 30%,
- An increase in stator 2 deflection angle of about 0.2°.

References


Fig. 15. Intermittency distribution on pressure side of stator 2

Fig. 16. Intermittency distribution on suction side of stator 2

positions $clp$ 0.06 (loss minimum) and $clp$ 0.25 (loss maximum).

Intermittency values near zero indicate a laminar boundary layer, and values near one indicate a turbulent boundary layer. For both clocking positions, the leading edge suction side boundary layer is laminar with values near zero. Laminar to turbulent transition takes place at the position $S/S_{ref}=0.45$, which is in good agreement with the static pressure distribution in Fig. 13. Downstream of the laminar separation bubble, both plots show a decrease in intermittency values. It clearly can be seen that at the position of minimum pressure loss the intermittency values tend to be smaller. These values are no indication for lower turbulence levels, but the absence of a threshold change for different spatial locations may be the root of the intermittency values between 60 and 90% close to the trailing edge.