Variable Stator Vane Penny Gap Aerodynamic Measurements and Numerical Analysis in an Annular Cascade Wind Tunnel
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ABSTRACT
This paper presents detailed measurements and post-test simulations of the penny cavity leakage flow and its interaction with the mainstream flow in an annular cascade wind tunnel. The annular cascade wind tunnel consists of a single row of 30 variable stator vanes, derived from a high-pressure compressor stator with inner and outer vane disks, called pennies, which - when assembled in the hub and casing walls - leave cylindrical-shaped ring gaps called penny cavities. The wind tunnel runs at a Mach number of 0.34 at the stator inlet and a Reynolds number of 3.82 x 10⁶ based on axial chord length at 50% span.

Two different penny gap sizes on the hub are compared to a reference case without a penny gap. Detailed 2D-traverses were performed with multi-hole pressure and hot-wire-probes covering 2.5 passages in the inflow and outflow of the stator row. Pressure taps were embedded in the airfoil surface and inside the penny cavity. Surface oil flow measurements were conducted with different colors for the vane suction side, pressure side, hub and the penny cavity to detect the secondary flow phenomena. Reynolds-averaged Navier-Stokes (RANS) simulations, using the measured boundary conditions, were compared to experimental data.

As a result, a relative increase in the total pressure loss coefficient of 1.9% for the nominal and 6.8% for the double penny gap was measured compared to no-penny cavity. The additional penny losses are limited to the lower 40% span. The post-test simulations are in good agreement with the measurements, showing that the outflow from the penny cavity on the suction side generates vortices, which cause additional losses. The penny vortices are detected in the outlet plane by an increase in turbulence intensity and streamwise vorticity. However, the additional penny losses are underestimated in the simulation by up to 7.3%. A change in the pressure fields with an increasing penny gap size, both around the airfoil and inside the penny cavity, can be seen in the numerical and experimental results. The outflow regions of the penny cavity, estimated by simulations, are confirmed by the results of the surface oil flow measurements.

In summary, this paper consolidates previous numerical analyses carried out by the authors [13-16] on penny cavity leakage flow effects with experimental data for different penny gap sizes.

Keywords: penny, variable guide vane, cavities, button, stator, HPC, leakage flow, cascade, compressor

Presented at International Gas Turbine Congress 2019 Tokyo, November 17-22, Tokyo, Japan
Review Completed on March 19, 2020

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between the shaft of each stator vane and the casing and is driven by the pressure difference between the primary gas path and the ambiences. The losses due to pivot leakage are reduced by using seals [4, 5].

The clearance leakage occurs at the radial gap between the vane and the endwall contours of hub and casing, leaving a hub and tip clearance. The pressure difference between the pressure and suction side of the vane causes unwanted gas recirculation through the radial gap. The tip clearance of VSVs is a half-gap, as it is interrupted by the vane pivot. The length of the radial gap on the hub depends on whether the stator vane is cantilevered or shrouded [6]. VSVs for both designs exist, as shown, for example, by Escurèt [7]. The hub clearance of cantilevered stator vanes and the development of the hub clearance vortex have been investigated, for example, by Lange [8]. Shrouded VSVs are mounted on the hub on an inner ring. A half-gap remains between the vane and the hub.

Pennies at the transition between pivot and vane are widely used. Pennies reduce the length of the half-gap. A new penny design, which has no radial gap, was proposed by Gottschall [9]. Escurèt [7] investigated the effect of pennies at the tip, which extend into the passage for about 0.5% of the vane height. Five different positions of the penny relative to the vane were investigated by Gottschall [10-12] in a linear cascade. The design with the half-gap at the rear of the vane and the penny at the front was favored, because high pressure differences at the vane front are avoided.

VSVs with penny have an additional small cylindrical-shaped ring gap between the penny and the endwall, called the penny cavity. The penny cavity cannot be sealed due to the assembly process, temperature issues and the limited installation space [15]. The penny leakage is, for the first time, the objective of a research project between the Institute of Jet Propulsion and Turbomachinery at RWTH Aachen University and MTU Aero Engines AG (MTU). The VSVs examined in this research project are shrouded and the penny is positioned in the front of the vane. The focus lies on the hub penny cavity. The sensitivity of the penny leakage to the gap size is investigated. A shroud leakage, which was investigated by Mattingly [3], is not taken into account. Results of the CFD studies about the penny leakage have already been published [13-16] as part of this research project. This paper is the first publication in which experimental data on penny leakage is reported.

The first time the impact of the penny cavity flow on the stator main flow was numerically investigated was by Wolf [13]. Wolf used the annular cascade wind tunnel geometry, RANS with k-ω turbulence model and a combined structured and unstructured mesh. Wolf showed the inflow and outflow regions of the cavity and highlighted the fact that the penny leakage is driven by the pressure difference between the pressure side and the suction side of the vane. Two main inflow regions, one at the leading edge and the other one at the vane pressure side, were described. The outflow occurs at two regions and forms a vortex which interacts with the passage vortex and lies next to the half-gap vortex. Wolf calculated an increase of 3.5% in total pressure loss coefficient for the nominal penny cavity compared to no-penny cavity.

Following on from this, Stummann [14] presented simulations of the hub penny leakage with turbulence resolving Detached-Eddy Simulations (DES) and compared them to RANS simulations. Stummann described in detail how the penny leakage from the suction side outflow region forms an asymmetrical counter-rotating vortex pair. The loss production caused by the secondary flow of the penny leakage is explained in the study. The nominal penny produced additional loss in the secondary flow region in the magnitude of 5.8% compared to when there is no-penny cavity. RANS overestimated the losses compared to DES by 2% in the total pressure loss coefficient.

In his PhD thesis Wolf [15] compared RANS simulations of the annular cascade wind tunnel stator with penny cavity to DES simulations. Wolf showed that RANS, compared to DES, reproduces the penny leakage more diffusely, but that it reproduces the principal effects sufficiently well. A meshing and adaptation concept to use standard RANS for the simulation of penny cavities in industrial compressor design was developed and calibrated with DES simulations. With an adapted sweep profile, Wolf was able to compensate for the reduction in stability caused by the penny leakage.

As part of his study, Stummann [16] provided an in-depth understanding of penny cavity flow and loss generation by applying scale-resolving DES calculations with penny cavity. Within the penny cavity, the two incoming mass flows are directed to the suction side, where the largest part of the leakage, 84% of the cavity mass flow, escapes. With an impulse ratio of 0.6 between penny leakage and main flow, the suction-side outflow forms the suction-sided and pressure-sided penny-vortex. The pressure-sided penny-vortex is sucked in by the cross-channel flow. The penny leakage causes 10% higher losses than when without penny. 99% of the losses are due to mixing and 90% come from the suction side. Stummann also investigated the influence of incidence and stagger angle on penny leakage and loss development. If the incidence is positive, the leakage increases due to the larger pressure gradient on the vane. Applied to a multi-stage compressor, Stummann found that penny leakage reduces efficiency by -0.4% and stall margin by -2.3%.

The objective of this paper is to validate the numerical results from the secondary flow and losses caused by penny leakage flow with experimental results. In the first instance, the three penny cavities investigated on the hub are described in detail: no-penny, nominal penny and double penny. The annular cascade wind tunnel of the Institute of Jet Propulsion and Turbomachinery at the RWTH Aachen University, which was reconstructed for the penny project, is described with all measuring planes and techniques. The numerical setup to be validated, in which experimental entry conditions are used for better matching, is also described. After an explanation of the methods, the experimental results are then compared to the numerical results for three penny configurations on different measurement planes.

PENNY CA VITY GEOMETRIES INVESTIGATED

Figure 1 shows the lower half of the VSV, the hub and the nominal penny cavity as applied to the annular cascade wind tunnel. Table 1 provides details of the geometry of the vane and penny cavity.

The design and arrangement of the penny cavity correspond to the front stages of a typical high-pressure compressor. The vane profile was adapted to the smaller mass flow at the annular cascade wind tunnel compared to the compressor. In order to increase the measurability, the VSV was scaled up, while keeping the geometry representative in the hub region. The stator array consists of 30 vanes.

The stator vanes have a cylindrical penny located at the front of the vane. The adjustable vane is guided via spindles within the hub and casing. The spindle diameter is only 40.5% of the penny diameter and set with a close fit in the inner ring. The length of the half-gap in the rear part of the vane covers 45.3% of the chord length. The ratio of the height of the half-gap to the chord length is representative of a typical compressor.

The penny cavity is highlighted in red in Fig. 1 and can be divided into an annular gap and a bottom gap. The cylindrical annular gap is between the hub and the penny and concentric to the penny. The bottom gap is below the penny, where it is defined by the hub and the spindle.
The experimental investigations described in this paper were carried out on the annular cascade wind tunnel of the Institute of Jet Propulsion and Turbomachinery at RWTH Aachen University. A settling chamber was placed in front of the wind tunnel. A cross-sectional view of the annular cascade wind tunnel is shown in Fig. 2. The front part of the hub was held by four struts, which were equally spaced, set apart 90° circumferentially. Downstream, a honeycomb and a turbulence generating sieve homogenized the wakes of the struts. A variable inlet guide vane (VIGV) with 48 adjustable vanes, which can be continuously adjusted via a motor and unison ring, provided the desired circumferential flow angle for the subsequent measuring section.

The central modules of the wind tunnel, which were redesigned and remanufactured for the penny project, began downstream of the VIGV. The stator row with 30 variable stator vanes (VSVs) and hub-side penny cavities (as described in Fig. 1 and Table 1), the contoured hub section and the casing with the inlet and outlet measurement planes were all newly manufactured. Each of the VSVs can be individually restaggered via a replaceable alignment plate within the casing. This paper only presents the results for the nominal stagger angle. As can be seen in Fig. 1, the inner shroud cavity that would normally be found in a compressor was not reproduced in the wind tunnel.

The stator row is an array with three different penny configurations, as listed in Table 2, each with 10 vanes. The stator row assembly, with its casing shroud, was mounted on the rear part of the hub, which was rotatable. The rotation takes place via the VSV servo and a bevel gear inside the hub. This allowed the module with VSVs and the rear hub contour to be rotated through 360°, so that every vane could be brought into the same measuring position. Thus, errors due to inflow asymmetries were avoided because the same inflow is always measured. The asymmetrical inflow in the wind tunnel originates from the incompletely mixed wakes of the struts and the VIGVs. Furthermore, there were no measurement errors due to rebuilding. The measurement uncertainty of the stator circumferential adjustment is in the range of 0.8% of the pitch. A numerical study confirmed that only the border vanes were influenced by the adjacent penny, so that measurements can be carried out on the remaining eight vanes.

The specifications of the two measurement planes are listed in Table 3. Downstream of the VIGV, the annulus has a constant channel height up to the inlet plane. The inlet plane was located 7.5 times the axial chord length behind the trailing edge of the VIGV, so that most of the VIGV’s wakes and endwall vortices were mixed out at the inlet plane. From the inlet plane onward, the pitch of the hub contour adjusts to add contraction into the flow path, as would be found in a similar compressor stage. At the inlet and outlet planes were access slots through which probes can be traversed in circumferential direction over 2.5 pitches and radially from hub to casing. As shown in Table 3, the measurement slots are offset by 23° relative to each other. The offset was determined in a numerical pretest, taking into account the flow of the gas through the stator array. This ensures that the same streamlines are measured in both measuring planes. Measured characteristic wakes in both planes confirm the offset, as can be seen later.

### EXPERIMENTAL SETUP

The experimental investigations described in this paper differ only in the size of the annular gap and are listed in Table 2. In the experiment, three penny configurations were examined: no-penny, nominal penny and double penny. In the simulations, two additional pennies with enlarged gaps were investigated. For the enlargement of the annular gap, the penny diameter was kept constant and the borehole in the hub was enlarged. The reference configuration without the penny was realized in the experiment by a tight fit, so that most of the VIGV’s wakes and endwall vortices can be individually restaggered via a replaceable alignment plate in the casing. This paper only presents the results for the nominal stagger angle. As can be seen in Fig. 1, the inner shroud cavity that would normally be found in a compressor was not reproduced in the wind tunnel.

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### Table 1 Variable stator vane geometry

| half-gap length / chord length | 45.3% |
|--------------------------------|-------|
| half-gap height / chord length | 3.9% - 4.3% |
| penny diameter / vane pitch at hub | 83.5% |
| penny diameter / chord length | 56.0% |
| spindle diameter / penny diameter | 40.5% |
| vane count | 30 |

The five penny configurations presented in this paper differ only in the size of the annular gap and are listed in Table 2. In the experiment, three penny configurations were examined: no-penny, nominal penny and double penny. In the simulations, two additional pennies with enlarged gaps were investigated. For the enlargement of the annular gap, the penny diameter was kept constant and the borehole in the hub was enlarged. The reference configuration without the penny was realized in the experiment by a tight fit, so that the remaining gap is between 0.009% and 0.017% of the chord length. The tight fit ensures that the vane is adjustable for further investigations, while no significant penny leakage is expected. The bottom gap was not changed in any of the penny configurations. The fit between the inner ring and spindle is between 0.015% and 0.028% of the chord length. The tight fit ensures that the vane is adjustable for further investigations, while no significant penny leakage is expected. The bottom gap was not changed in any of the penny configurations.

### Table 2 Hub penny cavity geometries investigated (*only RANS*)

| penny      | no         | nominal (P0) | nominal enlarged* (P1) | double (P2) | double enlarged* |
|------------|------------|--------------|------------------------|-------------|-----------------|
| annular gap | 0%         | 100%         | 155%                   | 200%        | 310%            |
| bottom gap | constant   | constant     | constant               | constant    | constant        |

### Table 3 Annular cascade wind tunnel specifications at the two measurement planes for probe traversing

| inlet plane | outlet plane |
|-------------|-------------|
| axial distance to vane LE/TE | axial chord length |
| -8° to +22° | -8° to +22° |
| circumferential position | uniform |
| +15° to +45° | adapted |
| points in the measuring grid | 544 |
| type of grid | 1856 |

Constant entry conditions were set for the project at the inlet plane at 50% channel height: these are listed in Table 4. The entry conditions were measured with three stationary 5-hole-probes in the inlet plane and were arithmetically averaged. The 5-hole-probes were arranged at a distance of 122.5° and -115° in the circumferential direction so that they do not correspond to a multiple of the VIGV pitch of 7.5°. Also, the stationary 5-hole-probes were at least 3 vane pitches away from the measurement region in the inlet plane from -8° to +22° to avoid interference of the measurements. For this project, the stagger angle of the VIGV was set to an incident-free inflow of the following stator.
2.9% of the pitch in circumferential direction and 2.3% span in circumferentially and radially. The finest spatial resolution was capture the wakes from the penny leakage and refined grid in the outlet plane, consisting of 1856 points, was adapted to the wall. The measuring grids of the inlet plane according to the mean yaw angle of 40°. The measuring grids of the 5 casing were measured with a 3 radius near the hub and placed any closer than 8% span to the hub and the casing wall. In addition, traverses over 2.5 passages from hub to casing with 3 radii, which were circumferentially uniformly distributed and can be seen in Fig. 5. The core area of the measuring grid was traversed by a 5-hole-probe. The 5-hole-probe used had a head diameter of 2.8mm. The 5-hole-probe was not placed any closer than 8% span to the hub and the casing wall in order to avoid measurement error. The three radii near the hub and casing were measured with a 3-hole-probe. Since its head height was only 0.7mm, the 3-hole-probe could be moved up to 2% span to the wall. The measuring grids of the 5-hole-probe and the 3-hole-probe overlapped each other. Since the triple-hot-wire-probe had similar dimensions as the 5-hole-probe, both used the same measuring grid. All probes were preset in the inlet plane according to the mean yaw angle of 40°. The measuring grid in the outlet plane, consisting of 1856 points, was adapted to capture the wakes from the penny leakage and refined circumferentially and radially. The finest spatial resolution was 2.9% of the pitch in circumferential direction and 2.3% span in radial direction.

All probes were calibrated in the institute’s free stream calibration channel mounted in the probe traverse devices to reduce measurement errors. The probes were calibrated in a yaw angle range from -30° to +30°, a pitch angle range from -15° to +40°, and a Mach number range from 0.1 to 0.45. A multi-parameter approximation, as suggested by Gallus [18], was used to evaluate the measurement data of the pneumatic probes. Since the outlet plane is only 13% of the axial chord length behind the trailing edge, particularly large velocity gradients were measured in the wake. In order to eliminate the error caused by the spatial displacement of the probe's pressure taps, a correction method for the data of the pneumatic multi-hole-probes, as proposed by Vinnemeier and Parvizinia [19, 20], was used.

For the penny configurations with nominal and double annular gap, one brass plug, each with 36 wall pressure taps, was manufactured and inserted into the inner hub ring. Fig. 3 shows four views of the brass plug of the double penny, indicating the position and arrangement of the pressure taps. On the bottom of the plug, a total of 20 pressure taps were distributed over two radii to allow the measurement of the penny bottom gap. In the penny annular gap, the 16 taps were arranged equidistantly every 22.5° at a constant distance of 2.3% of the vane height below the hub contour.

Two vanes of each penny configuration were provided with pressure taps at 10% span. Due to the thin wall thickness of the compressor vane, the instrumentation of the suction and pressure sides had to be split on two vanes. The rotation of the hub was used for the measurement, so that the pressure taps both from the pressure side and the suction side were evaluated at the same circumferential position. The suction side of one vane was provided with 10 pressure taps, which were clustered in the area of the expected suction peak at approx. 10% of the chord length. The pressure side of the neighboring vane was provided with 5 pressure taps, which were arranged equidistantly. The pressure-side and suction-side instrumented vanes of the no-penny configuration can be seen in Fig. 4. The pneumatic access to the pressure taps is made by pressure tubes, which are soldered into milled-in channels.

### Table 4 Annular cascade wind tunnel specifications at the inlet plane at midspan, *based on axial chord length*

| Parameter                  | Value |
|----------------------------|-------|
| Total Temperature [°C]     | 50    |
| Total Pressure [Pa]        | 140000|
| Reynolds Number [-]*       | 382000|
| M [-]                      | 0.34  |
| yaw angle [°]              | 42    |

The measurement technology used to detect penny leakage consisted of pressure taps on the pressure and suction side of the vane, as well as pressure taps in the penny annular and bottom gaps. The adjustment of the probes in a circumferential direction was carried out with the circumferential traverse devices in the inlet and outlet planes, which covered a range of 2.5 pitches. For radial traversing and yaw angle, the probes were mounted in a probe traverse device, which was placed on the circumferential traverse devices in both planes.

Main data for the measuring grids in the inlet and outlet plane can be found in Table 3. The measuring grid in the inlet plane comprised 544 points, which were circumferentially uniformly distributed and can be seen in Fig. 5. The core area of the measuring grid was traversed by a 5-hole-probe. The 5-hole-probe used had a head diameter of 2.8mm. The 5-hole-probe was not placed any closer than 8% span to the hub and the casing wall in order to avoid measurement error. The three radii near the hub and casing were measured with a 3-hole-probe. Since its head height was only 0.7mm, the 3-hole-probe could be moved up to 2% span to the wall. The measuring grids of the 5-hole-probe and the 3-hole-probe overlapped each other. Since the triple-hot-wire-probe had similar dimensions as the 5-hole-probe, both used the same measuring grid. All probes were preset in the inlet plane according to the mean yaw angle of 40°. The measuring grid in the outlet plane, consisting of 1856 points, was adapted to capture the wakes from the penny leakage and refined circumferentially and radially. The finest spatial resolution was 2.9% of the pitch in circumferential direction and 2.3% span in radial direction.
RANS simulations with 155% and 310% of the annular gap were performed, in order to better understand the influence of the penny gap size on penny leakage.

\[
\text{Exp: inlet plane, no penny (P0)}
\]

**RESULTS AND DISCUSSION**

Selected results of the numerical and experimental investigations on penny leakage on the annular cascade wind tunnel (according to Fig. 2) for five different annular gap sizes (according to Table 2) are presented below. The entry conditions, as shown in Table 4, were kept constant for all tests presented. The staggering angle of the VSV was in the nominal position, so that the penny was aligned with the hub contour and the inflow was incident-free.

First, the results which were obtained at the outlet plane downstream of the stator with hub-side penny cavities are explained. The results on the vane profile and inside the penny cavity are then described in detail. Finally, the results of surface oil flow visualization are presented.

**Stator Exit Flow Field**

The results downstream of the stator row are evaluated in the outlet plane, 13% of the axial chord length downstream of the trailing edge, as shown in Fig. 2 and Table 3. The experimental data were obtained with multi-hole and hot-wire-probes. This section is divided into total pressure loss, streamwise vorticity, and turbulence intensity. All numerical and experimental results of the stator outlet flow exceed 2.5 pitches and point in flow direction.

**Total pressure loss.** The normalized total pressure at the outlet plane downstream of the no-penny configuration for the experiment and the RANS simulation is shown in Fig. 6. In the plot of the experimental data, the outer 2% span is missing to keep a minimum distance to the wall with the 3-hole-probe. For comparison, the hub and casing contours are shown. The total pressure

\[
P_{t,norm} = \frac{P_t}{P_{t,in,m}}
\]

is normalized to the inlet condition with the mass-averaged total pressure \(P_{t,in,m}\) in the inlet plane at midspan. The results of the experiment and the RANS simulation are in good agreement for the no-penny configuration. Increased total pressure losses are visible in the wakes and in the areas of the half-gap and passage vortices on the hub and the casing. The total pressure loss due to the half-gap vortex on the hub is higher than on the casing, because of the larger half-gap due to the convex hub contour. The "L-shaped" pattern in the normalized total pressure, shown in the inlet plane in Fig. 5, is also apparent in the free flow in the outlet plane. This confirms that the displacement of the two measuring planes by 23° (Table 3) was reasonably selected and that the same streamlines are considered in the inlet and outlet plane. The normalized total pressure from the RANS simulation also shows the "L-shaped" pattern, since the 2D boundary conditions (Fig. 5) were taken from the experiment. The
transition model ensures that the width of the wakes is reproduced extremely well by the RANS simulations. Compared to the experiment, the half-gap vortex on the casing side is more concentrated and stronger in the RANS simulation.

**Fig. 6** Normalized total pressure by Eq. (1) at the outlet plane over 2.5 pitches downstream of the no-penny configuration: 5-hole-probe and 3-hole-probe (Top), RANS (Bottom)

In Fig. 7 differences in the normalized total pressure between the double penny and no-penny configuration at the outlet plane for the experiment and the RANS simulation are shown. The difference in the normalized total pressure is calculated as in Eq. (1) with the mass-averaged total pressure in the inlet plane at midspan. The additional total pressure loss generated by the penny leakage is in the area of the loss region of the hub-side half-gap vortex. The losses caused by the penny leakage and the hub-side half-gap vortex are superimposed. The additional penny loss results from the formation of a penny vortex and the resulting mixing losses, as described by Stummann [14]. The location of maximum penny loss is slightly above the half-gap vortex and more distant from the suction side of the vane. The additional penny loss in the RANS simulation and the experiment is about the same order of magnitude, but the extent of the loss region in the RANS simulation is slightly larger.

The pressure loss due to the half-gap vortex on the casing side disappears in the plot showing the differences. In contrast to the RANS solution, the wakes do not disappear in the plot showing the differences in the experiment, partly due to the limited accuracy of 0.8 pitches in the circumferential positioning of the stator.

Also larger differences in total pressure loss between the two passages are visible. The additional total pressure loss of the penny in the right passage of the experiment is significantly smaller than in the left passage. This difference is also present in the RANS solution, but is smaller. The difference in total pressure loss between the passages results most likely from the asymmetrical inflow, as shown in Fig. 5.

**Fig. 7** Difference in normalized total pressure by Eq. (2) between double penny and no-penny configuration at the outlet plane: 5-hole-probe and 3-hole-probe (Top), RANS (Bottom)

The circumferentially mass-averaged differences in total pressure loss according to Eq. (2) over 2 passages are shown for the nominal and the double penny configuration in Fig. 8. The maximum total pressure loss is at 15% span for all penny configurations in both the experiment and the RANS simulations. Roughly speaking, doubling the annular gap doubles the maximum total pressure loss. The experimental and numerical results at 13% of the axial chord length behind the vane trailing edge show that the additional penny loss affects the region close to the hub by up to approx. 40% span. In the RANS simulations the influence of penny leakage is estimated as being slightly stronger in the maximum value.

**Fig. 8** Circumferentially mass-averaged differences in normalized total pressure by Eq. (2) over 2 passages at the outlet plane for nominal and double penny in the experiment and in RANS
The total pressure loss coefficient is determined for the RANS simulations and the experimental data using

$$\omega = \frac{p_{\text{in}} - p_{\text{out}}}{p_{\text{ref}} - p_{\text{in}}}.$$  \hspace{1cm} (3a)

The static pressure $p_{\text{in}}$ at the inlet plane is area-averaged, the total pressure $p_{\text{in}}$ at the inlet plane and the total pressure $p_{\text{out}}$ at the outlet plane are mass-averaged in the radial and circumferential directions over 2 passages. The development of the relative difference in total pressure loss coefficient

$$\frac{\Delta \omega_{\text{penny}}}{\omega_{\text{penny}}} = \frac{\omega - \omega_{\text{ref}}}{\omega_{\text{penny}}}$$  \hspace{1cm} (3b)

compared to no-penny cavity over the size of the annular gap for the experiment and the RANS simulations is shown in Fig. 9 and Table 5. The relative difference in total pressure loss coefficient, as calculated with RANS, increases in a roughly linear fashion with the annular gap size. The measurement uncertainties in the experimentally determined relative differences in the total pressure loss, shown in Fig. 9, display a linear behavior, but with a smaller slope than in the RANS data. The total pressure losses caused by the penny leakage are predicted by the RANS simulations much more strongly than the experiment shows. When calculating with RANS, the additional penny losses for the double annular gap are 7.3% more than in the experiment. As shown in Fig. 7 and 8, the RANS simulations overestimate the size and the maximum value of the penny loss area.

![Fig. 9 Development of the relative difference in total pressure loss coefficient by Eq. (3b) over the size of the annular gap calculated for the experiment and the RANS simulations](image)

**Table 5 Relative difference in total pressure loss coefficient**

|                | penny  | nominal (P1) | nominal enlarged* | double (P2) | double enlarged* |
|----------------|--------|--------------|-------------------|-------------|-----------------|
| Exp            | 1.9%   | -            | 6.8%              | -           | -               |
| RANS           | 6.5%   | 11.2%        | 14.1%             | 20.2%       |

The most intensive vortices measured in the no-penny configuration are the half-gap vortices (HV) on the hub and casing. The pressure difference across the vane causes a leakage flow through the half-gap from the pressure side to the suction side. As the leakage flow joins the main flow, it is dragged by the latter and folds into a vortex. On the casing, the half-gap vortex rotates counter-clockwise on the hub clockwise in flow direction. The differences between the RANS simulation and the experiment are small. The half-gap vortex on the casing is estimated to be too small by the RANS simulation.

The passage vortex (PV) on the hub and the casing is caused by the crossflow from the pressure side to the suction side at the boundary layers of the endwalls. As the crossflow meets the suction side of the adjacent vane, the crossflow deflects towards the center of the channel. Due to the conservation of mass, fluid flows into the boundary layer and the passage vortex is formed. The passage vortex rotates counter-clockwise on the hub and clockwise on the casing in flow direction. The passage vortex is influenced by the half-gap vortex and carried outwards.

![Fig. 10 Non-dimensional streamwise vorticity by Eq. (4) at the outlet plane over 2.5 pitches downstream of the no-penny configuration: 5-hole-probe and 3-hole-probe (Top), RANS (Bottom)](image)
The influence of penny leakage on the prevailing vortices present in the stator row is visualized with the non-dimensional difference in streamwise vorticity

\[
\Delta \omega_{sw} = \frac{\omega_{sw} - \omega_{sw P0}}{\omega_{ref}}
\]

compared to no-penny cavity. Figure 11 shows the difference between double penny and no-penny cavity in non-dimensional streamwise vorticity at the outlet plane for the experiment and the RANS simulation. Stummann's numerical studies [14, 16] show that the suction-side outflow of the penny cavity forms an asymmetric, counter-rotating vortex pair: A suction-sided penny-vortex (SSPV) rotating clockwise and a pressure-sided penny-vortex (PSPV) rotating counter-clockwise in flow direction. The low-energy SSPV already disappears in the blade passage due to the pressure gradient and can therefore not be observed in the outlet plane. In contrast, the counter-clockwise rotating PSPV absorbs the crossflow and strengthens the PV, which has the same direction of rotation. The experiment confirms the strengthening of the PV by the PSPV in the outlet plane and this is marked in Fig. 11 as PSPV+PV. In addition, both the experiment and the RANS solution show two more changes in streamwise vorticity caused by the influence of the double penny affecting the HV. Due to the influence of the penny cavity, the HV is weakened in its vorticity and additionally shifted towards the channel center by the influence of the counter-clockwise rotating PSPV. The weakening and the displacement of the HV can be seen in Fig. 11, both in the experiment and in the RANS simulation.

**Turbulence Intensity.** Traverses with triple-hot-wire-probes were performed to measure unsteady flow parameters, such as the turbulence intensity

\[
T_u = \sqrt{T_c \left( \frac{c_{m}^2 + c_{u}^2 + c_{r}^2}{c} \right)}
\]

The difference in turbulence intensity

\[
\Delta T_u_{Penny} = T_u - T_u^{P0}
\]

between double penny and no-penny cavity at the outlet plane for the experiment and RANS is shown in Fig. 12. The triple-hot-wire-probe was not moved by more than 8% span to the endwalls to keep a minimum distance to the wall. Both the RANS simulation and the experiment are quite similar in terms of quality. Each passage has two centers with increased turbulence intensity. The large increase in turbulence intensity on the left side results from the strengthening of the passage vortex by the pressure-sided penny-vortex. The smaller center with increased turbulence intensity on the right side shows the displacement of the half-gap vortex, which is caused by the influence of the penny vortex. The region of the half-gap vortex weakened in vorticity can be recognized in Fig. 12 by the local minimum of \( \Delta T_u \). It is striking that the turbulence intensity even increases around the local minimum of \( \Delta T_u \), despite a significant decrease in vorticity (compare Fig. 11 and 12). Overall, only an increase of turbulence intensity can be observed due to the influence of the penny cavity.

**Exp:** outlet plane, double - no penny (P2-P0)

**RANS:** outlet plane, double - no penny (P2-P0)

Fig. 11 Difference in non-dimensional streamwise vorticity by Eq. (5) between double penny and no-penny cavity at the outlet plane: 5-hole-probe and 3-hole-probe (Top), RANS (Bottom)

**Exp:** outlet plane, double - no penny (P2-P0)

**RANS:** outlet plane, double - no penny (P2-P0)

Fig. 12 Difference in turbulence intensity by Eq. (7) between double penny and no-penny cavity at the outlet plane: triple-hot-wire-probe (Top), RANS (Bottom)
Figure 13 shows the circumferentially area-averaged differences in turbulence intensity relative to no-penny cavity for nominal and double penny in the experiment and in the RANS simulations. The data were circumferentially averaged over 2 passages. The results show that the influence of the penny leakage on turbulence intensity reaches up to approx. 35% span for any annular gap size, which is similar to its influence on the total pressure (see Fig. 8). The difference between the RANS simulation and the experiment is quite large, with 1% for the nominal penny and 2% for the double penny.

Profile Pressure Distribution

The influence of penny leakage on the profile flow is shown at 10% span for the experiment and RANS in Fig. 14. The profile pressure is scaled with

$$c_p = \frac{\bar{p}_{\text{in,10\%}} - \bar{p}_{\text{in,10\%}}}{\bar{p}_{\text{t,10\%}}}$$

using the circumferential area-averaged static pressure $\bar{p}_{\text{in,10\%}}$ and the mass-averaged total pressure $\bar{p}_{\text{t,10\%}}$ at the inlet plane at 10% span. The non-dimensional coordinate $x_{\text{rel}}$ is normalized with the axial chord length, starting at the leading edge. The vanes with pressure taps were rotated in circumferential direction to the central position, so that the results compare well with the measurements of the stator exit flow field (see wake of center vane in Fig. 6).

The pressure distribution around the airfoil controls the pressure distribution on the hub and the pressure distribution on the hub controls the penny cavity leakage flow, as has been shown by Stummann [14]. With increasing penny annular gap size, the pressure at the suction peak at about 10% - 15% of the chord length increases. The penny leakage causes a blockage which slows down the flow on the suction side of the vane. After a separation bubble downstream of the suction tip, the flow reattaches at approx. 30% of the chord length, which can be seen for all penny configurations in the pressure distribution of the RANS solution. The profile pressure decreases on the rear part of the suction side beginning at about 40% of the chord length with increasing penny annular gap size. The numerical results are in good agreement with the experiment for the suction side. However, the influence of the penny leakage on the pressure side differs for the experiment and the RANS simulations, but seems to be smaller.

Penny Cavity Flow

The results of the pressure taps in the bottom gap and in the annular gap (according to Fig. 3) are compared for the double penny with the RANS results. The static pressure in Fig. 15 and 16 is scaled using Eq. (8). The non-dimensional coordinate $x_{\text{rel}}$ and $y_{\text{rel}}$ are normalized with the axial chord length, starting at the leading edge. A generic vane profile is shown for reasons of comparison. The vane with the instrumented brass plug was rotated in circumferential direction to the center vane position, as was also done with the blades for determining the profile pressure distribution.

Annular gap. The pressure distribution in the annular gap of the double penny at 2.3% of the vane height below the hub contour is shown in Fig. 15 for the experiment and for the RANS simulation. It is known from previous publications [13-16] that the inflow regions into the penny cavity are located at the leading edge (1) and in the rear area of the pressure side (2). The two outflow regions can be found in low pressure regions on the hub. The outflow region with the stronger mass flow and impulse ratio is located on the suction side (3) and the weaker outflow region is on the pressure side (4). For the pressure distribution inside the annular gap, distinct pressure minima can be seen in the area of the two
inflow regions, as well as in the area of the outflow region on the suction side. The RANS simulation estimates the pressure minima in the region of the outflow on the suction side (3) more severely. The weak outflow region on the pressure side (4) cannot be identified in the pressure distribution in the annular gap.

**Fig. 15** Pressure distribution by Eq. (8) inside the annular gap at 2.3% of the vane height below the hub contour for the double penny; generic vane profile: pressure taps as shown in Fig. 3 (Top), RANS (Bottom)

**Bottom gap.** The pressure distribution inside the bottom gap for the double penny for the experiment and for the RANS simulation is shown in Fig. 16. In addition to the generic vane profile, the inner and outer diameters of the bottom gap are marked. In the RANS plot, additional streamlines are drawn. The step size of the contour levels (0.005) corresponds approximately to the measuring accuracy of the pressure sensors.

As known from previous investigations [13-16], the leakage entering the penny cavity from the two inflow regions flows mainly to the outflow region on the suction side. Only part of the incoming mass flow reaches the outflow region on the pressure side. The leakage flow entering from the pressure side meets the vane spindle and forms a stagnation point there (I). The stagnation point is clearly visible in the experiment and the position corresponds to the calculated position in the RANS simulation.

A large part of the incoming leakage flow from the pressure side does not reach the bottom gap, but flows at high speed through the annular gap to the suction side outflow. Due to the high velocities, a pressure minimum (II) is generated, which is confirmed by the experiment. Here too, the position corresponds well with the RANS solution.

**Fig. 16** Pressure distribution by Eq. (8) inside the bottom gap for the double penny; generic vane profile: pressure taps as shown in Fig. 3 (Top), RANS (Bottom)

**Surface oil flow visualization**

Surface oil flow measurements on the hub, on the vane profile, and in the penny cavity were performed. Four different colors were used: for the hub (white); the penny cavity (red); the vane suction side (blue); and the vane pressure side (green). The results of the surface oil flow measurements and the wall shear stress calculated with RANS simulations are shown in Fig. 17 and 18 for the double penny.

The surface oil flow measurements in Fig. 17 show very clearly the two outflow regions on the suction side (3) and the pressure side (4) of the penny cavity. Both the outflow on the suction side and the outflow on the pressure side move to the suction side due to the passage crossflow of the endwall boundary layer. Of particular note is the two-part outflow region (3) on the suction side, as predicted by the numerical studies of Stummann [14, 16]. The front outflow shows the trace of the suction-sided penny-vortex, which already dissipates in the vane passage and is dragged into the hub suction.
side corner separation. The trace of the rear outflow shows the pressure-sided penny-vortex, which strengthens the passage vortex and interacts with the half-gap vortex. In the outflow, the trajectory of the combined penny vortex and hub-side half-gap vortex is visible (A), as measured in the stator exit flow field (see Fig. 11, 12).

In conformity with Fig. 15 additional oil flow measurements within the penny cavity, not shown here, confirm the inflow regions by the white colored oil that flows into the penny cavity. The oil flow measurements also provide results in the bottom gap and confirm the streamlines according to Fig. 16.

CONCLUSION
Experimental data of the penny cavity leakage flow were compared to RANS data for different annular gap sizes. The main results are:

1. The penny leakage flow causes increased relative losses of 1.9% in total pressure loss coefficient for the nominal annular gap and 6.8% for the double annular gap according to 3-hole and 5-hole-probe measurements.
2. The total pressure losses caused by penny leakage are strongly overestimated in the RANS simulations, compared to the experiment with 6.5% for the nominal annular gap and 14.1% for the double annular gap.
3. The additional penny losses detected at 13% of the axial vane chord length downstream of the trailing edge are limited to the lower 40% span in the experiment and in the RANS simulations for all investigated annular gap sizes.
4. The interaction of the pressure-sided penny-vortex, resulting from the penny leakage on the suction side as predicted by Stummann [14, 16], with the passage vortex, was shown in the streamwise vorticity calculated from multi-hole-probe data. Furthermore, a displacement and a weakening of the half-gap vortex due to the influence of penny leakage were observed.
5. Two centers with increased turbulence intensity were measured and calculated with RANS showing the influence of penny leakage. The largest increase in turbulence intensity can be observed in the area of the passage vortex, strengthened by the pressure-sided penny-vortex. The smaller increase shows the displacement of the half-gap vortex. The influence of penny leakage on turbulence intensity reaches up to approx. 35% span for all investigated annular gap sizes. However, the RANS simulations overestimate the maximum turbulence intensity by a factor of 2.
6. The penny leakage flow changes the pressure distribution of the vane, which was shown at 10% span for the experiment and the RANS simulation. The penny leakage flow reduces the suction peak by increasing annular gap size.
7. The inflow and outflow regions of the penny cavity predicted by
CFD and shown by Wolf and Stummann [13-16] were verified with the results of static pressure taps inside the penny annular gap and surface oil flow measurements.

8. The flow inside the bottom gap of the Penny cavity generates a distinctive stagnation point and pressure minimum, which is confirmed by the data of the pressure taps inside the penny cavity.

Future work at the Institute of Jet Propulsion and Turbomachinry will contain measurements of restaggered VSVs, as well as additional penny cavity geometries. In addition, Particle Image Velocimetry measurements inside the penny cavity bottom gap, as well as in the stator passage, are being performed in cooperation with the University of the German Federal Armed Forces in Munich. For this purpose, plugs of the penny cavities made from glass were manufactured. First results are very promising and differences between the penny cavities are being measured in the cavity, as well as in the outflow in the passage. A better understanding and more effective validation of the simulations will be gained as a result of this.

ACKNOWLEDGEMENTS

The results presented in this paper have been derived from the project “AG Turbo: Stabiler Hochdruckverdichter” (COOREFLEX 1.2.7b). This work was funded by the German Federal Ministry of Economics and Technology (BMWi). The authors would like to thank Markus Schafferus for his work on the surface oil flow visualizations.

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