# INVESTIGATION OF THE INFLUENCE OF DIFFERENT RIM SEAL GEOMETRIES IN A LOW-PRESSURE TURBINE

### P.Schuler, K.Dullenkopf and H.-J.Bauer

Institut fuer Thermische Stroemungsmaschinen Karlsruher Institut fuer Technologie (KIT) 76131 Karlsruhe, Germany

### ABSTRACT

The sealing of the machine's inside against hot-gas ingestion is commonly provided by blowing relative cold compressor air radially out through the turbine wheelspace. Rim-seals located inside the wheelspace are primarily designed to keep the required amount of sealing at a minimum. A further possible function of the rim-seal follows from the desire to reduce the aerodynamic losses contributed by the interaction of the emerging sealing flow with the boundary layer of the incoming main flow. Investigations performend in the EU project MAGPI concentrate on the interaction between the sealing flow and the main gas flow and in particular on the effect of different rim seal designs regarding the loss-mechanism in a low-pressure turbine passage. Two different rim seal designs inside a linear lowpressure turbine cascade rig have been analysed in detail. Both, the simple axial gap and the more complex compound design were investigated under the influence of different sealing mass flow rates. Furthermore, a configuration without any cavity in the main gas flow served as a reference case. Extensive measurements of the total pressure loss over the turbine blade have been conducted by means of a five-hole probe. Additionally, the blade loading has been measured at several blade heights. A considerable increase of total pressure losses was observed due to the presence of a cavity with any rim seal design, even for no sealing flow. Higher sealing mass flow rates intensified this effect which becomes manifested in a strengthening of the secondary flows downstream the cascade. Experiments revealed also significant differences in loss-increment depending on the rim seal design used. Deeper insight into the interaction of the flows close to the

rim seal is given by results of Laser-Doppler-Velocimetry measurements. The rounded shape of the compound design, which implies an axial overlapping, represents a promising prevention against hot-gas ingestion. While the axial gap design is characterized by higher losses, it also suffers considerable hot-gas ingestion in front of the blade leading edge. A parametric study regarding a possible optimization of the axial gap design is presented in this work.

# NOMENCLATURE

Latin letters:		
$C_x$	m	axial chord-length
Ма		Mach number
Re		Reynolds number
Т	Κ	temperature
Ти		turbulence intensity
U	m/s	flow velocity
h	m	blade height
'n	kg/s	mass flow rate
Р	Pa	pressure
р	Pa	pressure
S	m	blade pitch
S		stream-wise direction
v	m/s	y - velocity component

w

m/s	z - velocity component
-----	------------------------

#### Greek letters:

offer fetters.		
α	deg	blade design angle
δ		velocity boundary thickness
δ*	mm	displacement thickness
γ	deg	sealing flow angle, $tan^{-1}(U_z/U_s)$
Π		pressure ratio, <i>Ptot</i> , <i>in</i> / <i>pstat</i> , <i>out</i>
Θ	deg	local flow angle
θ	mm	momentum thickness
CĘ		vorticity coefficient
ζ		total pressure loss coeffcient
Abbriviation:		
ITP		Industria de Turbo Propulsores
LE		blade leading edge
TE		blade trailing edge
Superscripts:		
,		Mass-averaged flow quantity
-		Area-averaged flow quantity
Subscripts:		
m		main flow
S		sealing flow
ref		reference
sta		static
tot		total
р		probe
x		x-direction component
у		y-direction component
Z		z-direction component
1		upstream blade LE
2		downstream blade TE
$\infty$		freestream

## INTRODUCTION

The desire of a continuous thermal efficiency improvement for turbo-machine has pushed the turbine entry temperature close to 2000 K. A further increment of the temperature is primarily limited by the maximum material temperature of the components. Also the ingestion of hot gases into the wheel space between stator and rotor-disc has to be prevented as the components inside the machine are only able to handle lower temperatures. This circumstance has become even more critical by the relatively flat radial temperature profiles produced by modern low emission combustors [1]. Even in the low-pressure part of the turbine, temperatures are considerably high and could cause a reduction of turbine life if ingestion is not prevented. In order to guarantee an adequate thermal environment inside the wheelspace, cool air is bleed off the compressor and ejected in radial direction through the wheel-space. This provides both, cooling for the turbine disk and sealing against the pressure difference between wheel-space and annulus flow. However, cooling penalties have to be taken into account as the cooling air "bypasses" the cycle and therefore can easily compensate the benefits of the increased turbine entry temperature. In modern low-pressure turbines the cooling flow rate through the wheel space is around 0.5% of the annulus flow [2]. According to Bohn et al. [3] the total amount of cooling flow used to both actively cool the disk and for sealing purposes causes about 6% of the engine specific fuel consumption.

In order to reduce the required amount of sealing flow, rimseals are applied at the wheel space. Bohn et al. [4] estimated, that up to 30% of the sealing flow could be saved if a appropiate rim-seal geometry is used. Numerous experimental works have been conducted the past 40 years on linear cascade rigs as well as on rotating rigs in order to investigate the phenomenon of hot-gas ingestion. The knowledge about its driving factors enabled the researchers to determine an adequate rim-seal design to minimize the required sealing flow rate. Early experiments conducted by Phadke and Owen [5] and [6] as well as later by Bohn et al. [4] and Gentilhomme et al. [7] revealed the inhomogeneous circumferential pressure distribution in the annulus flow to be one of the driving factors. Further investigations on a 1.5stage turbine published by Bohn et al. [3] and [8] showed that the asymmetric static pressure distribution is dominated by the local pressure rise in front of the rotor blade leading edge and its interaction with the wake generated by the upstream stator vanes.

Early experimental works, such as these from Phadke and Owen or Bhavani et al. [9] detected hot-gas ingestion when rotational speed was increased. This was confirmed by the results presented some years later by Bohn et al. [4] and Geis et al.[10]. The latter determined the minimum cooling flow rate necessary to prevent ingestion by means of laser light scattering within a rotor-stator system. The signal intensity of the scattered light detected by a photomultiplier indicated the strength of the ingestion. It was found out that the requested sealing flow rate rose as the disk speed increased. The author stated that the faster rotating disc reduced the static pressure inside the wheel space and thus providing higher probability for hot-gas ingestion.

Different designs of rim-seals were applied in the course of the mentioned investigations, and it is agreed by many authors [3, 5, 8, 9, 11, 12], that designing the rim-seal with axial overlapping represents an effective way to prevent hot-gas ingestion. It is argued, that the enclosed volume in between the outer and inner seal serves as a damper against pressure fluctuations inside the annulus and encourages at the same time the development of an inner recirculation zone as an additional barrier against hot-gas ingestion.

As the primary function of the rim-seal is to reduce the required amount of sealing flow, the aerodynamic spoiling caused by the sealing flow, when it is ejected through the rim-seal, was

2



Figure 1: VORTEX SYSTEM IN A TURBINE CASCADE by Wang (1997)

not subject of the mentioned investigations. However, the desire to further improve the aerodynamic efficiency of the turbine and the latest advances in analytical and computational methods have moved the focus of current research on this area. The capability of improving the aerodynamic efficiency is seen in the interaction between the ejected sealing flow and the incoming boundary layer of the main flow. The resulting turbulences in the near wall flow affect the formation of secondary flows, which are the main contributor to total losses inside a turbine passage [13]. The aerodynamic losses associated with secondary flows have been reported in detail recently by Simon and Piggush [14]. Sharma and Butler [15] as well as Burd and Simon [16] estimated these losses to be responsible for 30-50% of the total pressure loss inside a turbine passage. A comprehensive review of the secondary flow pattern inside a turbine cascades has been presented in [17– 19]. One representative model of the flow pattern is described by Wang [19] and shown in Fig.1. The general representation of a linear cascade, depicting the airfoils and end walls in a turbomachine, shows the separation of the incoming boundary layer in front of the leading edge. The radial total pressure gradient of the near wall flow leads to a roll up of the boundary layer in front of the blade leading edge forming the two legs of the horseshoe vortex [20]. The pressure side leg of the horseshoe vortex  $V_{ph}$  is fed by the fluid located in the boundary layer and converts into the passage vortex  $V_p$ . The position of the considerably weaker suction side vortex  $V_{sh}$  depends on the rotational speed of the passage vortex, which in turn depends on the blade geometry and the overall flow conditions. Apart from the very small corner vortices, labeled with the subscript c, a small wall vortex  $V_{wip}$  of high-energy originates near the merging point of the two legs of the horseshoe vortex.

Burd and Simon pointed out, that coolant ejection through a slot might reduce secondary flows in strength and size allowing thermal protection as well as wheel space sealing without any aerodynamic penalties. In fact, they demonstrated with experiments inside a stator cascade including contoured endwall the possibility to suppress the horseshoe vortex. The sealing flow was ejected through a slot below 45 deg in axial direction. They argued, that the high sealing flow rate re-energized the low-momentum endwall flows and suppressed the endwall cross flows. This observation was also reported by Kost and Nicklas [21], who conducted detailed aerodynamic measurements in a high pressure linear turbine cascade with sealing flow ejection through a slot. However, the majority of publications dealing with this phenomenon report of a detrimental effect on turbine efficiency caused by the sealing flow. De la Rosa Blanco et al. [2] performed experiments in a linear cascade, where the sealing mass flow rate and its tangential velocity were varied independently. Even without any sealing flow and only by the presence of the upstream slot, they noticed a considerable change in the secondary flows as well as an increment of the total pressure loss. This was attributed to the possibility of flow exchange between the main flow and the fluid inside the cavity. An additional ejection of sealing flow led to a strengthening of the secondary flows and a further increment of losses.

Popovic and Hodson [22] used the same test-facility as de la Rosa Blanco and investigated the aerothermal performance of an overlapping rim-seal in front of a highly-loaded turbine blade. They also recognized, that higher cooling levels by means of an increased sealing flow rate led also to higher aerodynamic losses. Gallier et al. [1] conducted experiments on a 2-stage axial turbine with a simple axial slot ejection. The results showed that an increase of the sealing flow rate could prevent hot-gas ingestion but also provoked a stronger disturbances of the boundary layer. In contrast to the case described by Burd and Simon, here the sealing flow led to an increased amount of low-momentum endwall flow resulting in a larger horseshoe vortex. Experimental works on a 1.5-stage axial turbine with different endwall contours presented by Schuepbach et al. [23] showed also a decrease of efficiency with increasing sealing flow rates. Additional timeresolved simulations revealed the formation of sealing flow jets in front of the blade row causing a blockage effect.

Hunter and Manwaring [24] conducted experimental investigations and numerical analysis on a multi-stage low-pressure turbine. They stated that the primary source of loss associated with the sealing flow was the mixing of circumferential momentum components. Losses increased with an increasing mismatch of circumferential momentum between sealing flow and the main gas flow which was turned by the upstream stator vanes. Similar conclusion can be found in the works presented in [2, 25–28] who consider the tangential velocity of the sealing flow to be a key parameter that determines the endwall flows structure within the blade passage. Demargne and Longley [25] carried out investigations on a compressor cascade. The results showed that an increased tangential velocity can compensate negative effects of any sealing flow rate. At tangential velocities above main gas flow values, the cooling flow counteracts the cross-passage pressure gradient and thus weakens the end wall secondary flows. Girgis at al. [26] reported that experiments on a 1-stage turbine rig showed an improvement of turbine efficiency of 0.3% per 1% sealing flow rate given a tangential component of 15 deg. in direction of blade rotation. A favourable side effect of ejecting the sealing flow with a tangential velocity component had been the reduced radial penetration of sealing flow into the main gas flow. Ong at al. [27] found out, that the sealing flow remains for a longer distance close to the endwall and that its penetration depth is reduced when increasing the tangential component. Reid et al. [28] conducted efficiency measurements and area traverses by means of a five-hole probe in a low speed axial turbine with axial overlapping rim-seal. The initial drop of turbine efficiency caused by the sealing flow had been compensated partialy by giving the sealing flow a tangential component in direction of the blade rotation.

The investigations mentioned above demonstrate the significant effect of sealing flow on both, the prevention of hot-gas ingestion as well as on the aerodynamic loss-generation inside a turbo-machine. Thus, with regard to a further improvement of turbo-machine efficiency, the aerodynamic spoiling caused by the sealing flow has to be taken into account when preventing hot-gas ingestion. A possible way to confine the detrimental effect of the sealing flow is the use of an adequate rim-seal design. Recent numerical studies on different rim-seal designs upstream a low-pressure turbine passage presented by Schuler et al. [29] confirmed this assumption and predicted considerable changes in total pressure losses depending on the rim-seal design used. The aim of this paper is to investigate experimentally the influence of different rim-seal geometries on the aerodynamic loss behavior inside a low-pressure turbine cascade. A simple axial gap and a more complex compound rim-seal design are compared for three different sealing flow rates. Results of extensive total pressure loss measurements downstream the cascade are presented, while laser-doppler-measurements close to the rim-seal give a further insight into the interaction of the flows. Additionally, results of a parameter variation based on the axial gap design are presented.

# EXPERIMENTAL METHOD The Cascade Rig

The investigations were performed in a linear cascade of five blades. The blade design was provided by ITP in the course of the MAGPI project and represents a typical low-pressure turbine. A schematic of the test-section is shown in Fig.2.

The rig is run in an open circuit which includes a centrifugal blower and a venturi pipe to measure the main mass flow rate. Inside the settling chamber honeycombs homogenize the main flow



Figure 2: CASCADE RIG

and reduce the swirl. Before the main flow enters the test-section, the turbulence intensity is increased to engine-representative levels by means of a turbulence grid. In order to compensate the mismatch between the flow velocities at the inner and outer radius of the cascade, two adjustable tailboards are used along the channel walls. These were moved to a position where the static pressure distribution along the surface of the three inner blades showed good agreement and thus where optimal periodic conditions for the cascade could be assumed. The maximum variation of the static pressure was measured along the suction side and revealed to be less than 1% of the incoming dynamic head. The

Table 1: RIG DETAILS AND OPI	ERATING CONDITIONS
------------------------------	--------------------

rig details				
$C_x$	75 mm			
$h/C_x$	1.3			
$s/C_x$	0.884			
$\alpha_1$	37.9 deg			
$\alpha_2$	66.3 deg			
operating conditions				
П	1.035			
Re	$5.6 \cdot 10^5$			
$Ma_2$	0.22			
$\dot{m}_m$	1.13 kg/s			
$Tu_1$	5.88%			
Incidence	0 deg			
$\dot{m}_s/\dot{m}_m$	0.0-1.0%			
γ	45 deg			

rim-seal is integrated into a cavity-module which is inserted into the test-section upstream the blade leading edge. The modular design allows for a reassembly without higher effort as well as for an easy exchange of the different rim-seal designs. The cavity covers nearly all passages formed by the five blades. The sealing air is bled off the main flow inside the settling chamber. It is guided through a secondary pipe-system where the sealing flow rate is adjusted via a control valve and measured by means of a hot-film sensor which is integrated into a measuring tube. The accuracy of the hot-film sensor is 3% of the measured sealing flow rate. A water cooled radiator provides for a temperature reduction of the sealing flow of around  $\Delta T = 20$ K before it is reintroduced through the rim-seal into the main flow. In order to simulate the rotor/stator movement the sealing flow is ejected through the rim-seal with an tangential component of  $\gamma = 45 \text{ deg}$ with respect to the span direction. This is achieved by an inclined flow channel of the cavity module including honeycombs with a length-diameter ratio of 4. The operating point of the cascade rig corresponds to the blade design's pressure ratio of  $\Pi = 1.035$ . It is defined by the inlet total pressure and the static pressure measured downstream the cascade. During experiments the inlet total pressure, which is measured with a pitot-probe  $1C_x$  upstream the blade leading edge, is controled to a constant value with an accuracy of  $\pm 10$  Pa. This is achieved by a control valve which automatically adjusts the opening of a bypass closely downstream of the blower exit. As a consequence of the open circuit, the corresponding Reynolds number, which is based on the blade suction side length, undergoes slight variations depending on the ambient temperature. Table 1 summerizes the essential design parameters of the test-rig and the operating conditions of the experiments.

#### The rim-seal designs

The results presented in this work concentrate on the experiments conducted with two different rim-seal designs, the axial gap and the compound design, respectively. Fig.3 depict the two geometries, which also have been under consideration in the numerical simulations reported in [29]. While the axial gap represents a simple gap upstream the blade leading edge, the more complex compound seal comprises an axial overlapping and forms a volume between the primary and secondary seal. Both designs agree in its dimension regarding the axial opening as well as in the distance between the downstream rim-seal edge and the blade leading edge, indicated by  $d/C_x$  and  $b/C_x$ , respectively. Additional experiments which were conducted without any cavity upstream the cascade served as a reference case. The parameter study regarding the axial design is based on the variation of  $d/C_x$  and  $b/C_x$ , respectively.

#### **Data Acquisition**

Pressure measurements are performed using a Netscanner of 16 integrated piezoresistive pneumatic sensors, each with a full-



Figure 3: RIM-SEAL GEOMETRIES

scale operating range of  $\pm 5psi$  and a maximum sampling of 500 measurements per second. The accuracy of the system is indicated with  $\pm 0.05\%$  of the full scale range. However, a regular in-house recalibration of the sensors improved the absoulte accuracy to  $\pm 5Pa$ . A SCXI-system is used to adjust the control valves as well as for the acquisition of the voltage data provided by the sealing mass flow sensor and the thermocouples, which monitor the fluid temperatures.

### **Pressure Measurements**

Measurements of total pressure were conducted using a calibrated five-hole probe with a spherical head of  $d/C_x = 0.04$ . This probe allows for the complete local determination of the total and static pressures as well as the relative flow directions at the same time. The probe is used in the non-nulling method and therefore requires an extensive calibration of the velocity and the two flow angles. The calibration method used for this work is similar to that described by Gerner et al.[30]. The calibration was performed for the yaw- and the pitch-angle in steps of 2 deg in a range of  $\pm$  20 degree. In oder to account for effects of a varying Reynolds number, this was repeated for several velocities representative of those expected during experiments. A fourth order

polynomial fit was used to correlate the pressure signals with the corresponding flow properties. The fitting error averaged  $\pm 0.25$ degree for the flow angles and  $\pm 13$  Pa for the pressure measurements. The larger the angular range comprised by the calibration the less accurate results the polynomial regarding the fitting of each data point. That means, if the calibration would comprise only a small angular range, indeed the resulting polynomial would be valid only for small flow angles, but at the same time it would provide an improved fitting for this angular range due to a smaller amount of data points. Therefore, the fitting error of the polynomial was considerably reduced during post-processing by adjusting the maximum required angular range of the calibration data iteratively to the measured local flow angle. The total pressures of the five-hole probe are referred to a total pressure  $P_{ref}$ which is measured by means of a pitot probe upstream the cascade. This allows for taking into account the possible variation in the operating point and the blockage effect caused by the fivehole probe. The five-hole probe was inserted into the test-section through slots in the upper channel wall and then traversed in two planes,  $1C_x$  upstream blade leading edge and  $0.25C_x$  downstream blade trailing edge. The corresponding measurement planes are indicated with P1 and P2 in Fig.4. Coordinate systems with index p clarify the local orientation of the probe. Index s indicates the flow direction. For upstream measurements the probe was aligned to the main flow direction, while for downstream measurements the probe orientation agreed with the blade design angle  $\alpha_2$  at the trailing edge. Measurements were confined to the



Figure 4: MEASUREMENT PLANES

center blade of the cascade and the probe was moved by means of a traversing system along one entire pitch and up to 86% of

the channel height. The resolution of the measurement grid in each plane was locally optimized and clustered corresponding to the local total pressure gradients. At each position 1000 samples were taken at the same time for each of the five sensors. An increment of the sample number as well as a further refinement of the measurement grid did not reveal any influence on the measurement results. The massflow averaged loss coefficient, which represents the crucial parameter to compare the aerodynamic performance of each rim-seal design, is defined as the total pressure difference across the cascade divided by the inlet total pressure, (Eq. 1).

$$\zeta'(z) = \int_{-0.5}^{+0.5} \frac{P_{tot,1} - P_{tot,2}}{P_{tot,1}} \cdot \frac{\rho \cdot U_2}{\dot{m}_2} \quad dy \tag{1}$$

For the calculation of the overall loss coefficient, the inlet total pressure is based on a massflow weighted value of the inlet total pressure to remove the loss caused by the low total pressure of the sealing flow:

$$\zeta' = \int_{0}^{h} \int_{-0.5}^{+0.5} \left( \frac{\dot{m}_m \cdot P_{tot,1} + \dot{m}_s \cdot P_{tot,s}}{\dot{m}_2 \cdot P_{tot,1}} - P_{tot,2} \right) / P_{tot,1}$$
$$\cdot \frac{\rho \cdot U_2}{\dot{m}_2} \quad dydz \tag{2}$$

In order to avoid probe interferences, measurements in the two planes were conducted one after another.

Furthermore, the local flow velocity as well as the xcomponent of the vorticity coefficient (Eq.3) were derived from the flow parameters gained by the five-hole probe measurements.

$$C\xi_x = \left(\frac{\partial w}{\partial y} - \frac{\partial v}{\partial z}\right) \cdot \frac{C_x}{\overline{U}_{ref}}$$
(3)

The local gradients were obtained by expressing the corresponding velocity-components measured at adjacent points with a fourth order polynomial which then was derived in y- and zdirection, respectively. The reference flow velocity  $\overline{U}_{ref}$  was measured at 75% channel height.

The three inner blades of the cascade are instrumented with pressure taps at three different span positions. Sensors at 50% span serve to verify the pressure distribution of the blade in its design point as well as to control the periodicity of the cascade. Sensors positioned close to the endwall, at 6% and 4% span, are give insight into the variation of blade loading under the influence of different rim-seal designs and sealing flow rates. Due to

the very thin design of the turbine blade, the instrumentation of the blade surface was confined to  $80\% C_x$ . The static pressure coefficient is defined as

$$Cp = \frac{\overline{P}_{tot,1} - p}{\overline{P}_{tot,1} - \overline{P}_{sta,2}}$$
(4)

, where p denotes the pressure measured at each pressure tap.

#### Laser-Doppler-Measurements

The interaction of the sealing flow with the main flow close to the rim-seal was investigated by 2D-laser-dopplermeasurements. These were conducted in the planes indicated with P3 and P4 in Fig.4 and up to 20% of the channel height. While the flow behaviour and hot-gas ingestion occuring in plane P3 is clearly affected by the upstream acting potential field of the blade leading edge, the flow in plane P4 is assumed to be more or less undisturbed. The laser-beams were introduced into the test-section through the window depicted in Fig.2. The system was used with on-axis backscatter light collection, which allows for the integration of transmitting and receiving optics in only one probe. It was operated with a Coherent INNOVA 90 argon-ion laser tuned for the blue (488.0 nm) and green line (514.5 nm). The optics consisted of a 60 mm front lens with a focal length of 400 mm, a beamsplitter and two photomultipliers. The measurement volume formed by the intersecting beams had a diameter of 116  $\mu$  m and a length of 2.44 mm. The entire system was moved on a three-axis traverse table. Both, the main flow and the sealing flow were seeded with liquid particles ejected through small pipes directly downstream the turbulence grid as well as into the lower part of cavity-module, respectively. The location of seeding represents a good compromise between a minimum flow disturbance caused by the seeding-pipes and a high quality of particle distribution. At each measurement point, a total of 6000 samples were taken at a maximum data rate of 8000 particles per second. Measurements close to the endwall revealed a considerable lower data rate, which could have been caused by disturbing wall reflections and a lower particle density.

# EXPERIMENTAL RESULTS Cascade Inlet Conditions

The cascade inlet conditions were measured 1  $C_x$  upstream the blade leading edge traversing the five-hole probe from  $0.02 \le z/h \le 0.86$  and along one entire pitch. Figure 5 shows the pitchwise averaged velocity profil. The data points between z/h = 0(no-slip) and z/h < 0.02, were extrapolated by a fourth order polynomial, as indicated by the dashed line. The boundary layer parameters are presented in Table 2. Table 2: BOUNDARY LAYER PARAMETERS  $C_x$  UP-STREAM LE

Velocity boundary thickness	$\delta/h$	0.109
Displacement thickness (mm)	$\delta^*$	1.875
Momentum thickness (mm)	θ	1.091



Figure 5: INLET BOUNDARY LAYER PROFILE AT  $1C_x$  UP-STREAM LE

#### Axial vs. Compound design

Figure 6 shows the mass flow averaged total pressure loss distribution over the channel height z/h at  $0.25C_x$  downstream the blade trailing edge for the axial and the compound design and for different sealing flow rates. Furthermore, the reference case without any cavity is added to the figure. The formation of secondary vortices inside the passage is reflected by two loss-peaks at z/h=0.35 and z/h=0.75. The measured distribution of the total pressure losses is not symmetric over the channel height, even for the reference case without any cavity upstream the cascade. This is mainly caused by the slots in the upper channel wall through which the five-hole probe is inserted into the test-section, as these revealed to affect the loss-peak at z/h = 0.72.

Regarding the results gained in the absence of sealing flow 6(a), it can be clearly seen, that only the presence of a cavity in front of the cascade already affects the loss behavior in a considerable way. Experiments with any rim-seal show higher losses compared to the reference case. The effect seems to be related mainly to the increased loss-peak around z/h = 0.32 and is confined to 60% of the channel height. A comparision between the axial design and the compound design denotes a slight improvement using the compound design. The reason for this improvement might be seen in the axial overlapping. It prevents ingestion and limits the interaction of fluids. However, there is a slight recovery of total pressure between  $0.05 \le z/h \le 0.15$  seen for the case of the axial design. The losses close to the endwall are identified only rudimentary. According to Treaster [31] a possible reason for this could be the wall-proximity effect, which alters the probe reading when approaching the wall. However, later investigations reported by Lee and Yoon [32] proved the wall-



(a) 0.0% sealing flow



(b) 0.5% sealing flow



(c) 1.0% sealing flow

Figure 6: MASS FLOW AVERAGED TOTAL PRESSURE LOSS

effect to cause only a slight increase in the static pressure reading while the total pressure reading remained nearly unaffected. Therefore, a feasible reason for the relative low loss-coefficient close to the endwall might be based on the formation of a new, thinner boundary layer caused by the the separation of the sec-



Figure 7: AREA AVERAGED EXIT FLOW ANGLE 0.25  $C_x$  DOWNSTREAM TE

ondary flows inside the passage, as detected experimentally by Moore [33] and Holley [34].

Figure 6(b) and 6(c) indicate for both designs an increment of total pressure losses, when sealing flow is ejected at a rate of 0.5% and when it is further increased to 1.0% of the main flow, respectively. At low sealing flow rates (Fig. 6(b)) a second loss-peak is about to be formed in the lower part of the channel at z/h = 0.26, while the right loss-peak has increased and expanded further into the center of the channel. However, differences between both designs are marginal. The tendency of increasing losses with higher sealing flow rate is confirmed by the results shown in Fig.6(c). At this stage the compound design performes better than the axial design, which is mainly based on the remarkable reduction of the left-side loss-peak. Furthermore, now also the compound design reveals a considerable recovery of total pressure in the area close to the endwall compared to the reference case.



Figure 8: OVERALL MASS FLOW AVERAGED TOTAL PRESSURE LOSS



Figure 9: TOTAL PRESSURE DISTRIBUTION 0.25 Cx DOWNSTREAM TE

The effect caused by the use different rim-seal designs upstream the cascade is also notable in the pitch-wise averaged exit flow angle, which is presented Fig.7. It can be clearly seen, that in comparison to the reference case without cavity, both rim-seal designs lead to an over- and under-turning in the area below and above z/h = 0.3, respectively. This effect intensifies with increasing sealing flow rate. However, considerable differences between the axial and the compound design can be seen especially in the absence of sealing flow, as shown by the dashed lines. The flow direction above z/h = 0.6 is not affected anymore.

Figure 8 presents the overall total pressure loss at  $0.25C_x$  downstream blade trailing edge. The losses are depicted for the three different rim-seal designs and for the different sealing flow rates, which have been investigated. The dashed lines indicate the increased loss for each rim-seal design if the impact of the low total pressure of the sealing flow was not removed from Eq.2. The level of minimum loss is shown by the horizontal line corresponding to the reference case without any cavity. It can be seen, that even without sealing flow, there is a considerable increase in overall loss when the axial or the compound rim-seal are inserted upstrem the cascade. Losses increase with higher sealing flow rate and the results show also that the compound design provides a certain improvement compared to the axial design. Consequently, for 1% sealing flow rate the compound design provides a reduction of overall total pressure loss of around 1.7%.

In consideration of the  $\zeta'$ -distribution presented in Fig.6, the focus of the following discussion is put on the area where the influence of the rim-seal design and the variation of sealing flow rates becomes apparent. Figure 9 shows the corresponding total pressure distribution of the different rim-seal designs for zero and 1.0% sealing flow rate measured with the five-hole probe at  $0.25C_x$  downstream blade trailing edge. The position of the trailing edge is located at y/s = 0. In Fig.9(a) the reference case without cavity is shown again for comparison. The contour plot indicates clearly the existence of two vortex cores. The vortex core at 25% blade height is considered to be the passage vortex (B). The vortex above at 30% blade height can be attributed to the wall-vortex (A). The region of higher losses close to the trailing edge of the blade could originate from a corner vortex (C).

The z/h-position of the loss-peak in Fig.6(a) is identical with the area where both vortices (A) and (B) agree in blade height. Thus, a possible variation of this loss-peak might be caused either by an increased pressure-loss generation inside one of these vortices or by a local displacement of these vortices. The "valley" formed between the two loss-peaks seen in Fig.6(c) is an indication for a spatial separation of the two vortices. Figure 9(b) and 9(c)indicate the influence of only the presence of a rim-seal. In comparison to the reference case, the possibility of flow exchange between the main flow and the cavity given by the axial design 9(b) leads to an apparent strengthening of the passage vortex and also slightly of the wall-vortex. In case of the compound design, this behaviour is less pronounced which is an explanation for the reduction of the loss-peak observed in Fig.6(a). In Fig.9(d) and 9(e) the resulting total pressures for 1.0% sealing flow rate are shown. It is apparent, that for both configurations the passage vortex is enlarged and that it contains an area of considerably reduced total pressure due to the high amount of sealing flow. Also the core of the wall-vortex at z/h=0.38 reveals a certain amplification. Nevertheless, the increment of the two loss-peaks in Fig.6(c) seems to be dominated mainly by the passage vortex. It can be noticed, that the existence of the left-side loss peak seen in Fig.6(c) is solely based on the reduction of total pressure inside the passage vortex and in its vicinity. Again, also the results seen in Fig.9 reveal a reduced total pressure drop for the compound seal and thus a better aerodynamic performance compared to the axial design.

The losses of total pressure can be correlated to the strength of secondary flows inside the passage, represented by the streamwise vorticity. This is shown in Figure 10 for the different rimseal designs. Furthermore, contour-lines of the total pressure are added to the plots. It can be clearly seen, that regions of reduced total pressure exhibit also a high magnitude of stream-wise vorticity. The large area of negative vorticity indicated with (D) represents the clockwise rotation of the passage vortex and corresponds to the loss core (B) in Fig.9(a). The position of loss core (A) and (C) agree with the area of positive vorticity, which are attributed to the weaker wall-vortex (E) and the small corner vortex (F) performing an anti-clockwise rotation. Comparing the reference case 10(a) with the experiments conducted for the axial design 10(b) and the compound design 10(c) at 1.0% sealing flow rate, it becomes apparent, that the increment of total pressure losses is related directly to a strengthening of the secondary flow's vorticity.

Fig.11 depicts the static pressure distribution along the blade surface for the axial design, the compound design and the reference case. The results gained for the reference case are shown separately in Fig.11(a) for the three different blade heights. The pressure distribution measured for the reference case at 50% blade height can be considered to be unaffected by the endwall flows. The pressure side of the blade does not reveal any remarkable changes at different blade heights. In contrast, the suction



Figure 10: VORTICITY 0.25 C<sub>x</sub> DOWNSTREAM TE

side reveals a considerable pressure-rise towards lower blade heights. This can be attributed to the lower velocities of the endwall boundary flows which result in a reduced blade loading. In the figures to the right, the reference case is compared to the results of the axial design and the compound design at 50% and 4% blade height. In the absence of sealing flow. This changes at 4% blade height, where also the effect of the different rim-seal designs becomes clear. While the compound design shows almost the same pressure distribution as the reference case, the use of the axial design has led to a further pressure-rise along the suction side. Fig.11(c) shows the corresponding measurements taken at 1.0% sealing flow rate. The ejection of sealing flow now also affects the pressure distribution at 50% blade height, which is reasonable as measurements of the total pressure losses showed also a variation up to 60% blade height. Furthermore, the ejection of sealing flow through the rim-seal shows an intensification of the pressure-rise along the blade suction side close to the endwall. While in case of zero percent sealing flow the maximum local pressure rise is about 12% (at  $x/C_x \approx 0.55$ ) in comparison to the reference case, the pressure rise in case of maximum sealing flow rate easily exceeds 25% compared to the reference case. However, this local pressure rise seems to be dominated by the sealing flow, as the difference between the pressure distributions of the two rim-seal designs, as seen previously, has now almost totally disappeared.

A deeper insight into the interaction of the sealing flow and main flow is given by the laser-doppler measurements upstream the blade leading edge close to the rim-seal. Figure 12 depicts a group of selected results, which have been gained for the different rim-seal designs and varied sealing flow rate in the measurement planes indicated in Fig.4 with P3 and P4, respectively. In the results shown, the magnitude of velocity refers to the difference regarding the velocities measured with the reference case. The direction of the corresponding velocity-component is indicated on the right side of the contour plots. Figure 12(a) and 12(b) show the velocity variation in z-direction in front of the blade leading edge (y/s=0) for the two rim-seal designs and still without any sealing flow. A comparison between the two results leaves no doubt about the advantage of the axial-overlapping provided by the compound design. While the variation around  $\pm 1$ [m/s] seen for the compound design does not represent any remarkable change compared to the reference case, the axial design shows a large area of negative velocity indicating a strong ingestion into the cavity driven by the potential field of the blade leading edge. Thus, the axial-overlapping represents an effective design feature to prevent the detrimental influence of the blade leading edge reaching far into the wheel space. If the cavity is sealed with a sealing flow rate of 1.0%, even in front of blade leading edge the ingestion seems to be reduced to a neglible amount, as can be seen in Fig.12(c) and 12(d). Almost no difference to the velocities measured for the reference case is seen



Figure 11: BLADE LOADING



Figure 12: DIFFERENCES IN FLOW-VELOCITIES

anymore. However, due to the pitch-wise pressure asymmetry the sealing is only achieved, if an increased penetration of sealing flow into the the main flow is accepted inside the passage. Experiments conducted for the same conditions but now in the measurement plane being shifted about y/s = 0.25 further into the passage demonstrate clearly the penetration of sealing flow into the main flow (Fig.12(e) and 12(f)). This agrees with the observations reported by Wellborn [35], who detected ejection of cooling air only in between the stator vanes of the investigated axial-flow compressor. In the present results the ejection of the sealing flow occurs mainly close to the downstream edge of the rim-seals. The area of sealing flow penetration seen for the compound design seems to exceed the one found for the axial design, but it is of a lower magnitude. The corresponding streamwise velocity-components are shown in Fig.12(g) and 12(h). Here, the influence of the ejected low-momentum sealing flow can be clearly seen. For both designs, as a result of the low-momentum sealing flow, the incoming boundary layer is further decelerated. The area covered by the low-momentum flow in the case of the compound design is larger than for the axial design and reveals a considerably stronger deceleration. This leads the author to the assumption that the rounded shape of the compound design allows the sealing flow to remain closer to the endwall with the consequence of a reduced aerodynamic spoiling of the main flow. However, further experiments are necessary to verify if this assumption holds along the entire pitch.

#### Influence of gap-size

Two further configurations based on the axial baseline design have been investigated, comprising a 100%-increment as well as a 80% -reduction of the axial gap-size d/Cx, respectively. The distance  $b/C_x$  between the downstream rim-seal edge and the blade leading edge remained the same. In Fig.13(a) the pitch-wise averaged total pressure loss distribution is plotted for the reference case and for all three axial-gap configurations measured with 0.0% sealing flow. Similar to the results seen before, also here the effects of a different rim-seal geometry become apparent mainly close to the endwall as well as in the region of the loss-peak. The results show, that an increment of the axial gap-size leads to an increase of the loss-peak as well as to an extension towards the center of the channel compared to the baseline configuration. In contrast, the configuration of reduced gap-size reveals a considerable improvement and agrees almost with the loss-distribution seen for the reference case. This is not surprising, as the configuration with the minimum axial gap-size provides a minimum possibility for hot-gas ingestion and consequently also for the flow exchange between the main flow and the cavity. As a consequence the wall-vortex and especially of the passage vortex are attenuated and reduced in their dimension. At the same time, the recovery of total pressure between  $0.05 \le z/h \le 0.15$  appears only for bigger axial gap-sizes.

Things change drastically, when the performance of the different axial design configurations is compared at 1.0% sealing flow rate, as shown in Fig.13(b). All configurations show a considerble increase of total pressure loss compared to the refrence case. Furthermore, almost no difference can be observed anymore between the loss-distributions of the three configurations. The previously mentioned advantage of the smaller gap-size seems to be vanished by the presence of the sealing flow. However, slight effects can be determined: Interestingly, the right-side loss-peak at z/h = 0.38 decreases with smaller gap-size, while the left-side loss-peak seems to increase with smaller gap-size. A comparison of the corresponding total pressure field revealed, that the lower right-side loss-peak is especially attributed to a weaker wallvortex. The higher left-side loss-peak is attributed to a stronger passage vortex and compensates for the advantage of a weaker wall-vortex. It is assumed that a configuration with smaller gapsize might benefit the formation of a jet when sealing flow is ejected. This leads to a deeper penetration into the main gas flow and thus to an increased aerodynamic spoiling manifested by a stronger passage vortex. As a consequence, if sealing air is ejected at 1.0% flow rate, the differences in overall total pressure



(b) Influence of gap-size,  $\dot{m}_s = 1.0\%$ 

Figure 13: VARIATION AXIAL GAP-SIZE

loss seems to be almost negligible for the three configurations.

#### Influence of axial gap-position

To investigate the influence of the axial gap-position on the design's aerodynamic performance, the gap has been held constant in size but moved to two different positions further upstream compared to the axial baseline configuration. Therefore the distance  $b/C_x$  (see Fig.3(a)) has been increased about to  $20\% of C_x$  and  $50\% of C_x$ , respectively. As can be seen in Fig.14(a), no remarkable effect is caused by the displacement of the axial gap, while no sealing flow is ejected through the axial rim-seal. The position and height of the loss-peak seen at z/h=0.34 are independent of the upstream position of the gap. If sealing air is ejected with 1.0% flow rate, again all configurations exhibit two loss-peaks. However, it seems that the displacement of the gap affects mainly the left-side loss-peak, which is dominated by the passage vortex. While the right-side loss-peak reveals only slight differences for the three configurations, the height of the left-side loss-peak is significantly reduced with increasing distance between the gap and the blade leading edge. It is assumed, that the displacement of the gap in upstream direction reduces on one hand the detrimental effect of the sealing flow and on the other hand the influence of the blade leading edge's potential field, resulting in a considerable weaker passage vortex.

### CONCLUSION

Two different rim-seal designs have been investigated regarding their effect on the aerodynamic spoiling in a linear turbine cascade. Extensive pressure measurements by means of a five-hole probe as well as laser-doppler-measurements have been conducted. The results confirm a remarkable impact of any rimseal geometry on the loss generation. Only the presence of a rim-seal has led to an increment of total pressure losses. This could be attributed to the exchange of flow occurring between the fluid inside the rim-seal cavity and the main flow, causing a detrimental modification of the boundary layer. If sealing flow is ejected through the rim-seal, a further amplification of the total pressure losses was identified for both rim-seal designs. A relative improvement was determined for the more complex compound design. This seems to be based on the axial overlapping and the rounded shape of the compound design. Laser-dopplermeasurements showed the axial overlapping to confine the ingestion into the cavity and thus reduce the possibility of flow exchange. The rounded shape might provide a more favorable conduction of the sealing flow leading to a reduced penetration into the main flow. Higher losses are based mainly on a strengthening of the passage vortex. Parametric studies on the simple axialgap design revealed possible improvements regarding its aerodynamic performance. A considerable attenuation of the passage



vortex was observed by increasing the distance between the gap and the blade leading edge. Further considerable improvements were gained by reducing the axial gap-size aiming at a reduction of the fluid exchanges. This design feature however provides only a benefit in absence of the sealing flow. With higher sealing flow rates, the small gap-size results in an intensified jetting of the sealing flow into the main flow.

0.7 0.8 0.9

0.5 0.6 z/h

(b) Influence of gap-position,  $\dot{m}_s = 1.0\%$ 

Figure 14: VARIATION AXIAL GAP-POSITION

#### ACKNOWLEDGMENT

0.

0.2 0.3 0.4

The present investigations were supported by the European Commission within the Framework 6 Programme, Research Project Main Annulus Gas Path Interactions (MAGPI), AST5-CT- 2006-030874. This financial support is gratefully acknowledged.

#### REFERENCES

- Gallier, K. D., Lawless, P. B., and Fletter, S., 2004. "Development of the unsteady flow on a turbine rotor platform downstream of a rim seal". ASME Turbo Expo 2004 (GT2004-53899).
- [2] de la Rosa Blanco, E., Hodson, H. P., and Vazquez, R., 2009. "Ef-

fect of the leakage flows and the upstream platform geometry on the endwall flows of a turbine cascade". *Journal of Turbomachinery*, Vol. 131.

- [3] Bohn, D. E., Decker, A., and Ohlendorf, N., 2006. "Influence of an axial and radial rim seal geometry on hot gas ingestion into the upstream cavity of a 1.5-stage turbine". ASME Turbo Expo 2006 (GT2006-90453).
- [4] Bohn, D. E., Rudzinski, B., and Suerken, N., 1999. "Inluence of rim seal geometry on hot gas ingestion into the upstream cavity of an axial turbine stage". ASME Turbo Expo 1999 (GT-248).
- [5] Phadke, U., and Owen, J., 1988. "Aerodynamic aspects of the sealing of gas turbine rotor-stator sytstem, part 1: The behaviour of simple shrouded rotating-disk systems in a quiescent environment". *Int. J. Heat and Fluid Flow*, Vol.9, No.2.
- [6] Phadke, U., and Owen, J., 1988. "Aerodynamic aspects of the sealing of gas turbine rotor-stator sytstem, part 2: The performance of simple seals in a quasi-axisymmetric external flow". *Int. J. Heat* and Fluid Flow, Vol.9, No.2.
- [7] Gentilhomme, O., Hills, N., and Turner, A. B., 2003. "Measurement and analysis of ingestion through a turbine rim seal". ASME Journal of Turbomachinery, Vol. 125, pp. 505–512.
- [8] Bohn, D. E., Decker, A., Ma, H., and Wolff, M., 2003. "Influence of sealing air mass flow on the distribution in and inside the rim of the upstream cavity of a 1.5-stage turbine". ASME Turbo Expo 2003 (GT2003-38459).
- [9] Bhavani, S., Khulnani, L.-C., Khodadadi, J., and Goodling, J., 1992. "Effective sealing of a disk cavity using a doubled-toothed rim seal". ASME Turbo Expo 1992.
- [10] Geis, T., Wiebelt, Kim, A., and S. Wittig, S., 1999. "Ingress determination by means of laser light scattering inside a rotor-stator system". ASME Turbo Expo 1999 (GT-247).
- [11] Bunker, R., Metzger, D., and Wittig, S., 1990. "Local heat transfer in turbine disk-cavities, part i: Rotor and stator cooling with hub injection of coolant". ASME Turbo Expo 1990 (GT90-25).
- [12] Montomoli, F., Massini, M., Maceli, N., Cirri, M., Lombardi, L., Ciani, A., D'Ercole, M., and De Porsperis, R., 2006. "Interaction of wheelspace coolant and main flow in a new aeroderivative lpt". *ASME Turbo Expo 2006*.
- [13] McLean, C., Camci, C., and Glezer, B., 2001. "Mainstream aerodynamic effects due to wheelspace coolant injection in a highpressure turbine stage, part i: Aerodynamic measurements in the stationary frame". ASME Turbo Expo 2001 (GT2001-0119).
- [14] Simon, T., and Piggush, J., 2006. "Turbine endwall aerodynamics and heat transfer". *Journal oF Propulsion and Power*, Vol. 22, No.2, pp. 301–312.
- [15] Sharma, O., and Butler, T., 1986. "Prediction of endwall losses and secondary flows in axial flow turbine cascades". ASME Turbo Expo 1986.
- [16] Burd, S. W., and Simon, T. W., 2000. "Effects of slot bleed injection over a contoured endwall on nozzle guide vane cooling performance: Part 1 - flow field measurements". ASME Turbo Expo 2000 (GT2000-199).
- [17] Sieverding, C., 1985. "Recent progress in the understanding of basic aspect of secondary flows in turbine blade passages". *Journal* of Engineering for Gas Turbines and Power, Vol. 107, pp. 248– 257.

- [18] Langston, L., 1980. "Crossflow in a turbine cascade passage". *Journal of Engineering for Power*, Vol. 4, pp. pp. 866–874.
- [19] Wang, H. e. a., 1997. "Flow visualization in a linear turbine cascade of high performance turbine blades". *Journal of Turbomachinery*, Vol. 119, pp. 1–8.
- [20] Herzig, H., Hansen, A., and Costello, G., 1953. "A visualization study of secondary flows in cascades". NACA Report 1163.
- [21] Kost, F., and Nicklas, M., 2001. "Film-cooled turbine endwall in a transonic flow field: Part i - aerodynamic measurements". *Journal* of *Turbomachinery*, Vol. 123, pp. 709–719.
- [22] Popovic, I., and Hodson, H., 2010. "Aerothermal impact of the interaction between hub leakage and mainstream flows in highlyloaded hp turbine blades". ASME Turbo Expo 2010.
- [23] Schuepbach, P., Abhari, R., Rose, M., and Gier, J., 2010. "Sensitivity of turbine efficiency and flow structures to varying purge flow". *Journal of Propulsion and Power*, Vol.26, No.1.
- [24] Hunter, S. D., and Manwaring, S. R., 2000. "Endwall cavity flow effects on gaspath aerodynamics in an axial flow turbine: Part 1 experimental and numerical investigations". ASME Turbo Expo 2000 (GT2000-651).
- [25] Demargne, A., and Longley, J., 2000. "The aerodynamic interaction of stator shroud leakage and mainstream flows in compressors". ASME Turbo Expo 2000 (GT2000-570).
- [26] Girgis, S.and Vlasic, E., Lavoie, J.-P., and Moustapha, S., 2002. "The effect of secondary air injection on the performance of a transonic turbine stage". ASME Turbo Expo 2002 (GT2002-30340).
- [27] Ong, J. H., Miller, R. J., and Uchida, S., 2006. "The effect of coolant injection on the endwall flow of high pressure turbine". ASME Turbo Expo 2006 (GT2006-91060).
- [28] Reid, K., Denton, J., Pullan, G., Curtis, E., and Longley, J., 2006. "The effect of stator-rotor hub sealing flow on the mainstream aerodynamics of a turbine". ASME Turbo Expo 2006.
- [29] Schuler, P., Kurz, W., Dullenkopf, K., and Bauer, H.-J., 2010. "The influence of different rim seal geometries on hot-gas ingestion and total pressure loss in a low-pressure turbine". ASME Turbo Expo 2010.
- [30] Gerner, A., Mauerer, C., and Gallington, R., 1984. "Non-nulling seven-hole probes for high angle flow measurement". *Experiments* of Fluids 2, pp. 95–103.
- [31] Treaster, A., and Yocum, A., 1979. "The calibration and application of five-hole probes". *ISA Transactions*, Vol.18, pp. pp.23–34.
- [32] Lee, S., and Yoon, T., 1999. "An investigation of wall-proximity effect using a typical large-scale five-hole probe". *KSME International Journal*, Vol.13, pp. 273–285.
- [33] Moore, H., and Greory-Smith, D., 1995. "Turbulence and transition in secondary flows in a turbine cascade". AGARD PEP 85th Symposium.
- [34] Holley, B., Becz, S., and Langston, L., 2006. "Measurements and calculation of turbine cascade endwall pressure and shear stress". *Journal of Turbomachinery*.
- [35] Wellborn, S. R., 2001. "Details of axial-compressor shrouded stator cavity flows". ASME Turbo Expo 2001 (GT2001-0495).