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SLOT INFLUENCE IN BOUNDARY LAYER SUCTION FOR SECONDARY FLOW CONTROL IN A HIGH SPEED COMPRESSOR CASCADE

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Abstract

A numerical and experimental study is presented in order to clarify more the influence of a suction slot design with respect to mass flow rate and slot area in a compressor with active flow control for secondary loss diminution. While the numerical simulations unveil no influence of the shape at constant mass flow rates, the experimental investigations show such a dependency of the suction efficiency. For the study a suction slot in side walls has been examined, the peacock slot in the junction between vane suction side and side wall at 50% to 100% chord, with a slot width of 0.6mm, 1mm and 2mm. The parameter study has in part been done numerically; the paper concentrates on the comparison with the experimental results at Mach number 0.67 and Reynolds number of 560.000.

This experimental study shows the dependency of the efficiency of the cascade with the mass flow and the slot width, respectively. Both parameters can be varied independently. It is shown that the slot width has a remarkable influence on the efficiency. With larger slot widths, the mass flow rate can be increased. While a slot width of 0.6mm allows for only about 2% of the main mass flow to be sucked from the cascade, with a slot width of 2mm more than 5% of the main mass flow can be drawn off the compressor. An increased suction rate in combination with an appropriate slot design decreases the total pressure loss, as soon as the suction rate is high enough to guarantee a constant suction mass flow in the slot. Otherwise, a recirculation appears that destroys the effect of the flow suction.

Key words: compressor cascade, secondary flow, active flow control, boundary layer suction

NOMENCLATURE

IPC		Intermediate pressure compressor
RANS		Reynolds averaged Navier-Stokes equations
NACA		National Advisory Committee for Aeronautics
Ma	[-]	Mach number
Re	[-]	Reynolds number

c	[mm]	vane chord length
h	[mm]	vane height
t	[mm]	cascade pitch
u	[mm]	cascade pitch coordinate
z	[mm]	vane height coordinate
m_{asp}	[%]	aspiration rate
β_1	[°]	inflow angle
β_2	[°]	outflow angle
ζ	[-]	total pressure loss coeff.
η	[-]	stator stage efficiency

INTRODUCTION

Axial flow compressors are used in stationary and flying gas turbines all over the world for electricity generation and aero engine powering. They are a robust part of machinery and have been refined in the past decades, where numerical simulations and new experimental techniques have their share in efficiency advances. The compressor flow phenomena have been understood well, different ways have been found to ensure qualitatively good flow through the passages. In the future, great leaps in efficiency will only be achieved by introducing active flow control techniques into axial compressors. Examples can be flow suction or blowing [1;3;5], implementation of plasma or other actuators [4] as well as combinations of the latter.

In the past, aero engine developers have been very reserved using active flow control in their machines. Sufficiently low kerosene prices and non reliable technology have been reasons for the engine manufacturers' focus on conventional engine design. The latest developments in the oil prices as well as robust active flow control mechanisms may in the near future be a trigger for their implementation in an actual machine.

The German Research Foundation (DFG) funds a project in which active flow control (AFC) is examined regarding efficiency and robustness in a modern axial compressor. In this context, investigations in a high speed compressor cascade have been conducted at the German Aerospace Center (DLR), Berlin. The compressor cascade (see Table 1), consisting of five linear aligned NACA65 IPC stator vanes, has been equipped with steady boundary layer suction in order to enhance the flow quality and reduce pressure loss due to secondary flow and separation.

Ma_1	Re	chord	span	pitch	β_1	β_s
0.67	560.000	40mm	40mm	22mm	132°	112.5°

Table 1: Design conditions of the examined NACA65-K48 cascade

EXPERIMENTAL SETUP

All experiments were carried out at the high-speed stator cascade wind tunnel of the German Aerospace Center (Dept. of Engine Acoustics, Inst. of Propulsion Technology) in Berlin. The test rig, shown in Fig. 2, has a rectangular cross section of 40 mm width and 90 mm height at the cascade inlet. The nozzle with a contraction ratio of 1:218 accelerates the flow to Mach number 0.7. A Reynolds number of up to 0.6×10^6 can be obtained.

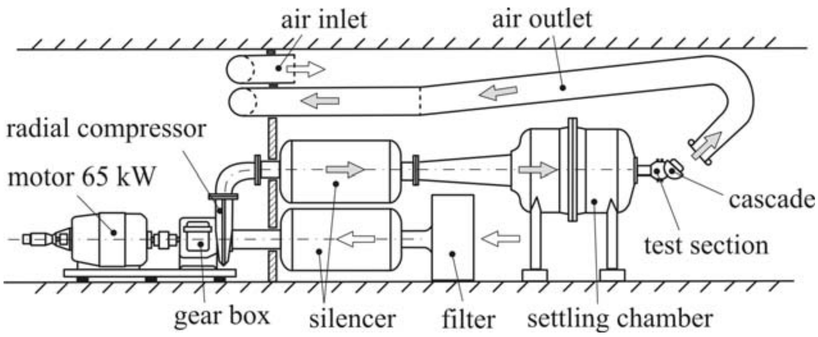


Fig. 1: DLR High Speed Cascade Wind Tunnel, Berlin

The boundary layer height of each inlet wall can be adjusted independently. The inlet length can be extended, so that the boundary layer thickness can be set from 3 to 12 mm. By aspiration the thickness can be decreased to less than 2 mm. The upper and lower boundary layers are aspirated to ensure periodic flow to the compressor cascade. Periodicity of the flow is monitored by a row of static pressure probes in measurement plane 1 (Fig. 2).

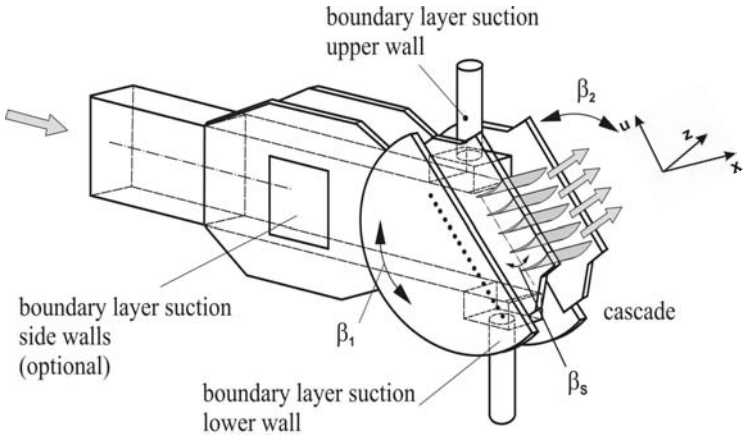


Fig. 2: Measurement setup and geometrical definitions

The inlet angle of the cascade β_1 can be adjusted geometrically from 122° to 146°. In the aerodynamic design condition measurement, β_1 is set to 132°. A detailed description of the wind tunnel and measurement procedures can be found in Liesner et al. [7].

Measurement Technique and accuracy

The measurement technique used is a rake of 26 Pitot tubes, which are traversed in circumferential direction in a measurement plane 16mm (0.4 times chord) behind the stator exit. The pressure information is used for a total pressure loss and pressure rise calculation and can also be basis for a determination of the stator stage efficiency, for which an exemplary rotor is included. The outflow angle is obtained in a set of Conrad angle probes, which are also traversed in the measurement plane. The accuracy of the whole apparatus is better than +/- 1.2%, this can also be found in the setup description.

NUMERICAL ANALYSIS

A numerical study by means of steady Reynolds-averaged Navier-Stokes simulations (RANS) is performed at the Institute of Fluid Mechanics and Engineering Acoustics (ISTA) of the Berlin Institute of Technology. Numerical results offer deeper insight into the separated 3D flow topology enabling determination of appropriate suction orifice positions. With regard to the active control method the numerical approach allows easy variation of the suction slot geometry. In this paper results of the basic flow configuration are evaluated against experimental data.

Computational Approach

An implicit pressure based solver with a fully conservative approximation of the Reynolds-averaged Navier-Stokes equations is employed in steady mode. The code, named “ELAN”, is based on curvilinear coordinates and uses cell-centered collocated storage arrangement on semi-block-structured grids for all quantities. The Menter SST-k-w model [8] is used for turbulence treatment. Transition is fixed at both, pressure and suction side. Therefore the production term of the turbulence model is triggered by a step function which is zero within the laminar and one within the turbulent regime. The continuity equation is conserved by the SIMPLE-algorithm whereby the decoupling of pressure and velocity is prevented through a generalised Rhie & Chow interpolation [11]. Due to symmetrical reasons only half of the span is considered within the computational approach. The grid is created with G3DMESH [10], a tool developed at DLR Cologne. In order to ensure sufficient resolution the structured mesh consists of 1.23 million control volumes, whereas 65 points are in spanwise direction over half of the span. Approximately 25 grid lines resolve the boundary layer up to the viscous sublayer in the wall normal direction with a maximum dimensionless wall distance of y+ max = 1:77. The computational mesh and the distribution of the dimensionless wall distance are depicted in Fig. 5 showing only every second grid cell for improved insight. The desired inlet flow conditions are achieved by variation of the static pressure p2 imposed at the outlet boundary located 2.5 times the chord length downstream of the blade. The boundary layer of the incoming flow is accounted for by means of a channel flow simulation ensuring accordance with the measured wall normal total pressure profile as shown in Fig. 6.

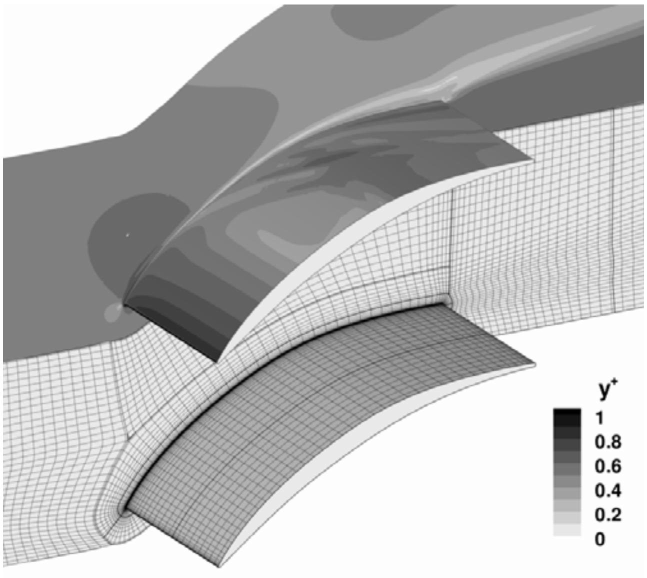
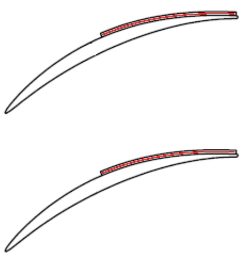


Fig. 3: Computational Grid and dimensionless Wall-distance

BOUNDARY LAYER SUCTION



The cascades are equipped with small slots in the side walls, through which boundary layer flow can be extracted and sucked out. The shape of the slots is following the design by R.E. Peacock from 1965 [9], where he was able to extinguish the whole amount of secondary flow out of the cascade in a low speed environment. The slots are positioned from 50 to 100% chord in the junction of the vane suction side and side wall have a thickness of 0.6 (original Peacock) to 2mm (1.5 - 5% chord).

In the experimental investigation, the suction is realized with 2-dimensional slots in the 10mm side walls, which are connected to a system of tubing. An additional side channel blower generates the low pressure for the suction.

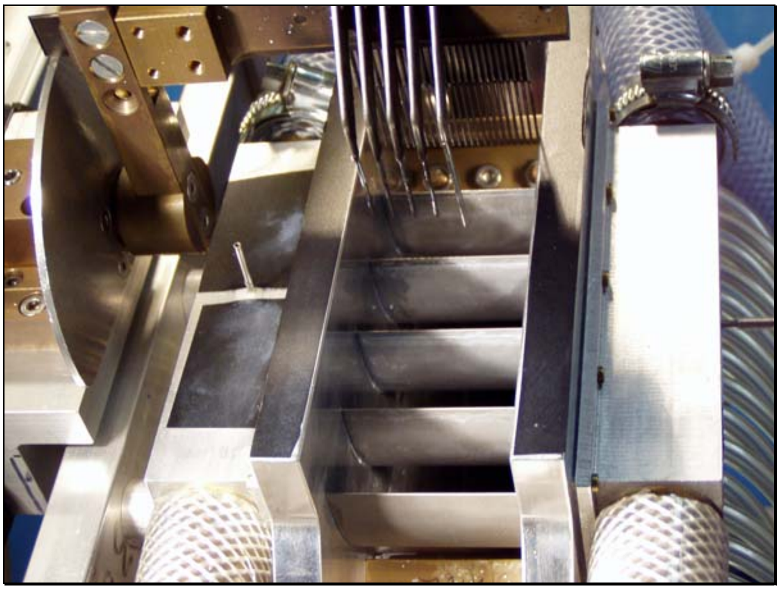


Fig. 4: Cascade with suction tubing and measurement pressure probes

In the numerical simulations, the suction orifices have been generated as a rededication of wall cells. A block has been created as an inlet with given velocity distribution. The velocity in negative wall normal direction has been determined in a way that the mass flow matches the required values (see Fig. 3). An outflow has not been chosen due to the lack of pressure boundary condition behind the orifices. Existing cells could be used for that purpose and the channels inside the walls have not been modelled in this attempt. Four different slot widths have been examined, 0.6mm (the standard Peacock geometry) and wider slot with 1, 1.3 and 1.6mm slot width.

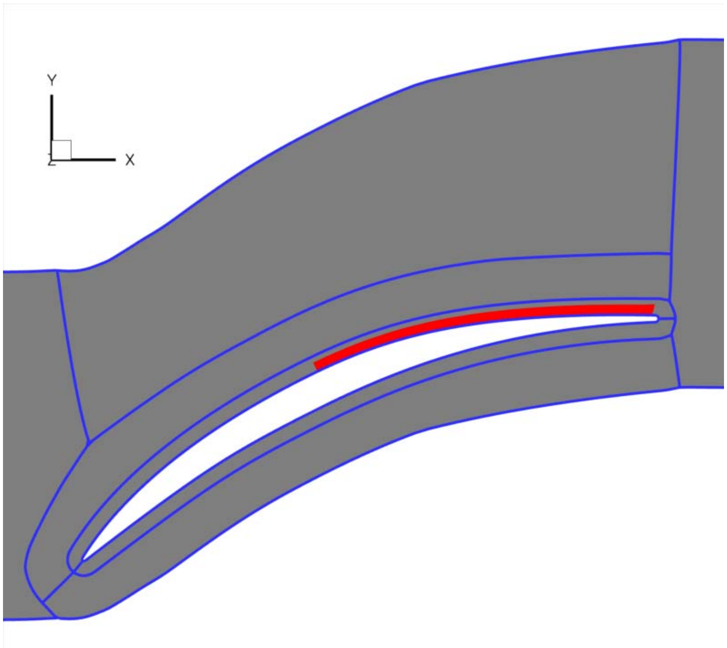


Fig. 5: Definition of cells as Inlet (red) with negative velocity form the suction orifices

RESULTS

The simulations and the experiments show great accordance in almost all fields of the investigation. The results of the simulations for the suction meet the results quantitatively and qualitatively. Figures 6 and 7 illustrate this accordance very vital. Shown are the total pressure losses of the base flow and the cascade with flow suction in the Peacock cascade.

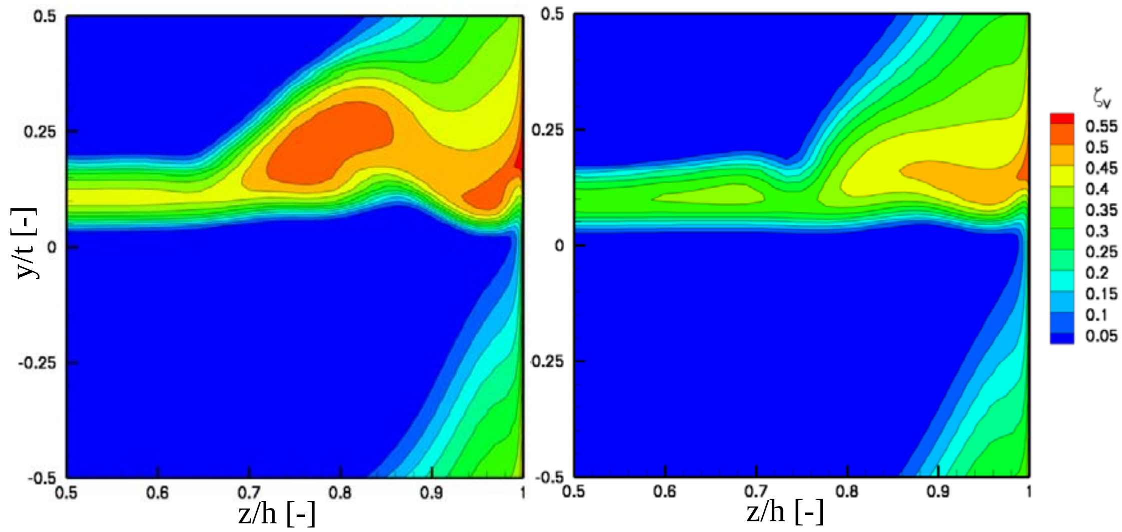


Fig. 6: Base flow cascade (left) and Peacock slot (right) at suction mass flow rate 2%, slot width 0.6mm, Mach 0.67, simulation

In the graphic above the total pressure loss coefficient ζ is plotted versus the vane height and pitch. In the area of pitch y/t=0 the trailing edge of the vane is located, the position of the wall is z/h=1. The vanes suction side is located above the trailing edge.
The numerical simulations show a wide area of reduced pressure loss in the junction of wall and side wall. This is the area where the secondary flow suction takes effect. The vortex system is influenced and weakened. The total pressure loss coefficient is decreased by an amount of 22% in this case.

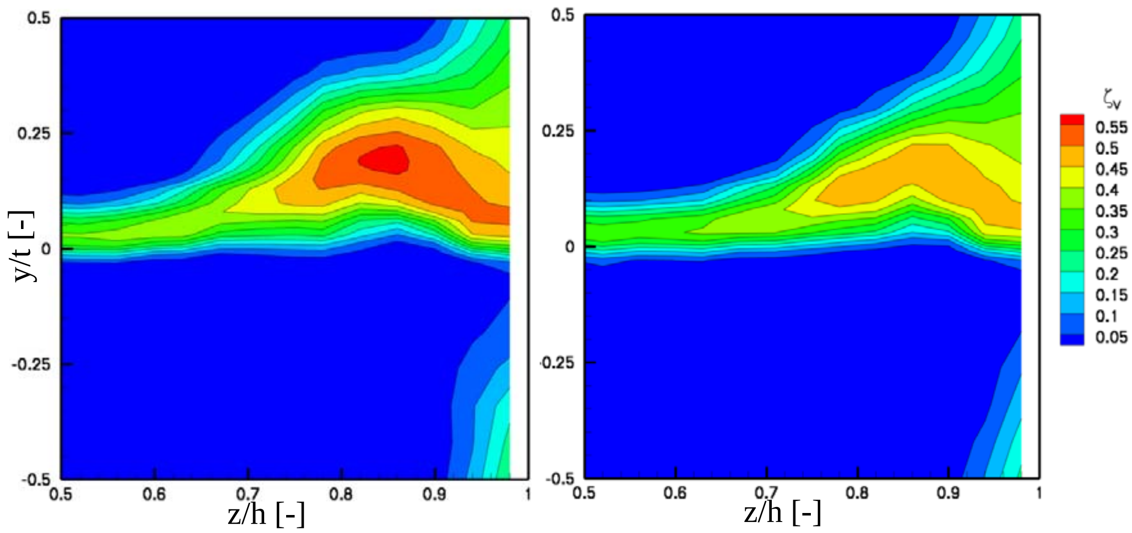


Fig. 7: Base flow cascade (left) and Peacock slot (right) at suction mass flow rate 2%, slot width 0.6mm, Mach 0.67, experiment

The experimental results show the same behavior as mentioned above. The comparison between these standard Peacock suction slot investigations has already been examined earlier by the authors [2;6].

The variation of the slot width has been investigated now experimentally and numerically. In the numerical results the suction slot has been modeled as a 2dimensional area on the cascade side walls. In the experiments, there was of course need for detailed construction of suction slots through the side walls, the wall of 10mm thickness has a 3D slot and the system includes tubing and vacuum chambers.

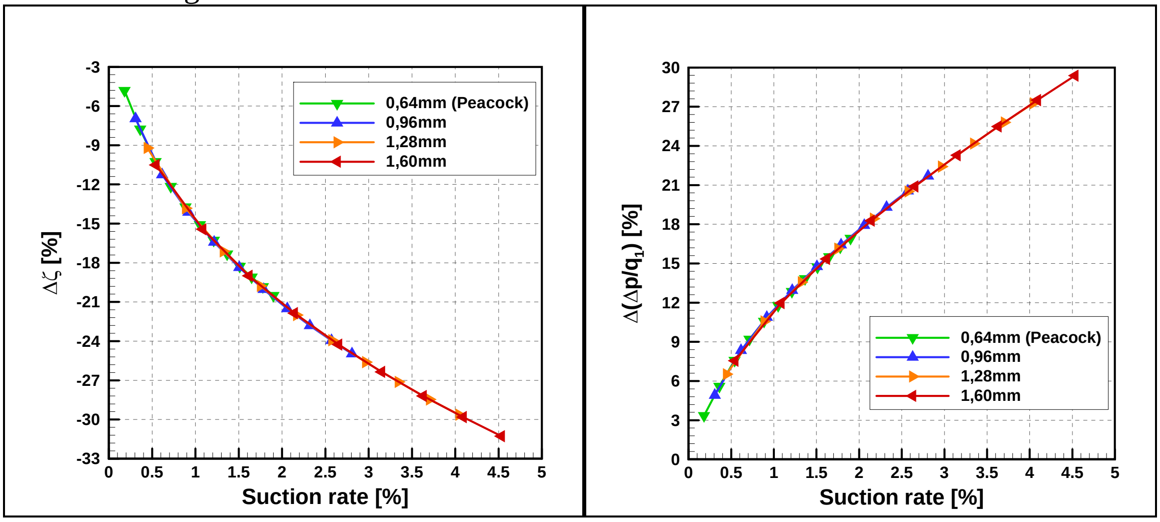


Fig. 8: Total pressure loss vs. Suction rate, Peacock slot, simulation data

Figure 8 is the compendium of the calculations with different suction slot widths. For different widths, different suction ratios could be realized. While with a standard Peacock slot only less than 2% of the main mass flow could be sucked out of the cascade, with larger opening areas the mass flow rate was raised to up to 5%.

In the numerical simulations, all curves are identical for a constant mass flow ratio. The total pressure loss (left) drops with increasing suction rate, as the secondary flow influence disappears. The vortex system is less distinct and of less significance. There is no dependency on the opening area visible at all. It is very nice to see that the positive effect of the suction starts immediately. With commencing suction, the pressure loss is beginning to be decreased. The pressure rise, contrary to that, increases with rising mass flow suction, no matter what the slot width is. This behavior is due to the decrease of the nozzle effect, that huge boundary layers produce. Since the secondary flow separated area is reduced, the main flow can distribute through the whole passage. That is why the static pressure in the outflow is higher than in the cases with more separation and higher boundary layers. This is the case with the higher suction rates; the more low-momentum flow is sucked out of the cascade, the better the outflow behavior.

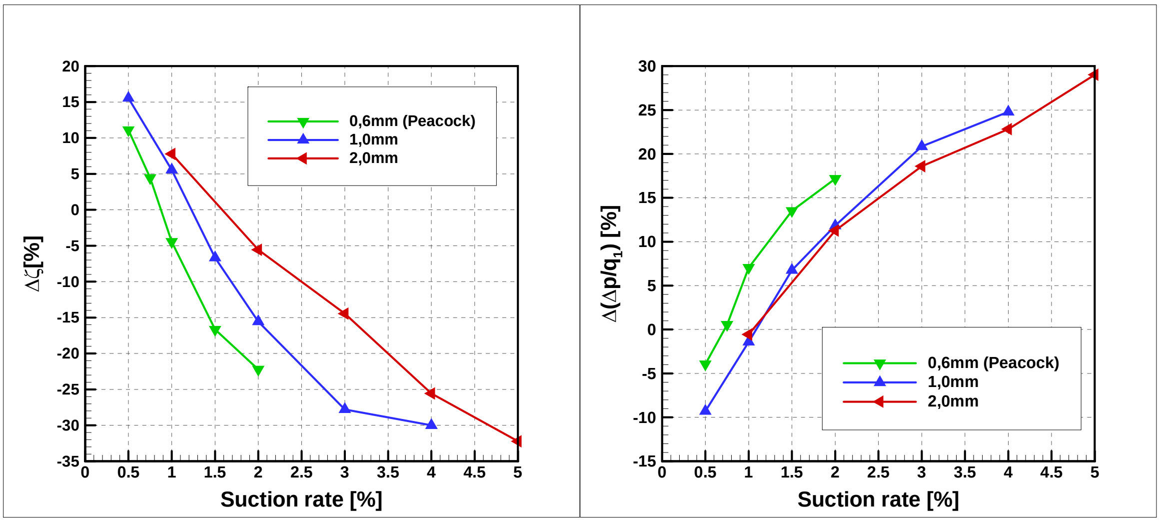


Fig. 9: Total pressure loss vs. Suction rate, Peacock slot, simulation data

The experimental results are shown in Figure 9. Three opening widths have been examined. Contrary to the previous mentioned numerical study, in the experiments there has been a noticeable difference between the suction through different suction slot orifices.

In the left part of Figure 9, the total pressure loss is shown for all three orifice areas. In the case with 2% suction, the smallest, original Peacock slot shows a total pressure loss decrease of 22%, as has been shown in previous investigations. But the wider slots cannot follow this trend and show significantly less pressure loss decrease. The effect that takes place here is a recirculation appearing in the suction slot, as soon as the suction velocities in the slot are sufficiently low. When the suction rate is high enough to ensure full area high speed flow in the suction slots, this effect does not appear. In contrast to the numerical simulations, a positive effect of the suction is only achieved upwards of a minimum suction rate, as is about 1 – 1.5% for all investigations. This is a sign for the recirculation appearing in the suction slots, when the negative pressure forcing the flow out of the cascade is not high enough to ensure a permanent outflow through the orifices.

CONCLUSION

A numerical and experimental study has been conducted in order to clarify the influence of the slot dimension as a function of the mass flow drawn out of the cascade. A parameter study is performed and the results are shown. For a given geometry of a suction slot, the orifice area has been varied together with the suction mass flow rate.

Experiments have been conducted in a subsonic high speed environment at Mach=0.67 and Re=560.000 at a cascade wind tunnel with NACA65 IPC vanes. Numerical RANS simulations have been carried out additionally in order to learn about the modeling of these active flow control techniques.

The paper shows a dependency of the efficiency of the suction slots with the area. For small mass flow rates, it is evident that the orifice area must be as small as possible. There is a minimum suction rate of 1-1.5% of the overall mass flow that has to be sucked out of the cascade for a positive effect at all. Below these suction rates, a flow recirculation and flow out of the suction orifices is detected. Above this border, the positive effect is increased with the mass flow rate. The maximum mass flow rate is determined by the size of the slot. As soon as Ma=1 is reached in the slot, the rate can not be increased and the slot is blocked. The effect of the suction is higher for a smaller opening area of the slot for a given mass flow rate. For large suction rates the slot has to be designed very narrow. The nearer the suction speed is at Ma=1, the better its efficiency.

In the numerical investigations it becomes obvious that it is required to model the suction slot in as much detail as possible. A constant suction velocity can only be assured, when the flow does not have the opportunity to recirculate in the suction slots, either by not modeling them in the numeric simulations or by attaching high negative pressure to force flow out of the cascade with high speeds.

ACKNOWLEDGEMENTS

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REFERENCES

[1] Gbadebo, S.A. (2008): "*Control of Three Dimensional Separations in Axial Compressors by Tailored Boundary Layer Suction*"; Journal of Turbomachinery, 2008

[2] Gmelin, C., Thiele, F., Liesner, K. and Meyer, R. (2011): "*Investigations of secondary flow suction in a high speed compressor cascade*". ASME Turbo Expo 2011, Vancouver, Canada. GT2011-46479

[3] Gümmer, V. (2008): "*Numerical Investigation of End Wall Boundary Layer Removal on Highly Loaded Axial Compressor Blade Rows*"; Journal of Turbomachinery, 2008

[4] Kerrebrock, Epstein, Merchant (2007): "*Design and Test of an Aspirated Counter Rotating Fan*"; Journal of Turbomachinery, 2007

[5] Leishman, Cumpsty, Denton (2007): "*Effects of Bleed Rate and Endwall Location on the Aerodynamic Behavior of a Circular Hole Bleed Off-Take*"; Journal of Turbomachinery 2007;

[6] Liesner, K., Meyer, R., Lemke, M., Gmelin, C. and Thiele, F. (2010): "*On the efficiency of boundary layer suction in a compressor cascade*", ASME Turbo Expo 2010, Glasgow, UK. GT2010-22336

[7] Liesner, K. and Meyer, R. (2008): "*Experimental Setup for detailed Secondary Flow Investigation by Two-Dimensional Measurement of total pressure loss coefficients in compressor cascades*", Proceedings of the XIX VKI Symposium, Sint Genesius-Rode, Belgium, 2008.

[8] Menter, F. (1992): "*Performance of popular turbulence models for attached and separated adverse pressure gradient flows*"; AIAA Journal, 30(8):2066–2072, 1992.

[9] Peacock, R.E. (1965): "*Boundary-layer suction to eliminate corner separation in cascades of airfoils*"; RM 3663; University Engineering, Cambridge

[10] Weber, A. (2008): „*G3DMESH v4.5.4*"; Manual; DLR, Institut für Antriebstechnik, Köln, 2008.

[11] Xue, L. (1998): „*Entwicklung eines effizienten parallelen Lösungsalgorithmus zur dreidimensionalen Simulation komplexer Strömungen*" Ph.D. Thesis, Technische Universität Berlin, 1998.