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NON-SYMMETRICAL BOUNDARY LAYER SUCTION IN A COMPRESSOR CASCADE

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In order to simulate non-symmetrical boundary layer suction in an annular compressor, a cascade investigation has been performed with single-sided suction slots only. A preceding investigation had revealed a high potential for loss reduction by two different types of boundary layer suction. The experimental investigation was performed with five NACA 65-k48 stator blades at the design Mach number of 0.67 and Reynolds number of 560.000. The two investigated suction geometries are a narrow slot following the design of Peacock and a wider slot of own origin, both slots are positioned on one side of the passage only. The Peacock slot is placed in the corner between suction side of the vane and the side wall, the wider slot is positioned from suction side to pressure side following the side wall's flow detachment line. With half the suction rate of the preceding investigation the efficiency of the cascade could still be enhanced. In the case with 2.5% suction rate the total pressure loss coefficient of the full passage was decreased by 13%, in the case with 1% suction rate the loss coefficient was decreased by even 10%. The outflow of the cascade is as expected no more symmetrical and the one sided suction has no large impact on the flow of the opposite side of the passage.

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

Nomenclature

Geometric and flow quantities

M, Re	–	Mach and Reynolds number, [–]
c, h	–	vane chord length and vane height, [m]
m	–	mass flow, [kg/s]
p	–	pressure, [Pa]
t	–	cascade pitch, [m]

u, x	–	cascade pitch and axial coordinate, [m]
z	–	vane height coordinate, [m]
β	–	cascade flow angle, [°]
ζ	–	total pressure loss coefficient, [–]

Subscripts

1	–	cascade inlet
2	–	cascade outlet

1. Introduction

Efficient compressors provide a high pressure ratio in combination with a low number of stages. Not only aerodynamic aspects have to be taken into consideration, also the weight plays a significant role in engine design. In the recent years, much effort has been done to improve compressor aerodynamics in order to enhance the turbo-machine efficiency.

Aerodynamic losses in axial compressors are dominated by the secondary flow losses (Scholz, 1956). The higher the pressure ratio in the compressor stage, the more likely the flow will detach in the corner of the suction side of the blade and endwall (Cumpsty, 2004). This secondary loss is triggered by movement of the wall boundary layer fluid to the suction side due to the strong pressure gradient. The flow on the suction side, facing an adverse pressure gradient, can not overcome the pressure and detaches from the surface. In the detached area in the corner, a vortex system develops and leads to strong pressure losses (Kang and Hirsch, 1991).

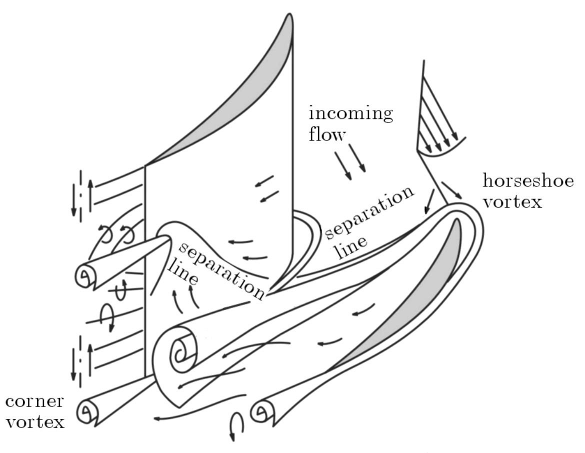

Fig. 1. Vortex system in a compressor cascade (Kang and Hirsch, 1991)

Three-dimensional blade design and casing treatment are the methods used most often for passive secondary flow control in the machine. For compressor cascades, there are different passive techniques available (Hergt *et al.*, 2006; Lin and Howard, 1989). The problem of secondary flow between the pressure and suction side of the passage remains, but decreases.

There is, in fact, a way to remove the secondary flow on the whole out of the compressor. Suction of the secondary flow has been proven by Peacock (1965) already in the 1960's to be able to make secondary flow losses disappear from a compressor with very low Mach numbers. An optimized suction slot is used to draw 0.1% of flow out of the passage and the secondary flow effect in the wake is no more found. That effect in low Mach number cascades has been used in a recent investigation at the German Aerospace Center (DLR) in Berlin to remove the secondary flow from a high speed cascade effectively. Different geometries have been used in order to decrease the pressure loss coefficients of such a compressor cascade. The full results of suction measurements can be found in a recent publication (Liesner and Meyer, 2010).

The solution found in this investigation is only theoretically a good solution. An annular compressor is a rotating machine. The stator vanes are connected as cantilevers to the casing, the rotor blades rotate with the hub at several thousand revolutions per minute. Therefore, it is very unlikely to suck both the hub and casing boundary layer symmetrically on fixed positions. In order to get insight into the behaviour of the flow in such non-symmetrical conditions, this investigation has been performed.

2. Experimental setup

All experiments were carried out at the high-speed stator cascade wind tunnel of the German Aerospace Center (DLR) 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 up to Mach number 0.7. A Reynolds number of $0.6 \cdot 10^6$ can be obtained.

The boundary layer height of each inlet wall can be adjusted independently (see Fig. 3). The upper and lower boundary layers are sucked to ensure a periodic flow to the compressor cascade. The periodicity of the flow is monitored by a row of static pressure probes in the cascade inlet plane. The side wall boundary layers can be sucked in order to vary the inlet boundary layer height, but this suction was not used in this investigation. The inlet boundary layer height has been set to 3 mm or 7.5% of the chord length.

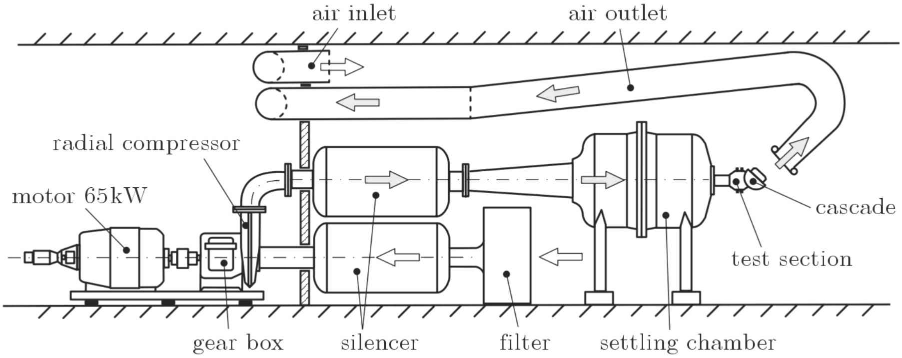

Fig. 2. Experimental high speed cascade wind tunnel at DLR BERLIN

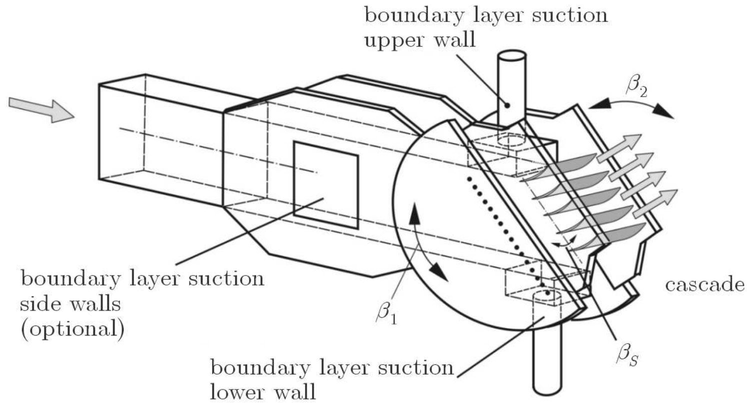

Fig. 3. Test section and cascade coordinates

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°. For flow data acquisition, a rake with 26 pitot tubes for total pressure measurement and four Conrad angle probes for outflow angle determination are used. The horizontal rake is traversed in the circumferential direction in the measurement plane 16 mm (40% chord) downstream of the stator exit. The rake Pitot tubes are made of 0.7 mm diameter tubes, their total area covers only 0.26% of the full outflow area. A previous investigation showed that there is no influence on the cascade flow measurable.

The total pressure loss coefficient is calculated for every measurement as a measure for the outflow quality. The pressure data is recorded in a multi-channel pressure transducer of high accuracy; temperatures are measured via Pt100 resistance thermometers. The systematic measurement error has been estimated according to DIN 1319 [3]. The measurement uncertainty of the total pressure loss coefficient is less than 1.2%. The repeatibility accuracy of the total pressure measurements is better than 0.2%.

To be able to compare the results of the measurements to other cascade investigations, a compressor stage efficiency has been introduced and adapted to the wind tunnel measurements. It uses a hypothetical rotor of known total pressure ratio and efficiency. With these constant data, the stator efficiency can be calculated using the total pressure in the measurement plane. Added by a term for the vacuum pump energy consumption, the calculated stator efficiency is a measure for the efficiency of the stator and the suction system. An increased efficiency corresponds to a net fuel saving.

A more detailed description of the wind tunnel and its measurement procedures can be found in the publication of Liesner and Meyer (2008).

3. Results of preliminary symmetrical investigations and reference cascade

In order to understand the working principles of the flow control, a good knowledge based on the uncontrolled cascade is essential. There have been many investigations on the NACA65-k48 cascade at this test facility (Hage *et al.*, 2007; Hergt *et al.*, 2006, 2007, 2008; Meyer *et al.*, 2007) and in other scientific institutes (e.g. Hübner, 1996; Scheugenpflug, 1989, Watzlawik, 1991). This is the primary reason why these investigations have been carried out with this kind of profile. All the measurements have been performed with equal inlet conditions of $\beta_1 = 132^\circ$, $\mathrm{M} = 0.67$ and $\mathrm{Re} = 5.6 \cdot 10^5$ based on 40 mm chord length.

The 2-dimensional plot of the total pressure loss coefficient versus the vane height and the cascade pitch is used to describe the local distribution of losses in the wake. The base flow is characterised by an area of high pressure loss above the suction surface of the vane. This is represented by the red area in the illustration. In this area, a recirculation and vortex system described in the introduction, is responsible for the defect of total pressure. It is caused by the rolling up of secondary flow in the rear part of the cascade.

This secondary flow loss is very large in the cascade examined. A state of the art cascade or axial compressor with modern 3-dimensionally shaped vanes shows a less distinct secondary flow loss. Nevertheless the problem remains the same. In the corner between the vane and side wall the flow detaches from the wall and rolls up into secondary flow vortices.

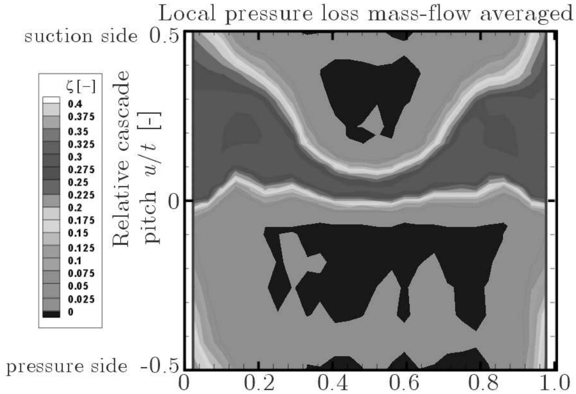

Fig. 4. Total pressure loss coefficient, reference cascade, M = 0.67, Re = $5.6 \cdot 10^5$

One possible way to eliminate this rolling up of secondary flow in a vortex and the separation is to suck the boundary layer out of the cascade. There have been studies that plane that the complete disappearance of the secondary flow is possible.

In the cascade, the flow needs to be sucked out by vacuum pump energy. This does, of course, energy consumption. In the turbo-machine, the ambient pressure is much lower than the compressor pressure. The boundary layer flow can be driven out of the machine by just opening a valve through the casing wall.

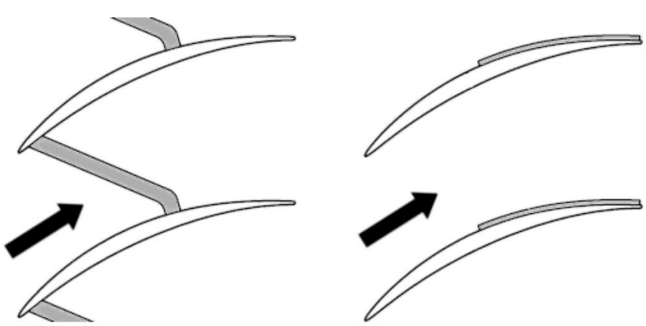

Fig. 5. Geometry of suction slots (red) in the compressor cascade walls, Type A (left) and type B (right)

In the preliminary investigation, the cascade has been influenced symmetrically. There have been orifices for boundary layer suction on both cascade side walls. The results were impressive, such as 38% of total pressure loss reduction with 5% of mass-flow was drawn off. In this case, the stage efficiency was increased by 2%, including the suction energy consumption. In another type of orifice, a total pressure loss reduction of 22% could be realised by drawing off 2% of the main mass flow only. This corresponds to a net stator efficiency increase of 1%. Keeping in mind that the secondary air system in an airplane uses compressed air for pneumatic systems and air condition, a useful implementation of the flow suction is possible.

The geometries have been derived from a literature study of the active flow control for type B, while type A geometry has been developed after a numerical study of the base flow cascade performed by the TU Berlins ISTA (Meyer and Thiele, 2010).

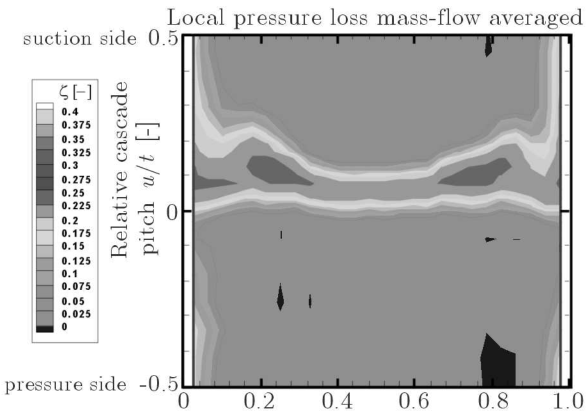

Fig. 6. Total pressure loss coefficient, symmetrical boundary layer suction, type A, M = 0.67, Re = $5.6 \cdot 10^5$, suction rate=5%

Figure 6 depicts the pressure loss distribution of the cascade with 5% suction, cascade type A. The vortex system has almost completely disappeared; the loss distribution of the high values seems almost like the wake of a flat plate. The secondary flow has been removed from the side walls in great parts.

Figure 7 shows a comparison between the base flow and the cascade with the symmetrical boundary layer suction of 5% of the main mass flow. It is obvious that in the region with high pressure loss in the base flow, there is a huge gain in pressure recovery. The separated region is very small compared to the base flow. The lower part of Fig. 7 shows the distribution of the flow deflection versus the vane height. Since there is a symmetrical distribution of the flow parameters, there is only a limited number of angle probes present.

Figure 8 shows the known downflow overturning distribution of the NACA 65 cascade according to Watzlawick. It illustrates the presence of the corner vortex and the passage vortex and their influence on the outflow angle. The corner vortex on the wall with its sense of rotation counterwise to the

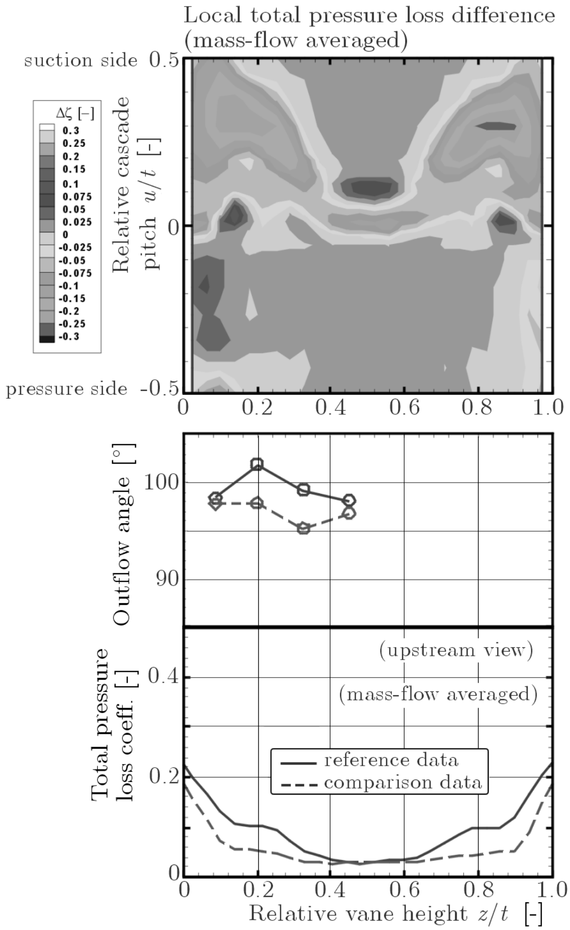

Fig. 7. Comparison of reference cascade and symmetrical suction type A

passage vortex turns the flow upwards in the area of the first angle probe. The other side of the strong passage vortex with rotation downwards makes the flow turn less in the area of the second pressure probe. As the influence of the vortex system diminishes towards mid-span of the cascade, the overturning decreases until it disappears near the centerline.

The deflection of the cascade is measured by subtraction of the exit angle β_2 from the inlet angle β_1. The lower the outflow angle, the higher the deflection.

The typical low deflection in the vortex area of the vane wake is no more to be seen. With the flow longer attached to the suction surface, it can be turned to a higher degree. This effect is mostly visible at the region between

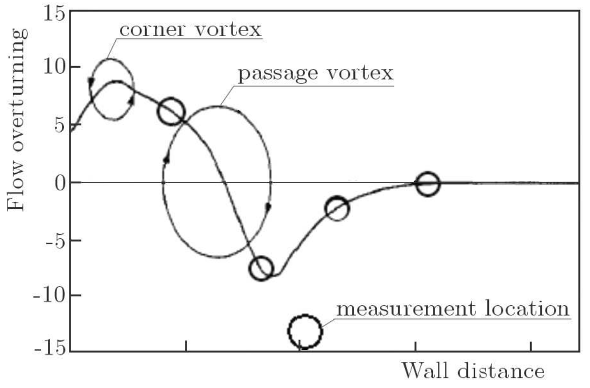

Fig. 8. Flow overturning near the wall according to Watzlawick (1991) and position of the angle probes in the channel

$z/h = 0.1$ and 0.4 as well as on the other side. It reaches to the mid-span of the passage, where it becomes small of course. The deflection is at the same time more balanced through the span and higher on the whole.

The lower part of the comparison (Fig. 7) shows the total loss coefficient pitchwise integrated. The cascade with flow control shows a wide plateau of moderate loss, from 0.1 to 0.9 of the vane height. The sidewall loss is also reduced significantly. In explicit numbers, the total pressure loss coefficient is reduced from 0.9 to 0.55, which corresponds to a reduction of 38%.

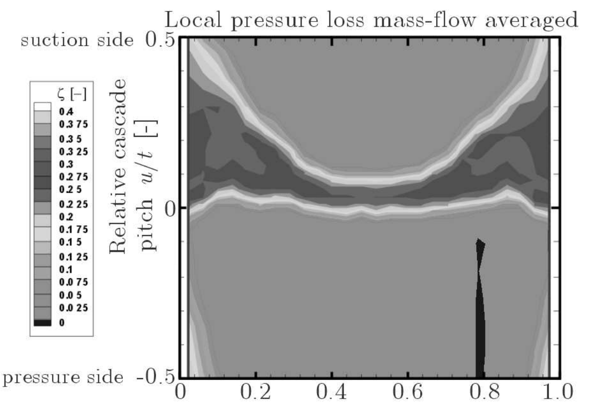

Fig. 9. Total pressure loss coefficient, symmetrical boundary layer suction, type B, $\mathrm{M} = 0.67$, $\mathrm{Re} = 5.6 \cdot 10^5$, suction rate=2%

The next picture (Fig. 9) shows the local total pressure loss distribution of the cascade with the boundary layer suction type B employed. A mass flow rate of 2% has been drawn off the cascade. Although the local distribution does not look too impressive, the comparison with the base flow brings more light into that investigation.

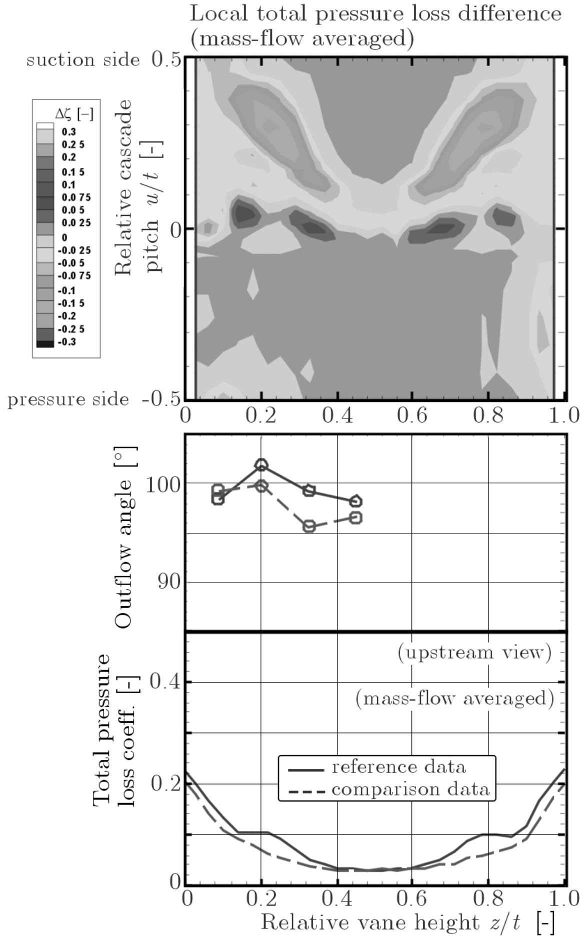

Fig. 10. Comparison of the reference cascade and symmetrical suction type B

While big parts of the wake remain unchanged, there are two large regions with great pressure loss reduction. This is the region where the vortex system joins the main flow in the central region of the passage. The vortices have been reduced in area and value.

The deflection of the passage is enhanced as well; the outflow angle distribution is smoother than in the cascade without flow control.

The integrated pressure loss versus vane height shows a drastic reduction of pressure loss in the region of the vortex main flow border. In absolute values, the total pressure loss coefficient has been decreased to 0.7, which corresponds to a reduction by 22%.

4. Results of non-symmetrical boundary layer suction

Knowing the results of the preliminary investigations, the question for possible implementation arises. The turbo-machine itself is a rotating hub inside a casing. This automatically leads to parts that are translated versus each other. A method of placing suction slots in the corners of both hub and tip is therefore virtually impossible. Possible instead would be the implementation of only one sided flow suction out of a machine. In order to figure out the effects of such a non-symmetric behaviour, the suction slots have only been realised on one side of the cascade. The suction flow rate has been bisected compared to the previous investigation, so the amount of flow drawn through the suction slots remains unchanged.

Unfortunately, the downflow angle determination is only present on the left side of the cascade. In the usual investigation, there is a proven symmetry in the passage, and it can be assumed that the channel deflection be symmetrical as well. In this setup it is not the case. Since the side with suction has been examined in the previous Section, the angle determination has been made at the side of the cascade without control. The effect is described later.

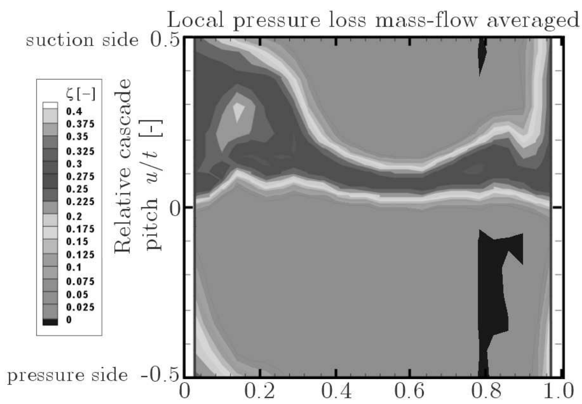

Fig. 11. Total pressure loss coefficient, non-symmetrical boundary layer suction type A, $\mathrm{M} = 0.67$, $\mathrm{Re} = 5.6 \cdot 10^5$, suction rate=2.5%

On the local total pressure loss distribution in Fig. 11, it is obvious that the suction slots have been placed on the right hand side of the cascade. The wake of the cascade is as non-symmetric as expected. On the right hand side, the secondary flow loss is little. There is only a small vortex formation in the corner between the vane and side wall. The effect is not as strong as in the symmetrical case. On the opposite side, the vortex is huge. It has been even extended a little in its dimension compared to the base flow.

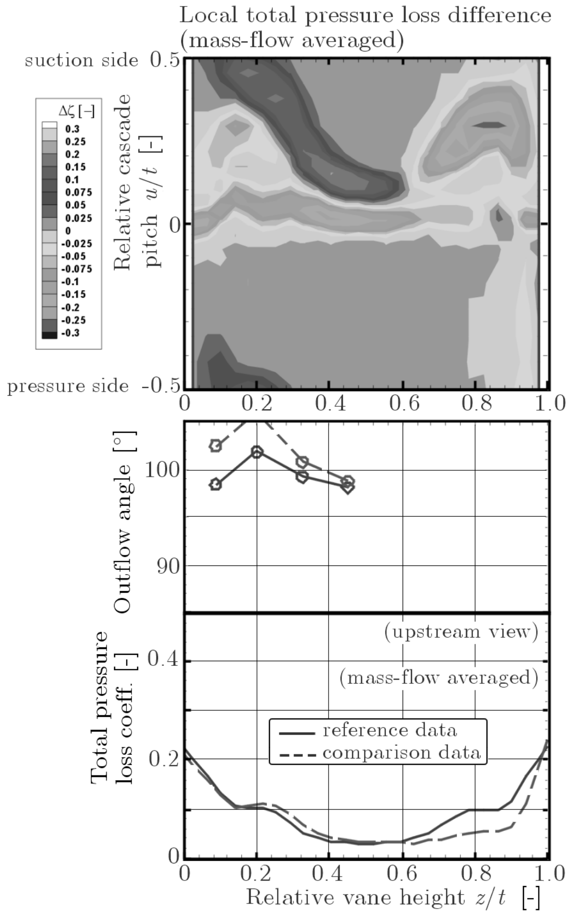

Fig. 12. Comparison of the reference cascade and non-symmetrical suction type A

In comparison with the base flow cascade (Fig. 12), the total pressure loss has still been reduced by 13%. The loss reduction is obviously taking place on the right hand side of the cascade where the suction is placed. On the left side only, a small loss increase is obvious. The big changes in the colour map are due to an offset of the high loss area from down upwards. The outflow angle of the cascade is higher than the one in the base flow cascade, which means a lower flow turning in the left part without suction. This can be explained by higher vorticity of the passage vortex, which induces higher speeds in the cascade pitch direction (see Fig. 8). In the comparison picture, this is also visible in the offset between the reference and controlled cascade.

The next investigation has been made with the cascade of type B and a suction rate of 1% of the mass flow (Fig. 13). Again, the right side has been

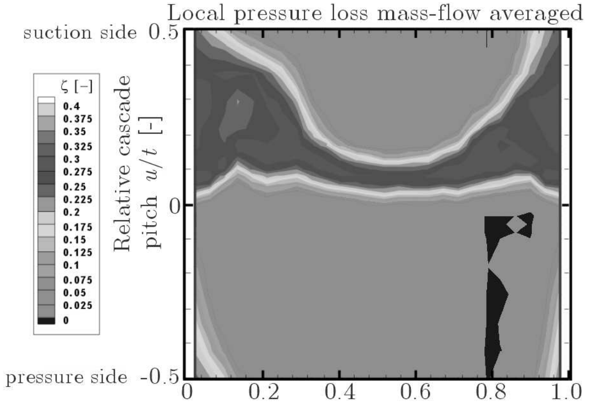

Fig. 13. Total pressure loss coefficient, non-symmetrical boundary layer suction type B, $\mathrm{M} = 0.67$, $\mathrm{Re} = 5.6 \cdot 10^5$, suction rate=1%

equipped with suction slots and the left one remains unchanged. This setup shows about the same behaviour as the previous one. The side with suction is showing less vortex structure, the region of high total pressure loss has been decreased in its dimension. On the left side, there is little change visible. The vortex on that side seems to be slightly bigger.

In the comparison layout (Fig. 14), the region where the boundary layer suction takes effect is obviously the border between the secondary vortices and the main flow. In the integral loss coefficient distribution, the step on the right side at $z/h = 0.7$ to 0.8 disappears. This is due to the fact that the passage vortex is weakened and therefore causes less total pressure loss. The pressure loss in this case is decreased by approx. 10%. The deflection of the passage side without suction is plotted, as in the previous example there is a small negative effect in the deflection obvious, this is due to the slightly bigger dimensions of the passage vortex in this case. The up-facing velocity of the vortex forces the flow to follow.

The offset of the reference cascade and the controlled one is obvious again; the lower deflection is responsible for this effect.

5. Discussion

The effect of the secondary flow in compressor cascades is known very well. The flow along the sidewall, following the pressure gradient from the pressure to suction side of the passage, causes the flow on the suction surface to separate

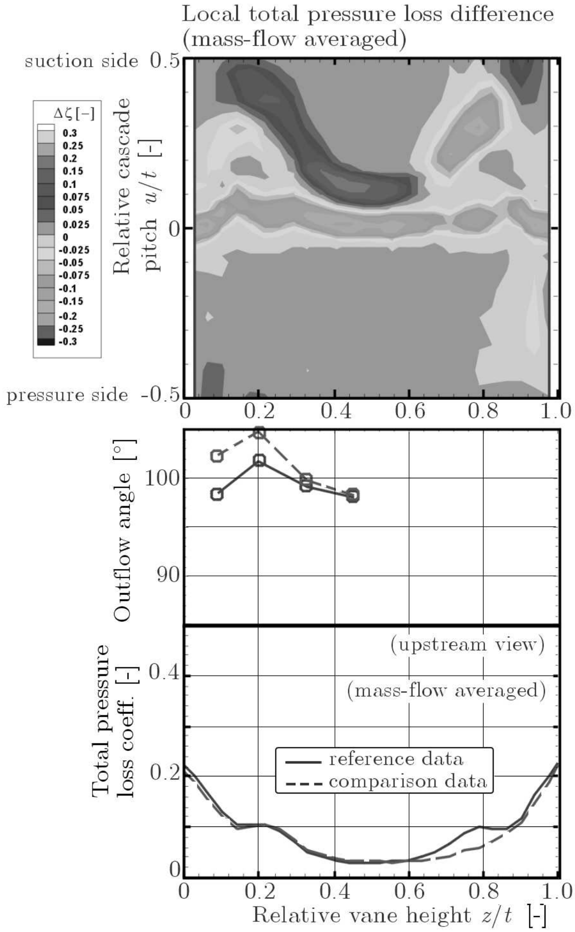

Fig. 14. Comparison of the reference cascade and non-symmetrical suction type B

from the vanes. Suppressing the secondary flow therefore, leads to a decrease of the flow detachment on the stator surfaces. Different techniques for secondary control have been employed.

The concept of boundary layer suction has been proven to be a powerful tool to get rid of the secondary flow separation in the corners between the stator vane and wall. With this investigation, the effect of non-symmetric boundary layer suction has been examined.

In a turbo-machine, however, it is impossible to have both hub and casing of a compressor equipped with fix position suction slots. It is necessary to decide whether one or the other will be implemented.

To decide how the flow behind the cascade does look like, not only pressure measurements are useful. It is interesting to see if the flow does a movement from the side without to the one with suction, following the pressure gradient. It would be preferable to obtain a two-dimensional flow behind the stator. The pressure distribution behind the cascade is not a real measure for the two-dimensionality. Laser measurements must show, whether the flow structure is sufficiently symmetric or not.

The measurements show the ability to employ one-sided suction into a compressor cascade. Their effect lies in the removal of the low speed secondary flow along the side wall. This secondary flow is likely to produce large detachment areas on the surface of stator vanes. With the secondary flow not reaching the suction surface, this detachment occurs differently. The 3-dimensional detachment on the surface is no more to be found.

In addition to the total pressure loss coefficient, there is determination of the cascade efficiency for every case measured (Bräunling, 2004). The efficiency is calculated with a reference rotor with constant pressure and temperature ratio. In the reference case, the cascade has an efficiency of 89.21%. The cascade with suction slots of type A has an increased efficiency of 89.92%, in which the vacuum pump energy consumption has already been taken into account. The efficiency of the cascade type B is 89.68%. In the annular compressor case these values can easily be improved, since the air can be drawn off the compressor without external pump energy.

Another interesting fact about this investigation is that the amount of the preferred flow suction determines the best shape of the suction slots. With a desired suction rate of only 1%, the configuration of type B is the recommended form. It has already been developed early and still did not lose its efficiency. For higher suction rates, different geometries are preferable, e.g. the one of cascade type B with a bigger cross section and different positioning on the endwalls.

6. Conclusion

The paper presents experimental measurement data on the non-symmetrical boundary layer suction in a compressor cascade with a Mach number of 0.67 and Reynolds number of $5.6 \cdot 10^5$. The cascade consisted of five NACA65-K48 vanes with a height of 40 mm and an aspect ratio of $z/c = 1$. The previously conducted investigations with boundary layer suction had shown a possible total pressure loss decrease of 38% (and 22%) using mass flow rates of 5% (and 2% respectively) in the symmetrical suction case.

In the non-symmetrical measurements, the flow through the suction orifices was kept constant, the resulting mass flow rate was 2.5% and 1%. With those settings, a total pressure loss enhancement of 13% and 10%, respectively, was reached. There is a positive effect on the efficiency of the cascade in pressure loss as well as in deflection of the flow on the side with suction orifices. The passage side without suction is slightly degraded, but there is no bigger influence than an obvious small increase of the vortex formation behind the cascade. The total pressure loss remains almost unchanged.

The efficiency of the cascade on the whole is improved; a further enhancement in an annular compressor is also expected.

Acknowledgement

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Problem zasysania w niesymetrycznej warstwie przyściennej kaskady sprężarki

Streszczenie

W pracy przeprowadzono symulację problemu zasysania czynnika w niesymetrycznej warstwie przyściennej sprężarki, analizując kaskadę z jednostronnymi wlotami ssącymi. Badania poprzedzające ujawniły wysoki potencjał w ograniczaniu strat przy zastosowaniu dwóch różnych rozwiązań sposobu zasysania w warstwie przyściennej.

Doświadczenia przeprowadzono, używając pięciu łopatek kierownicy sprężarki NACA 65-K48 projektowanej do przepływu przy prędkości 0.67 Ma i liczby Reynoldsa 560000. Geometria obydwu rozwiązań zasysania wykorzystuje projekt wąskiej szczeliny wlotowej R.E. Peacocka oraz projekt własny szczeliny szerszej, przy czym obydwa typy umiejscowiono wyłącznie po jednej stronie kanału przelotowego. Szczelinę Peacocka umieszczono w rogu pomiędzy stroną ssącą łopatki oraz ściany, natomiast szczelina szeroka pokrywa obszar od strony ssącej do sprężającej wzdłuż linii odrywania przepływu. Przy tempie zasysania stanowiącym połowę wydatku stosowanego w poprzedzających badaniach nadal odnotowano możliwość zwiększenia sprawności kaskady. Przy spadku tempa ssania do 2.5%, współczynnik całkowitych strat ciśnienia zmalał o 13%, natomiast dla dalszego spadku do 1% współczynnik ten zmniejszył się o 10%. Zgodnie z oczekiwaniami wypływ z kaskady utracił symetrię, a zasysanie jednostronne nie zmieniło znacząco charakteru przepływu na przeciwnej stronie kanału przelotowego.

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