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Eliminating the head instability of an axial-flow pump using axial grooves

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Abstract

This article presents a new type of casing treatment which completely eliminates the drop in the head characteristic of a single-stage, axial-flow pump without any noticeable negative effect on the pump efficiency. The casing treatment consists of 60 shallow axial grooves, which are located only in front of the rotor. These shallow axial grooves differ from other commonly investigated types of casing treatments with respect to their mechanism of action. For a qualitative as well as a quantitative analysis of the mechanism of action of grooves, oil flow pictures were taken and also radial velocity profiles were measured using calibrated pressure probes. These investigations reveal that, unlike other types of casing treatments, the grooves do not delay the onset of stall by controlling the rotor inflow before reaching the stall point. However, when stall occurs, the shallow axial grooves reduce the detrimental swirl component of the stall cell, which otherwise leads to the drop in pump head at the stall point. The pump efficiency at design is not affected by these types of grooves. In deep stall, the efficiency is even higher for the grooved casing due to better inlet flow conditions.

Keywords

Axial-flow pump, stall, stall margin improvement, casing treatment, axial grooves

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Introduction

Casing treatments for improving the stall margin of axial-flow compressors and pumps have been subject to intensive research for several decades. However, for a very long period of time, it was believed that a certain amount of loss in peak efficiency was inevitable for obtaining any stall margin improvement. Several investigations suggest that the more effective a type of treatment is in improving the stall margin the larger the adverse effect on the efficiency becomes.^{1–3} Due to this negative implication of casing treatment, the so-called active control has been chosen in many cases as a possible alternative for delaying stall inception. However, with respect to energy input and costs, the comparably simple method of casing treatment still seems to be an attractive concept. Meanwhile, improved experimental as well as numerical methods have led to investigations that allow for a deeper understanding of the working mechanism of casing treatment. In the literature, the most widely investigated type of casing treatment is the so-called ‘axial skewed grooves’. These grooves are located in the casing wall over the rotor blades and are reported to have two main impacts: (a) removal of fluid from the pressure side of the blade and re-injection of this fluid

in front of the rotor, thereby energizing the fluid in the casing boundary layer and (b) reduction of blade tip load due to positive swirl introduced by the skewed grooves.^{3,4} Furthermore, the axial position of the grooves relative to the rotor inlet was found to have a major impact on the peak efficiency.⁴ The configuration with the smallest overlap of axial grooves and rotor blades showed the best efficiency curves.

Every type of casing treatment mentioned in the literature has one thing in common: They try to delay the onset of stall. Hence, depending on their effectiveness, they are more or less capable of moving the stall point to smaller flow rates. However, the type of casing treatment presented here has a different purpose. In the investigated

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axial-flow pump, stall occurs at the same flow rate with and without casing treatment. Instead, this type of casing treatment – shallow axial grooves located only in front of the rotor – impacts the flow structure in deep stall such that the loss in pump head is avoided. Hence, this type of casing treatment does not only improve the stall margin, but also completely eliminates the drop in pump head. Furthermore, since the grooves are located only in front of the rotor, the grooves have no noticeable influence on the flow structure before stall inception and hence do not have a measurable impact on the peak efficiency. Moreover, the grooves have a positive impact on cavitation and vibrations – two major factors that limit the operating range of pumps.

Experimental set-up

The experimental investigations were carried out in a closed test loop at the Pfleiderer-Institut fuer Stroemungsmaschinen, Technical University Braunschweig. The investigated single-stage, axial-flow pump has an outer diameter of $D_{\text{rotor}} = 349.4 \text{ mm}$ and 6 rotor and 11 stator blades. The pump casing has an inlet diameter of $D_{\text{in}} = 340 \text{ mm}$ and an outer diameter of $D_{\text{out}} = 350 \text{ mm}$ starting from the centre point of the blade profiles. The transition area between these two diameters has a spherical shape of 350 mm in diameter. Since the rotor also has a spherical shape, the tip gap has a constant height of 0.3 mm in front of the blade centre point. Behind the blade centre point, the tip gap increases to a maximum value of about 3 mm (Figure 1). The profiles of the rotor blades are composed of circular arc mean

Figure 1. Sectional drawing of smooth casing at rotor inlet.

camber lines and NACA 65 thickness distributions. Except for the measurement of the effective vibration velocity (here $n = 1485 \text{ min}^{-1}$), the experiments were carried out at 1000 min^{-1} and the stall point of the machine was found to be at $Q_{\text{stall}}/Q_{\text{des}} = 0.75$. The applied type of casing treatment consists of 60 equally spaced axial grooves (depth 4 mm and width 10 mm) which start about 200 mm in front of the rotor and end right at the leading edge of the outer blade profiles. Hence, in this machine, there is no overlap of grooves and rotor blades. Figure 1 shows a sectional drawing of the smooth casing at the rotor inlet to depict the casing geometry. Figure 2 is a drawing of the pump in the grooved casing and Figure 3 a view into the open grooved casing.

A variety of experimental methods were applied comparing the smooth and grooved casings. Due to limited space, however, this article will focus only on the results of the oil flow pictures and radially traversed pressure probes. More complete information on the entire set of experiments can be found in Kosyna et al.⁵ and Goltz.⁶

For optical access, the pump casings (both smooth and grooved) were equipped with plexiglas windows of 400 mm axial length (also smooth and grooved), which allowed roughly a 90° sector (corresponding

Figure 3. Photograph of grooved pump casing when test loop was opened.

Figure 2. Sectional drawing of grooved casing and cross section of grooved wall.

to about one-and-a-half rotor blade passages) to be observed.

These plexiglas windows were used for taking the oil flow pictures of the casing wall, the hub wall, and the rotor and stator blades at flow rates both before stall onset as well as in deep stall. Due to the similar density of water and the chosen paint, it was physically justifiable to take the oil flow pictures at the rotor blades, also, despite the slightly different centrifugal forces on water and paint. Oil flow pictures visualize the surface pattern of the shear stress lines, which closely approximate the stream lines of the flow next

Figure 4. Scaled drawing of pump and measuring planes for radially traversed pressure probes and schematic drawing of circumferentially averaged main flow structures in deep stall.

to the wall. They represent time mean portraits of the absolute casing wall flow and the relative blade surface flow. Following the criteria described in Tobak and Peake,⁷ important features of the flow field, such as separation and attachment lines, can be identified from the oil flow pictures.

To get more quantitative information on the flow structure, pressure probes were traversed radially upstream and downstream of the rotor both with and without grooves. Figure 4 shows the axial positions of the measuring planes. First, a calibrated three-hole probe was used to take circumferentially averaged measurements. A calibrated one-hole probe with an integrated miniature pressure sensor was applied for resolving the flow components circumferentially. The data were taken at 20 spanwise positions. The time-step between two measurements was chosen in a way that 450 circumferential positions were resolved, leading to a circumferential resolution of 0.8° . The mean values over 145 rotor revolutions were taken for each position. Hence, only periodic flow structures could be monitored. A detailed description of the measuring techniques that were used may be found in Goltz.⁶

Results and discussion

The pump with the smooth casing was found to have a clear stall point at $Q_{\text{stall}}/Q_{\text{des}} = 0.75$, followed by a transient descent into stall (Figure 5). With the

Figure 5. Pump head and efficiency characteristic without and with grooves.

grooved casing, however, the decrease in pump head can be avoided completely. As the experimental results confirm, the efficiency in stable operation is unaffected by the grooves. Furthermore, the NPSH3 curve and the effective vibration velocity show a much more advantageous characteristic (Figures 6 and 7).

How the grooves work and why the efficiency is not affected by these types of grooves can be understood by analysing the flow structures without grooves at stable operation and in deep stall. Two flow models – one for the flow in the stable operating range (Figure 8(a)) and one for the flow in deep stall (Figure 8(b)) – were developed by summarizing the findings from the oil flow pictures and the pressure probes. The authors are convinced that these two flow models also hold for other axial-flow as well as mixed-flow machines. Even axial-flow compressors have shown the same main flow structures.⁸

In Figure 8, the main flow enters from the left-hand side. The pressure-driven backflow through the tip clearance collides with the incoming casing boundary layer and the two flow structures separate from the wall. In the entire stable operating range, the

separation line is formed at nearly the same axial position, that is at the rotor leading edge (Figure 8(a)). At the blade suction side, the two layers coil up into the tip leakage vortex, which is tight and straight and which leaves the blade passage towards the stator. It is also important to note that in the stable operating range, the backflow region is extremely thin, covering only the tip gap region. At the stall point, however, the balance between incoming boundary layer and tip gap backflow is destabilized. Right at the stall point, the separation line ‘jumps’ upstream and a new balance is re-established. In deep stall, the separation line is located far ahead of the rotor (see also Goltz et al.⁹). There is no tip leakage vortex any longer. The vortex that develops instead from the incoming flow and the backflow has increased dramatically in the radial direction and covers approximately one-third of the blade span (see Vortex B, Figure 4). Its axis extends mainly in the circumferential direction. The backflow through the rotor gap has a huge absolute circumferential component which is sustained when the flow is returned at the separation line. Hence, the portion of the vortex re-entering the

Figure 6. NPSH3 curve without and with grooves.

Figure 7. Effective vibration velocity measured at the casing without and with grooves at $n = 1485 \text{ min}^{-1}$ and $\text{NPSHA} = \text{NPSH3}_{des}$.

rotor has a very high positive swirl component which is also transmitted to the main flow through friction. According to the Euler turbine equation, it is very obvious that this positive pre-swirl leads to a decrease in the pump head.

The same considerations taken for the grooved casing lead to the result that in the stable operating range, the grooves are simply passed through in the

positive axial direction. Hence, the grooves only contribute to an increased casing surface area and, therefore, to slightly higher friction losses. In practice, this influence was not noticeable. When the stall point is reached, the separation line 'jumps' into the grooved section. As a result, the grooves are now passed in the negative flow direction. The analysis of the oil flow picture at the grooved casing (Figure 9) leads to the

Figure 8. Flow models of flow at the stable operating range (a) and in deep stall (b).

Figure 9. Oil flow picture at grooved casing in deep stall (a) and schematic sketch of flow structure (b).

assumption that there is (at least) one vortex formation in each groove. The backflow hits the grooves and a vortex roll-up picture at the grooved casing (Figure 9) leads to the assumption that there is (at least) one vortex formation in each groove. The backflow hits the grooves and a vortex rolls up between the two separation lines. The swirl component of the backflow is reduced by the impact of hitting the grooves and due to the friction of the 60 vortices inside the 60 grooves. Therefore, a smaller amount of swirl is transmitted to the main flow.

Figure 10 shows the axial and circumferential components of the time mean absolute velocities in front of the rotor, which were measured with the mentioned three-hole probe. From these measurements it becomes clear that there is no swirl component (c_u) in front of the rotor at the design point. However, when there is backflow in deep stall (Vortex B with huge swirl component has formed), a huge portion of the flow entering the machine has a very high positive swirl component. From Figure 10, it becomes clear that the grooves do not prevent the formation of Vortex B but drastically decrease the swirl component inside Vortex B as well as in the main flow.

In order to show the effect of the grooves in the circumferential direction, the circumferentially resolved velocity distributions are also presented. Corresponding to Figure 10, the relative flow rate of $Q/Q_{des}=0.5$ was chosen exemplarily for the flow in deep stall (Figures 11 to 13). However, altering the flow rate revealed that even within deep stall, the flow structure changes noticeably from one flow rate to another. More detailed information on this aspect may be taken from Goltz.⁶

Figure 11 shows the absolute velocity components without ((a) and (c)) and with ((b) and (d)) grooves in front of the rotor, whereas Figure 12 is set up in the same manner for the axial position behind the rotor. In Figure 11(a) and (b), it can be seen that the axial backflow region with and without grooves is very similar. For the grooved casing, the main flow (high positive axial components) moves to somewhat higher radii. The major difference between the smooth and grooved casings, however, appears in the plots of the circumferential components. The circumferential components can be reduced from $c_{u/max}=15$ m/s for the smooth casing to $c_{u/max}=6$ m/s for the grooved casing. Hence, the grooves do not prevent the

Figure 10. Axial and circumferential components of the time mean absolute velocities in front of the rotor.

formation of Vortex B (Figure 4), but they strongly decrease the swirl in this vortex.

Figure 12(a) and (b) reveals that Vortex A behind the rotor becomes more dominant for the grooved casing. Due to this enlarged Vortex A, the main flow is displaced towards higher radii. The circumferential velocity components behind the rotor are quite similar however (Figure 12(c) and (d)).

Looking at the magnitude of the relative velocities in Figure 13(a) and (b), one has to keep in mind that there is backflow at radii higher than $r = 150$ mm. For smaller radii, however, it becomes clear that the rotor blades in the grooved casing are facing higher relative velocities entering the rotor, whereas the relative velocities behind the rotor are considerably smaller with the grooved casing. Therefore, the fluid undergoes a much stronger deceleration in the grooved casing.

For finding an answer to the question as to why in deep stall the pump with grooved casing has an about 30–50% higher pump head than the pump without grooves, several aspects have been considered.

1. Due to the reduced positive pre-swirl, the Euler head obviously becomes higher with the grooved casing.
2. Due to an improved inflow relative to the rotor blades, the fluid undergoes a much stronger deceleration in the grooved casing.
3. Due to the reduced swirl in Vortex B, friction losses can be avoided in the grooved casing.

In order to get an estimate of the friction losses inside Vortices A and B, the dissipated energy inside the two vortices was calculated. Based on the

Figure 11. Absolute velocity components in front of the rotor at $Q/Q_{des} = 0.5$. Absolute axial velocity component: (a) without grooves and (b) with grooves. Absolute circumferential velocity component: (c) without grooves and (d) with grooves.

Figure 12. Absolute velocity components behind the rotor at $Q/Q_{des} = 0.5$. Absolute axial velocity component: (a) without grooves and (b) with grooves. Absolute circumferential velocity component: (c) without grooves and (d) with grooves.

experimental data, the dissipated energy was computed implementing a finite difference method (central) in Matlab. However, the estimate led to similar values for the smooth as well as the grooved casings. The values did not have the same order of magnitude as the difference in pump head between grooved and smooth casings in deep stall. Therefore, it can be stated that the increase in pump head is not due to avoiding friction losses but mainly due to the better inflow conditions as a result of the grooved casing.

Conclusions

A new type of casing treatment was tested: shallow axial grooves which are located exclusively in front of the rotor.

1. These axial grooves have a mechanism of action which differs significantly from former types of

casing treatments. The axial grooves do not move the stall point to smaller mass flow rates, i.e. they do not delay the formation of Vortex B. On the contrary, the axial grooves allow for the formation of Vortex B but reduce the detrimental swirl inside the vortex.

2. Since the grooves are located only in front of the rotor and the separation line 'jumps' only when reaching the stall point, the grooves do not affect the flow before stall onset.
3. Contrary to previous experience published in the literature, this type of casing treatment is very effective without any negative effect on the efficiency.
4. The mechanism of action of the grooves is not based on avoiding friction losses, but the grooves improve the inflow into the rotor channels, which results in a better deceleration through the blade passage.

Figure 13. Magnitude of relative velocity at $Q/Q_{des} = 0.5$. In front of the rotor: (a) without grooves and (b) with grooves. Behind the rotor: (c) without grooves and (d) with grooves.

A similar effect is reported by Perez et al.¹⁰ in taking measurements at the same test rig with a double-inlet nozzle in front of the rotor instead of the casing grooves. The double-inlet nozzle separates the spinning backflow and the non-spinning main flow and, hence, has a similarly positive effect on pump head and efficiency.

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Appendix

Notation

b	width of grooves
c	absolute velocity

D	diameter
H	pump head
l	length of grooves
NPSH3	net positive suction head with a decrease in pump head of 3%
NPSHA	available net positive suction head
Q	flow rate
r_A	outer radius of Vortex A
r_B	inner radius of Vortex B
rx, ry	radius
t	depth of grooves
v_{eff}	effective vibration velocity
$\bar{w}$	magnitude of relative velocity
η	efficiency

Indices

ax	axial
des	design point
in	at inlet
out	at outlet
stall	stall point
u	circumferential