Influence of the rotor blade aspect ratio on the performance of an axial fan

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Abstract. An axial fan with four different number of blades for the rotor is investigated. In this contribution varying the blade aspect ratio means the number of blades is changed while every other dimensionless design parameters and the design point are kept constant. By varying the blade aspect ratio, the length of the blades alters and as a result the Reynolds number which is related to the chord length of each blade is changed as well. The considered Reynolds numbers are lying between $2.38 \times 10^5$ and $7.13 \times 10^5$. For the stator the blade aspect ratio as well as the spacing ratio are the same during the investigation.

For comparison and validation both numerical and experimental investigations are performed. For the numerical investigations, due to the low Reynolds numbers the $k$-$\omega$-SST transition model with the intermittency function is applied. Transient simulations are carried out. The experimental investigations are performed a pipe test bench. The results are compared in terms of performance differences for part load, overload and design conditions. The flow structure in the rotor passages is visualized and compared using the numerical results to show the influence of the blade aspect ratio on the performance.

1. Introduction
Improving and designing axial fans with good aerodynamic results is still an essential area of research. It is important to operate fans efficiently not only in the design point, but also in the part and overload range. In literature, there are only a few investigations on the influence of the blade aspect ratio on the performance of axial fans. Mostly, compressors or compressor cascades are explored. Even if the fluid is assumed as incompressible there exist some differences in designing a compressor or a fan. Here an axial fan is designed and investigated. The aim of this contribution is to develop a first approach for the selection of the blade aspect ratio by means of the efficiency and the pressure rise coefficient.

Changing the blade aspect ratio $\Lambda$, while keeping every other dimensionless design parameter constant, means changing the number of blades. In literature, it is not always clear which parameter is varied in case of changing the number of blades. This is the case for the approach by Jorgensen of 1970 [1]. In his book he presents a formula for the calculation of the number of blades in dependence of the diameter ratio $\mu$:

$$n_{\text{blade}} = \frac{6\mu}{1 - \mu}.$$  \hspace{1cm} (1)

Any information about the choice of other design parameters or a validation of the formula are missing. Using equation 1 the number of blades lies between 3 and 14 in dependence of $\mu$. It is
the only approach that can be found in literature using an equation for the calculation of the number of blades. All other investigations try to give an experimental or a numerical approach. The length of the blades is changed while changing the blade aspect ratio and thereby the Reynolds number. In 1954 Scholz was the first who assumed that lower Reynolds numbers affect the secondary losses in a positive way and that the profile losses increase \[2\]. Regarding the performance, Horlock et al. came in 1964 to the result that with decreasing \(\Lambda\) stall occurs at lower flow coefficients \[3\]. They did their experimental investigations using a compressor cascade with air, which was assumed as incompressible. In their investigations, they took into account that the Reynolds number differs for different aspect ratios. Horlock et al. detected a critical Reynolds number in dependence of the aspect ratio. Losses are increasing rapidly below that aspect ratio. The results were confirmed by Shaalan who investigated a compressor cascade as well \[4\]. He assumed the fluid to be incompressible. The influence of the Reynolds number is still treated in investigations until today. In the last years Schmidt et al. \[5\] and Peters et al. \[6\] approved this statement investigating a compressor cascade. Both confirmed that with lower Reynolds numbers and thus higher aspect ratios the profile losses are increasing. They stated a decrease of the efficiency using higher aspect ratios. Matai and Yavuzkurt \[7\] and To and Miller \[8\] verified that earlier stall occurs with higher aspect ratios in their investigations using compressors as well. To et al. found out that up to a blade aspect ratio of 0.7 the secondary losses and the profile losses can be split up for further examinations. If the blade aspect ratio is less the secondary losses of the hub and the shroud meet at the midspan so that the profile losses can no longer be separated from the secondary losses anymore. In accordance to Smith \[9\], Fahmi \[10\] and Britsch \[11\] To and Miller \[8\] have found a relation of the tip clearance to chord ratio, which changes because of a constant tip clearance. They have stated that the maximum static pressure rise depends on the ratio of the tip clearance to chord. To and Miller have confirmed that the maximum static pressure rise is independent of the aspect ratio. Overall in industries the trend is going to higher blade aspect ratios. This was mentioned by Wennerström in 1989 \[12\] and has been confirmed by Broichhausen and Ziegler in 2005 \[13\].

Most of the investigations which can be found in literature are treating compressors. A detailed analysis of axial fans is still missing. This contribution tries to give a first approach of the flow behaviour in axial fans.

2. Design process

The design point is specified through the flow coefficient \(\phi\), the pressure rise coefficient \(\psi\) and the rotational speed \(n\). The diameter at the shroud \(D_{\text{shroud}}\) and the diameter ratio \(\mu\) are chosen. The spacing ratio \(t/l\) and the thickness ratio \(d/l\) at the hub and at the tip of the blades are predefined with a linear distribution from hub to shroud. With these values the velocity triangles as shown in figure 1a are calculated. Index 1 is in front of the rotor, index 2 between the rotor and the stator and index 3 is behind the stator. The thickness distribution of the profiles is prescribed using bisuper ellipses \[14\]. The stagger angle \(\lambda\) is calculated using the Martensen Method \[15\]. It is varied until the wanted deflection of the flow \(\Delta\beta\) is achieved. Additional information of the design method can be found in \[16\]. An overview of the design values is given in table 1a.

In this paper, different aspect ratios of an axial fan are examined. This is done by varying the number of blades. Fans with 3, 5, 7 and 9 rotor blades are evaluated. The above mentioned design parameters \(\phi\), \(\psi\), \(n\), \(d/l\) and \(t/l\) are held constant during the investigation. Only the aspect ratio \(\Lambda\) is modified. This means, the chord length \(l\) of the blades is varied. The Reynolds number changes because of the various chord lengths, too. \(\Lambda\) is varied between 0.205 (3 blades) and 0.616 (9 blades). The number of blades of the stator (11 blades) is kept constant in this investigation. In table 1b the data of the different fans is summarized.
3. Experimental setup
A single-staged axial fan is investigated experimentally using a pipe test bench. The test bench consists of an upstream pipe of 1.35 m length in front of the rotor. A 1.4 m outlet section is used behind the stator. The inner pipe diameter is 0.25 m. A tip clearance of 0.5 mm is set at the rotor. The tip clearance is held constant during the whole evaluation. To determine the mass flow through the fan a jet nozzle in combination with a static pressure transmitter is used at the inlet. The static pressure is measured 0.6 m in front of the rotor and 0.12 m behind the stator. The location behind the stator was chosen to avoid disturbances of the flow, due to the mounting of the fan. The calculation of the mass flow and all pressure measurements are in accordance to ISO 5801. For the variation of the mass flow a throttle is built in at the outlet. The rotor is operated at 3000 min\(^{-1}\) with a synchronous motor. The aerodynamic power is determined using a torque-meter. To measure the bearing friction the torque is measured in idle run without a rotor. To determine the aerodynamic power \(P_{\text{aero}}\) the resulting friction is subtracted from the measured torque.

4. Numerical model
The simulations are carried out with ANSYS CFX 17.2. For the simulation a 3D model of the test bench is used, which is shown in figure 2a. In figure 2b, the different fan versions that are examined are shown. The meshes consist of block structured grids. For the meshing of the rotor and the stator an o-grid is applied for a better resolution of the boundary layer. A mesh study was performed and the best mesh size was emphasized with a total number of nodes of approximately \(1 \times 10^7\). The number of nodes, distributed along the chord length of the rotor vanes, are scaled with the chord length for different \(\Lambda\), so that all meshes have the same size. The fluid is air at 25 °C, which is assumed as incompressible. At the inlet a mass flow and at the outlet a constant pressure boundary of 0 Pa is set. The mass flow in the simulation is
varied between 60% and 120% of the design mass flow. The SST transition model with the intermittency function is used as turbulence model. High Resolution and Second Order Backward Euler are applied as numerical schemes. Due to the turbulence model, the boundary layer has to be resolved with $y^+$ of approximately 1. The mean $y^+$ is 0.26 and the maximum 1.97, which is in accordance to the turbulence model. The numerical model consists of an inlet domain, a rotor domain, a stator domain and an outlet domain. For the transient simulations at the interface between the rotor and the inlet, and the rotor and the stator transient rotor stator interfaces are used. All other interfaces are GGI interfaces. After a transient response the transient simulations are carried out with a time step of $5.56 \times 10^{-5}$ s. Four full revolutions with a 1° resolution – obtained with the before mentioned time step – are calculated until convergence. For the calculation of the performance curves the arithmetic averaged values over one blade passage are utilized.

The evaluation of the static pressure is done on the planes in front of the rotor and behind the stator, as shown in figure 2a. For comparison, the locations of the planes in the simulation are the same as in the experiment. For the evaluation of the performance the flow coefficient $\phi$, the work coefficient $\psi$ and the aerodynamic efficiency $\eta_{aero}$:

$$\phi = \frac{c_m}{u_m}, \quad \psi = \frac{\Delta p_{tot}}{\frac{1}{2} u_m^2}, \quad \eta_{aero} = \frac{P_{aero}}{P_{mech}} = \frac{\phi u_m D^2 (1 - \mu^2) \Delta p_{tot}}{8 n M}$$

are calculated. The calculation is done using the meridian velocity $c_m$ and the circumferential velocity $u_m$ at the midspan. Additionally the pressure coefficient $c_p$ is used:

$$c_p = \frac{p_x - P_{stat,1}}{\frac{1}{2} w_1^2}.$$
maximum pressure rise, for all investigated Λ stall occurs earlier in the simulation than in the measurement. One possibility for the occurring deviations might arise out of some uncertainties in manufacturing. Differing tolerances might be leading to different tip clearance sizes. Another possibility is that vortices are over- or underestimated in the simulation due to the turbulence modelling. With decreasing aspect ratio these influences might be growing.

In figure 4, the characteristic curves of the experiment and the simulation are compared for all Λ. The pressure rise coefficient $\psi$ is plotted in figure 4a. Different aspect ratios have only a small influence on the maximum flow coefficient that can be achieved in the experiment. In the design point ($\phi = 0.5$) and in overload conditions only small differences regarding the aspect ratios can be observed. In the numerical results the differences for the pressure rise coefficient are growing with increasing flow coefficient for different Λ. That means the slopes of the pressure rise curves are changing for different aspect ratios. The steepest slope can be observed for Λ06 and the lowest slope for Λ02. Regarding the experiment, the observed trends of the results in terms of slope are the same, but not as clear as in the simulation. In the experiment, Λ02 achieves the highest pressure rise, whereas in the simulation it is Λ03. Comparing the aspect ratio with the lowest pressure rise coefficient at stall point in both the simulation and the experiment it is Λ06. Higher maximum pressure rise coefficients arise in the experiment. The higher Λ the faster stall happens after achieving the maximum pressure rise. This can be observed in the experiment as well as in the simulation. In terms of $\phi$ the order of stall occurrence in the simulation and the experiment is the same.

Looking at the aerodynamic efficiency in figure 4b, the points of maximum efficiency in the experiment are shifted to lower flow coefficients compared to the simulation. For all Λ the maximum efficiency occurs at nearly the same flow coefficient in the measurement. In the simulation, the order of the maximum efficiency cannot be clearly defined due to the limited numbers of simulated points. For Λ06 in overload condition the efficiency drops faster than for the other Λ. In the simulation the best efficiency is obtained for Λ04. The order for decreasing
efficiency is Λ₀₃, Λ₀₆ and the lowest efficiency is achieved with Λ₀₂. For the experiment the order is changing. The best efficiency is obtained with Λ₀₆. The remaining order is with Λ₀₄, Λ₀₃ and Λ₀₂ the same as in the simulation. These differences between the experiment and the simulation might be because of the weightless rotors in the numerical investigations. Due to the different length of the rotor blades, the extension of the rotor is varying for the different Λ. As a result of the different weights, various forces act on the bearings because of the different tilting of the shaft. These tilting forces are higher for fans with lower Λ than for those with higher Λ. In overload condition, the differences in efficiency between different Λ are higher in the simulation than in the experiment. Overall, the aerodynamic efficiency in the measurement in the design point and in overload condition is lower than the prediction of the numerical results. Only in part load when stall occurs in the simulation, but not in the experiment the efficiency in the experiment is higher than in the simulation.

5.2. Flow structure
In figure 5 the pressure coefficient \( c_P \) at the midspan along the chord length of the blade is shown for the design point. The blade coordinate starts with 0 at the leading edge (LE) and ends with 1 at the trailing edge (TE). The upper curves correspond to the pressure side and the lower ones belong to the suction side. All evaluated Λ show a similar progression on the pressure side. They differ only slightly. Blades with lower Λ have a higher pressure coefficient all over the blade than blades with higher Λ. As a result, Λ₀₂ has the highest and Λ₀₆ has the lowest pressure coefficient distribution.

On the suction side, all investigated Λ reach roughly the same pressure coefficient until two third of the chord length. A zero pressure gradient occurs at two third of the chord length. This is typical when it comes to transition from laminar to turbulent [17]. At this point the flow starts to separate, because of an adverse pressure gradient and a laminar separation bubble is formed. The bubble reattaches when the pressure coefficient rises steeply. The phenomenon is differently distinct in its appearance. Most clearly, it can be seen for Λ₀₃ and Λ₀₆. The location and the extension of the transition region is, in proportional terms, nearly the same for all Λ except for Λ₀₂. For the latter, the transition region is smaller than for the other cases.

In figure 6, the rothalpy is shown as a contour plot on a plane that is situated at 40% chord length. The figure depicts for all Λ the variable \( I \) to visualize the losses on that plane. In the first row the rothalpy in the design point is shown. At the shroud the tip vortex can be seen for all aspect ratios. The vortex is marked for Λ₀₂ in the figure with a red arrow. The circumferential extension of the tip vortex differs slightly regarding the different aspect ratios. The extension in case of the smallest aspect ratio Λ₀₂ is barely half of the passage. The higher Λ the larger is the circumferential extension of the tip vortex related to the spacing. For Λ₀₆ the tip vortex extends slightly beyond half of the passage spread. A similar behaviour can be recognized for the lower flow coefficient. In the second row of figure 6, the flow coefficient with the maximum pressure rise, with regard to the simulation, is shown for each Λ. This flow coefficient (\( \phi = 0.35 \)) is the same for all Λ except for Λ₀₆. Here the maximum pressure rise occurs at a higher flow coefficient:
\( \phi = 0.4 \). For \( \Lambda_02 \) the tip vortex reaches slightly over half of the passage spread, whereas for all other \( \Lambda \) the tip vortex impacts the next blade. Concerning the tip vortex, it has to be kept in mind, that for all fans a constant tip clearance is used, so that influences might arise because of a change in \( s/l \) and not on account of \( \Lambda \). In the plane a second vortex – the right part of the horseshoe vortex – can be noticed in all passages for both flow coefficients. This vortex is marked with a blue arrow for \( \Lambda_02 \). With two third circumferential passage extension in the direction of \( \omega \) the percentage position of this vortex is the same for all aspect ratios. Only the shape and the radial extent differs for the two considered flow coefficients.

6. Conclusion

6.1. Summary

In this study a comparison of a fan with four different blade aspect ratios \( \Lambda \) was performed. The performance curves obtained from the numerical simulation have been validated with experimental results. It has been shown that in the design point and in overload conditions both curves correspond to each other. In heavy part load, the numerical results overestimate the stall point of the fan for every examined \( \Lambda \). As a result, the maximum pressure rise occurs at higher flow coefficients in the simulation than in the experiment. For different aspect ratios the maximum pressure rise arises at different flow coefficients. The order – regarding the flow coefficient – of the investigated fans when stall occurs is the same in the simulation as in the experiment. Analysing the pressure coefficient, it can be seen that laminar separation for all \( \Lambda \) exists. For \( \Lambda_02 \) the transition region is smaller compared to the other aspect ratios, so that a more detailed view into the flow structure on the blade is required. Transition takes part at the same dimensionless location for all aspect ratios. The comparison of the rothalpy on a plane at 40 % chord length shows the tip vortex and its different circumferential extensions for different aspect ratios. Near the hub a part of the horseshoe vortex can be seen.

6.2. Outlook

In this paper the losses were only examined to a limited extent. In future studies the loss coefficient and the loss mechanisms have to be studied. In addition to the comparison of the performance curves, the flow structure inside the rotor has to be validated by experiment. Transition regions and their differences regarding different aspect ratios should be studied on the blades. Furthermore, a detailed view on the stall point is needed.

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Nomenclature

| Latin Symbols | Greek Symbols |
|---------------|--------------|
| $c_m$ | $\Lambda = h/l$ | blade aspect ratio |
| $c_P$ | $\lambda$ | stagger angle |
| $d/l$ | $\mu = D_{hub}/D_{shroud}$ | diameter ratio |
| $l$ | $\nu$ | kinematic viscosity |
| $l$ | $\rho$ | density |
| $M$ | $\phi$ | flow coefficient |
| $n$ | $\psi$ | pressure rise coefficient |
| $n_{blade}$ | $\xi$ | blade coordinate |
| $p$ | $\sigma$ | mechanical |
| $P$ | $s$ | subscripts |
| $t/l$ | $s$ | in front of the rotor |
| $t$ | $t$ | behind the rotor |
| $t/l$ | $t$ | mechanical |
| $u_m$ | $u$ | circumferential velocity at midspan |
| $w$ | $w$ | relative velocity |

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