REMARKS ON THE MERIDIONAL DESIGN OF MIXED FLOW FANS

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ABSTRACT

Due to a large range of application areas for fans in the industrial, the household and the agricultural sector, there are already many fans of different types, in different shapes and sizes available at the market. If the design point requires large flow rates with a sufficiently large increase in pressure, the recommended fan type is between the axial and radial fan type, within the range of mixed flow fans. But there are only very few recommendations for mixed flow fans design methods available in the literature.

In order to prevent a shroud separation in the mixed flow fan, 3 different radii (R80, R130 and R180) were investigated. An increase in pressure and efficiency of the mixed flow fan with the radius R130 was determined by the measurement results. The influence on the characteristic curves of different downstream angles Chi (χ) between χ = 30° (almost axial) and χ = 90° (radial) was investigated in terms of pressure rise, flowrates and efficiencies. The mixed flow fan with a downstream angle of χ = 68° has increased flow rates with better efficiencies compared to the investigated centrifugal fan. The mixed flow fan type already achieves higher pressure rises in a single stage than axial fans and can be easily integrated in fluid systems with axially parallel inlet and outlet flow. They represent a serious alternative in the application field between axial and radial fans.

KEYWORDS

LOW PRESSURE FAN, MIXED FLOW FAN, MERIDIONAL DESIGN, PERFORMANCE CURVES, MOTOR COOLING FAN

NOMENCLATURE

β_1	[°]	inlet blade angle
β_2	[°]	outlet blade angle
b.2	[m]	outlet blade width
D.1a	[m]	outside diameter of the blade leading edge
D.2a	[m]	outside diameter of the blade trailing edge
D.2i	[m]	inside diameter of the blade trailing edge
D.n	[m]	hub diameter
D.s	[m]	outside diameter of the suction side
Δp_0	[Pa]	pressure difference of the measuring orifice
Δp_1	[Pa]	pressure increase of the fan
H	[%]	hygrometer
p_a	[Pa]	ambient pressure
R	[mm]	radius
T	[T]	temperature
Tm	[Nm]	torque
U	[rpm]	rotational speed
χ	[°]	downstream angle

INTRODUCTION

Low pressure fans are defined as fluid flow machines that convert mechanical energy into flow energy at a pressure ratio up to 1.1 (Pfleiderer and Petermann, 2005). The fluid pressure increase from the fan inlet to outlet amounts to 10000 Pa. Fluid Flow Machines for larger pressure increases are named blowers and compressors.

The direction of the airflow through the impeller from inlet to outlet classifies fans into axial, centrifugal and mixed flow fan types. For large flow rates axial fans are a good choice, centrifugal fans on the other hand are suitable for large pressure increases. At enhanced demands for pressure increase axial fans often need require two or more impeller stages. That causes higher costs, and additional space is also required. A suitable alternative would be a fan type that provides high flowrates, like the axial fan, with relatively large pressure increases. Therefore in mixed flow fans, the pressure rise due to centrifugal forces, is combined with the axial fan type.

A special field of application for fans is the cooling of motors and electrical components. Motor cooling fans are installed directly on the motor shaft inside the housing. They ensure the removal of the heat, yielded by engine components, away and into the passing air flow. Major demands for the design point of these fans are large flow rates with the associated high flow velocities for sufficient cooling capacity. The airflow through the engines, especially through the narrow gap between the stator and rotor, as well as the numerous flow deflections causes high pressure losses. Thus, the requirements for the cooling fans are high flow rates for cooling with comparatively high pressure increases caused by the limited installation space.

The exhaustive literature research of fans, particularly with regard to the type of mixed flow fans, revealed that only very few publications and almost no recommendations are available for the design methodology of mixed flow fans. The three-dimensional, viscous, turbulent and unsteady flow conditions in mixed flow fans are difficult to compute. The complex double-curved blade geometry was previously more complicated and also more expensive to manufacture than, for example, centrifugal blades of constant thickness. This is probably one reason, why only a comparatively small range of products is available for mixed flow fans on the market. With the very rapid development of 3D printing processes, a meridional contour of a mixed flow fan is easy to manufacture.

Classical design methods of flow machines refer to axial and radial fans, whose effects and characteristics can clearly be assigned to an axial or radial flow guide. In 1952 Wu published a quasi-three-dimensional design method for mixed flow fans (Wu, 1952). It was proposed to obtain steady flow relative to the blades by an iterative solution between two families of related stream surfaces. In the elaboration of this method Hirsch and Warzee used a finite element method as a numerical solution approach (Hirsch and Warzee, 1979). Wang extended the quasi-three-dimensional method to a full three-dimensional method by considering the two stream surfaces as variable (Wang, 1985). Füzy describes a method to calculate the blade profile geometry on curved stream surfaces, assuming a friction-free and incompressible flow field (Füzy, 1962). Detailed flow investigations were carried out by Carey and Mizuki in the wake of mixed flow impellers (Carey, 1985 and Mizuki, 1979). Geometry parameters and their effects on the impeller flow were systematically investigated in the scientific work of Felsch and Anschütz (Felsch, 1985 and Anschütz, 1994). Felsch is particularly concerned with the determination of the flow areas in the meridian section as well as the determination of the blade geometry (Felsch, 1985).

DIAGONAL BLADES

Previous studies at the Fluid System Dynamics Department at the Technical University of Berlin focused on the influence of different blade shapes in a mixed flow fan with fixed hub and shroud contour, pictured in Figure 1, left. (Thamsen and Treder, 2014)

The design of 4 different mixed flow blades has been performed based on axial and centrifugal design methods, which needed to be adapted to the diagonal geometry. Axial fans, mostly constructed with airfoil blades, operate on the principle of flow deflection and the generated lift.

Centrifugal fans produce more static pressure than axial-flow fans of the same diameter and the same rotational speed because of the additional centrifugal force (Bleier, 1998). Since the airfoil lift contributes only a small portion of the produced pressure in centrifugal fans, the improvement due to airfoil blades is not as pronounced in centrifugal fans as it is in axial flow fans (Bleier, 1998).

The profiling of blades in mixed flow fans has only led to marginal improvements in efficiency compared to non-profiled blades. It seems that the generated blade lift is therefore less significant in the mixed flow fans with hub and shroud contour, pictured in Figure 1. The flow deflection due to the centrifugal force is of much greater importance, similar to centrifugal fans.

Figure 1 shows 10 “Pfleiderer Blades”. These diagonal blades were adapted from the radial design method of Pfleiderer (Pfleiderer and Petermann, 2005). Pfleiderer considered the vortex in the opposite direction to the rotating blade channel. This vortex affects the relative velocity and thus the pressure conditions in the impeller and leads to a smaller deflection of the flow. To achieve the specified design point, this reduced deflection must be considered.

First infinitely many, infinitely thin blades are assumed. The three-dimensional effects of the flow and the influence of the finite number of blades and thickness are taken into account by Pfleiderer with empirical correction factors. Since no empirical factors are available for mixed flow fans, the existing correction factors for radial flow machines were used for the calculation of the outlet blade angle β_2. The determination of the inlet blade angle β_1 is done according to the shock free blade inlet criterion into the impeller. It is carried out using point-wise calculation of the blade, with adapted streamlines to the hub and shroud contour (Staerk, 2013).

A very good agreement to the theoretical design point resulted in the experimental studies. The best efficiency point (BEP) was slightly displaced towards low load range in the measured performance curves of the mixed flow fan (Thamsen and Treder, 2014).

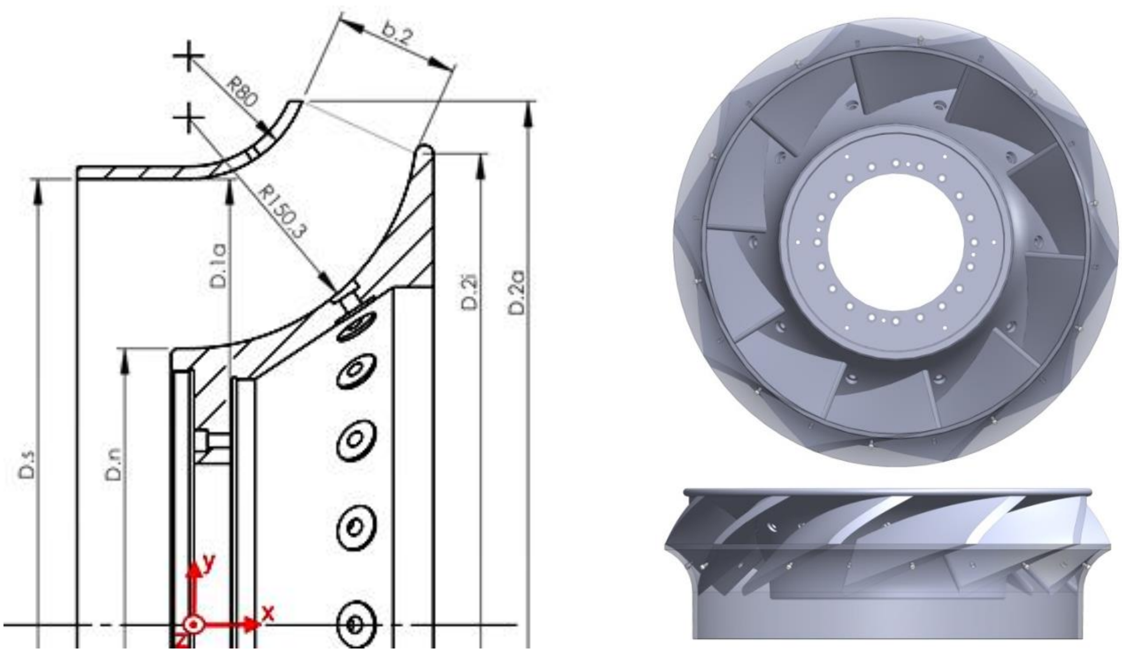


Figure 1: Hub and shroud contour of a mixed flow fan with a shroud radius of R80 (left) and ten backward curved constant thickness blades, installed in the mixed flow fan, adapted from the radial “Pfleiderer” blade design method

EXPERIMENTAL SETUP

The experimental investigations took place on a ventilator test stand corresponding to EN ISO 5801. The test stand is illustrated in Figure 2 and is currently capable of supporting tests at speeds of up to 14000 rpm. The inlet measuring orifice is calibrated with the LDA method (Laser-Doppler- Anemometry). A suitable inlet nozzle was constructed to allow a flow-conform transition from the measuring chamber to the inlet of the mixed flow fan.

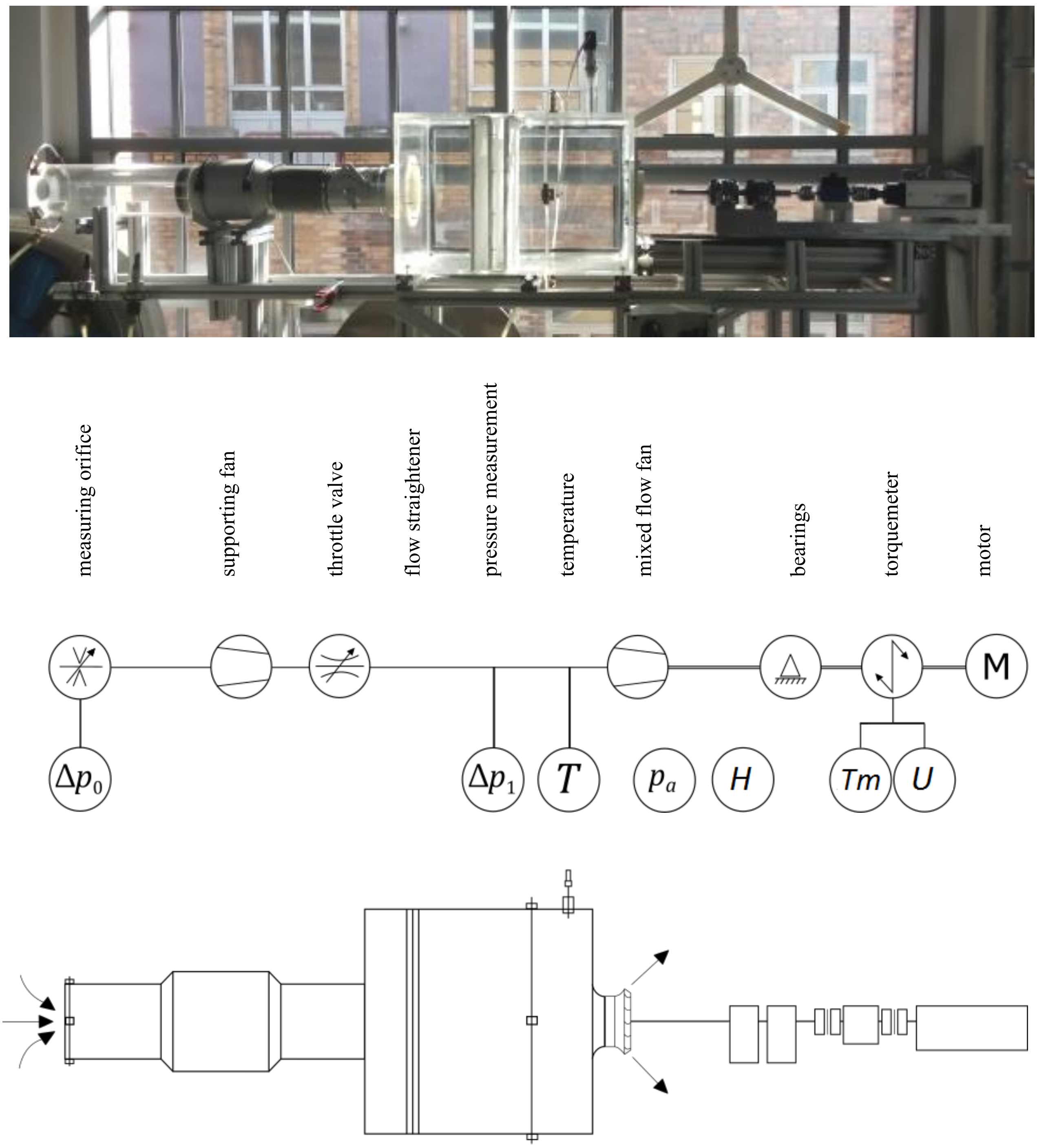


Figure 2: Ventilator test stand corresponding to EN ISO 5801, located at the Fluid System Dynamics Department at the Technical University of Berlin

The measurements of the mixed flow fan (D.2a = 0.085 m) were performed at five different speeds (5000, 7500, 10000, 11000 rpm) and also has been studied regarding different size. The theoretical calculations of the characteristic curves with the affinity laws were consistent with the measurement results. The affinity laws could be transferred for the investigated mixed flow fan (Treder and Thamsen, 2015).

The flow rate, the pressure increase and the efficiency were recorded with numerous measurement sensors which are integrated in the test stand, pictured in Figure 2. The data is generated with the sensors in the following measuring range and accuracy: piezoresistive pressure transducers [p_a 0.8… 1.2 bar, deviations < 0.1% and Δp_0, Δp_1 0…0.03 bar, < 0.2%, (1bar = 10^5 Pa)], a temperature sensor [0…100°C, ± 0.3+0.0005(T)], a hygrometer [0…100%, ± 2%] and a torquemeter [± 2Nm, < 0.1%]. The control of the test stand is supported via the measurement computer assuring a timely processing of measurement data.

VARIATION OF SHROUD RADIUS

In centrifugal fans the inlet flow is deflected from axial to radial direction. If the flow is decelerated, there is an increased probability of flow separation on the shroud. In mixed flow fans, similar to centrifugal fans, there also is a flow deflection from axial to diagonal direction. The probability of flow separation at the shroud is even increased because of the relatively wide blade channels.

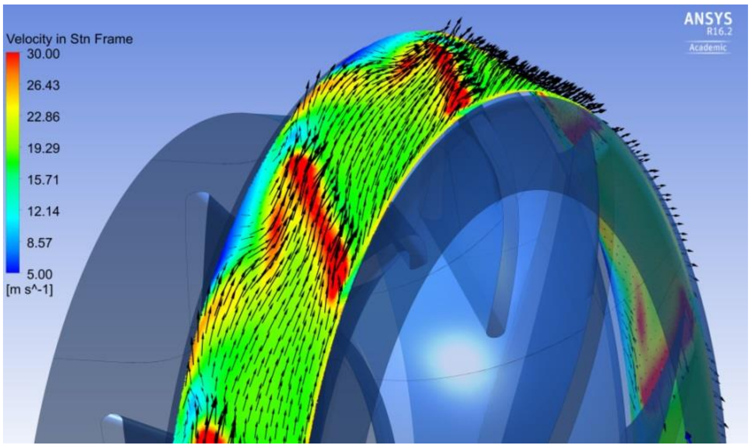


Figure 3: Flow separation on the shroud of a mixed flow fan with radius R80 and downstream angle χ =68°

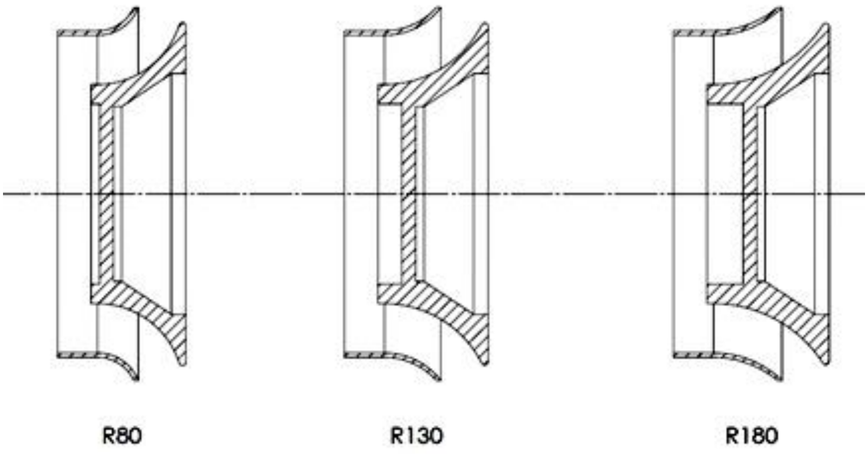


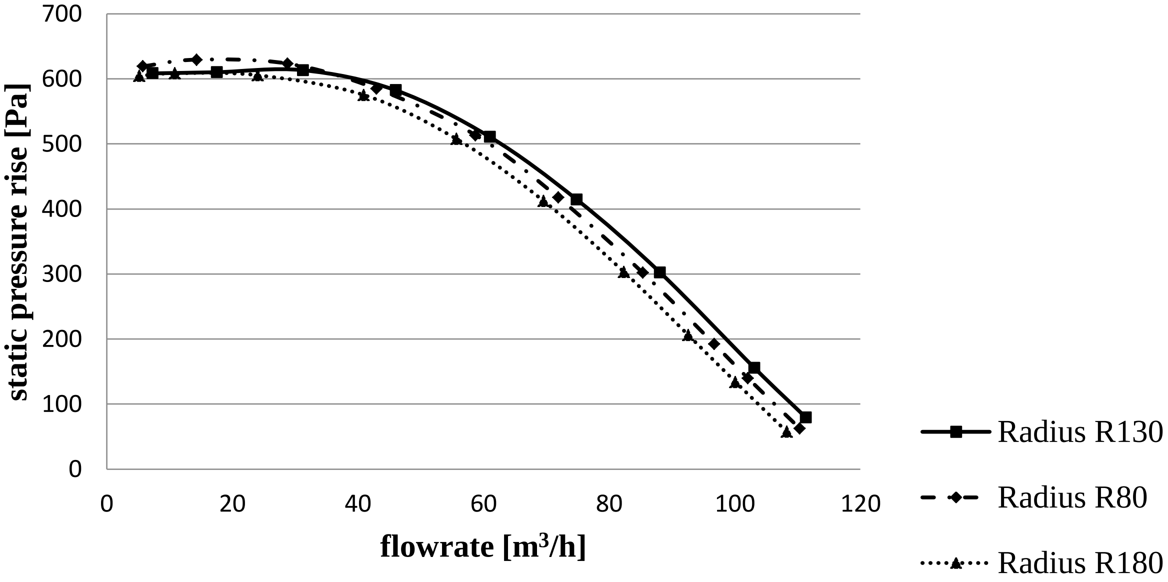
Figure 4: Modified shroud of a mixed flow fan with 3 different radii (R80, R130 and R180)

Previous PIV measurements in the wake of the mixed flow fan (Hammer, 2012) suggested flow separation on the shroud. The results of later numerical flow simulations of the mixed flow impeller also pictured a vortex structure on the shroud just before the impeller outlet (Figure 3, Figure 6). The flow seperation on the shroud was detected in the velocity measurements and also backed by calculation in the flow simulations (Karras, 2016).

In order to prevent the shroud separation, different hub and shroud contours for mixed flow fans were investigated. The radius of the shroud has been increased from R80 to R130 to R180, pictured in Figure 4. The blade width and the outer diameter have been kept constant in order to enable a good comparability of the experimental investigations.

Experimental results of different radii R80, R130 and R180

Three prototypes of the mixed flow fans with different radii were manufactured via selective laser sintering process and measured at the ventilator test stand, pictured in Figure 2. The characteristic curves are plotted in Figure 5 below.



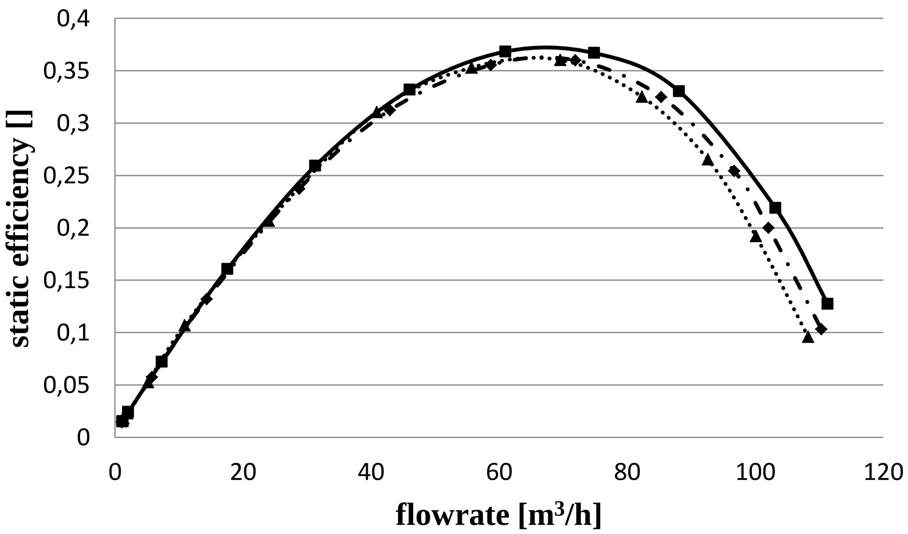


Figure 5: measured characteristic curves for three mixed flow fans with different shroud radii of R80, R130 and R180

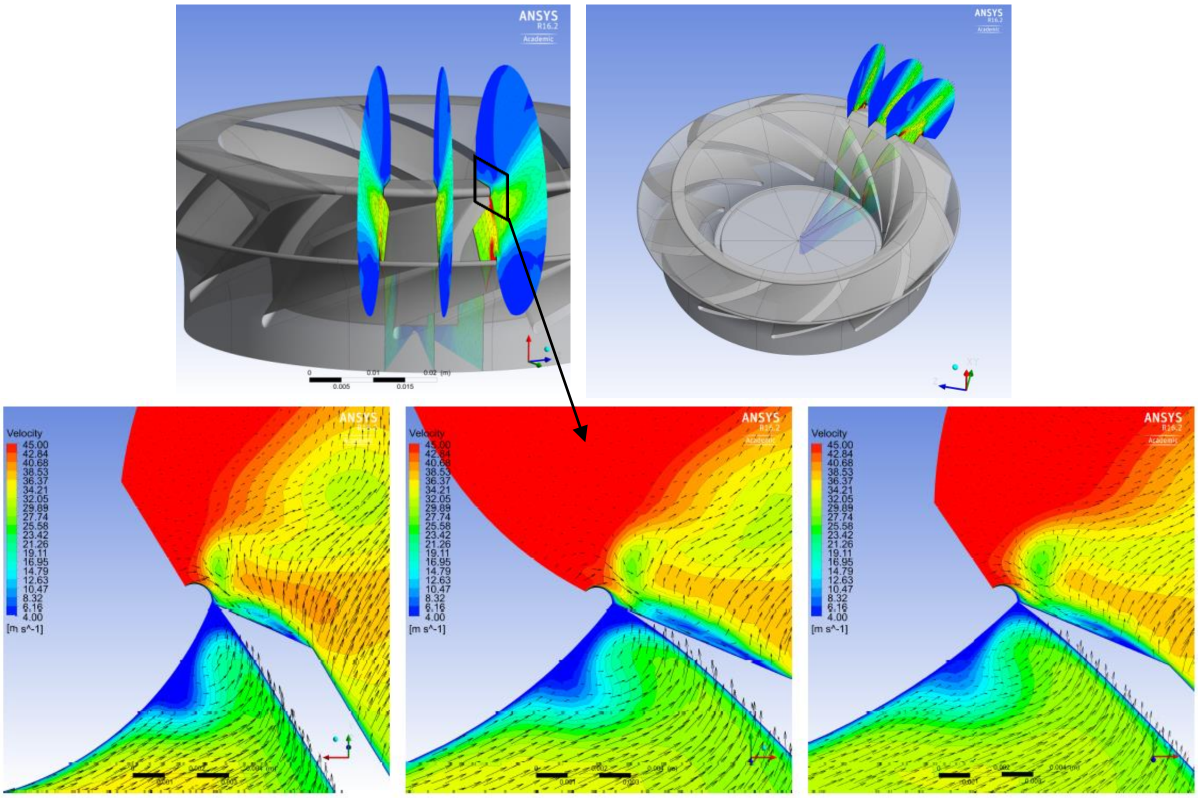
Based on the radius R80 (dashed dotted line) an increase in pressure and efficiency of the enlarged radius R130 was determined by the measurement results. A further increase in the radius from R130 (Figure 5, black line) to R180 (Figure 5, dotted line) leads to a deterioration of the performance curve and the efficiencies. With a constant outer diameter and a constant width of the fan blade, the blade channels are relatively long in a shroud contour of R180 and cause increased friction losses.

The blade design has been adapted to the different shroud contour with the chosen design point of a flowrate of 90 m³/h and a pressure rise of 330 Pa.

Previous investigations (Staerk, 2013) figured out that the radial design method for centrifugal impellers can be adapted for this type of mixed flow fan. However, there is a shift of the best efficiency point towards lower flow rates relative to the design point. The design methodology of Pfleiderer calculates the outlet blade angle β_2 with the use of empirical coefficients (Pfleiderer and Petermann, 2005). These coefficients are based upon a radial design with a high impact of centrifugal force in the airflow. Further research in the field of mixed flow fans and improved design methods with collected empirical data and adjusted coefficients may lead to improved efficiencies of mixed flow fans and to a better curve performance even.

Numerical results of different radii

In the meridional contour of a fan with a shroud radius of R80 and a downstream angle $\chi = 68°$ a shroud flow seperation is pictured in Figure 3. The flow simulation results of the mixed flow fans with the different radii R80, R130 and R180 in the area of the shroud outlet are plotted in Figure 6(a-c).



(a) $R80, \chi = 68°$ (b) $R130, \chi = 68°$ (c) $R180, \chi = 68°$

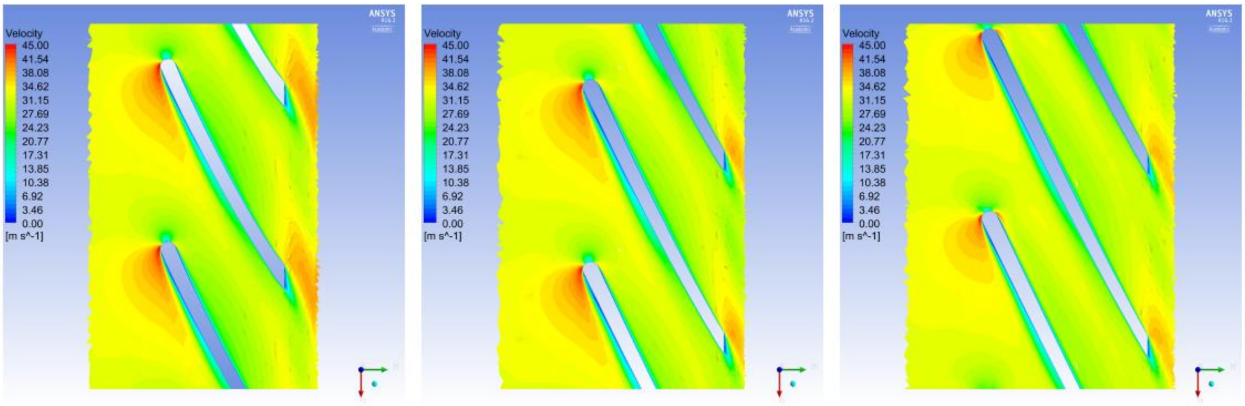
Figure 6a-c: Vortex structure of flow separation on the shroud for mixed flow fans of various radii

The turbulent flow was approximated using the Reynolds-averaged Navier-Stokes equations (RANS) and the Menter-Shear-Stress-Transport (SST) turbulence model. The numerical simulations were validated with the measurement results.

The simulation results identified a reduced vortex structure caused by the shroud flow separation with increasing radius from R80 to R180. With a very smooth flow deflection from the axial inlet to the diagonal outlet flow with a radius of R180 the flow separation on the shroud is still visible (Figure 6c). In the mixed flow fan with a radius of R130 an increased pressure rise at higher flowrates were measured with an improved efficiency rate (see Figure 5, black line). Although the shroud separation is still visible in the simulation results in Figure 6(b), assumingly reduced friction losses in the shorter blade channels in R130, compared to the relatively long blades in the contour with the radius of R180, led to the improved measurement results.

In Figure 7a-c, the blades and the blade channels are pictured two-dimensionally in the mid span between the hub and shroud with different shroud radii R80, R130 and R180. The relative velocities are plotted in the design point of the mixed flow fan.

The simulation results in Figure 7a-c suggest a deviated inlet flow towards the blade leading edge at the design point. The difference between the inlet flow and the inlet blade angle β_1 causes increased losses. Future research work on mixed flow fans should consider an adjusted calculation of the inlet blade angle β_1.



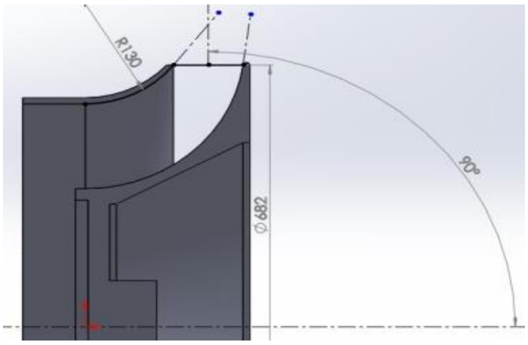
(a) $R80, \chi = 68°$ (b) $R130, \chi = 68°$ (c) $R180, \chi = 68°$

Figure 7a-c: blade to blade sections for mixed flow fans of radii R80, R130 and R180

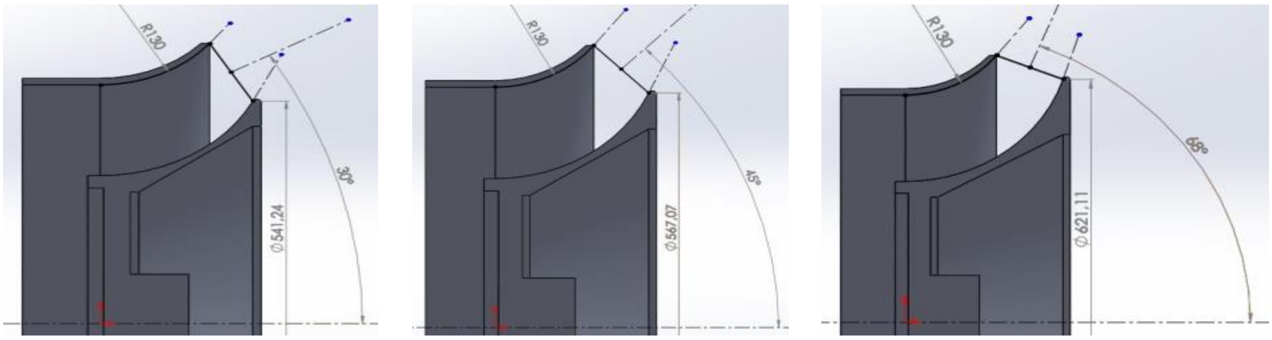
VARIATION OF DOWNSTREAM ANGLE

In centrifugal fans the flow is diverted from axial inlet to radial outlet, approximately 90° to the axis. In axial fans the inlet flow and the outlet flow occur roughly in parallel to the axis. The influence of different downstream angles between 0° (axial) and 90° (radial) on the characteristic curves was measured. The pressure rise, the flowrates and the efficiencies are investigated as a function of the downstream angle χ in the meridional section. The sectional view of the different mixed flow fans with downstream angles of $\chi = 30°$, $\chi = 45°$, $\chi = 68°$ and $\chi = 90°$ are pictured in Figure 8.

The downstream angle χ is defined between the axis of the fan wheel and the orthogonal to the connection line of the outer edge of hub and shroud, dimensioned in Figure 8. Previous examinations were carried out on the original mixed flow fan with a downstream angle of $\chi = 68$ °. The further selected downstream angles $\chi = 30°$, 45° and 90° are evenly distributed in the range between axial and radial outlet flow. The experimental investigations of the mixed flow fans took place on a ventilator test stand corresponding to EN ISO 5801, illustrated in Figure 2. The mixed flow fans were tested with free outlet conditions, without a volute.



a) χ = 90°



b) χ = 30° c) χ = 45° d) χ = 68°

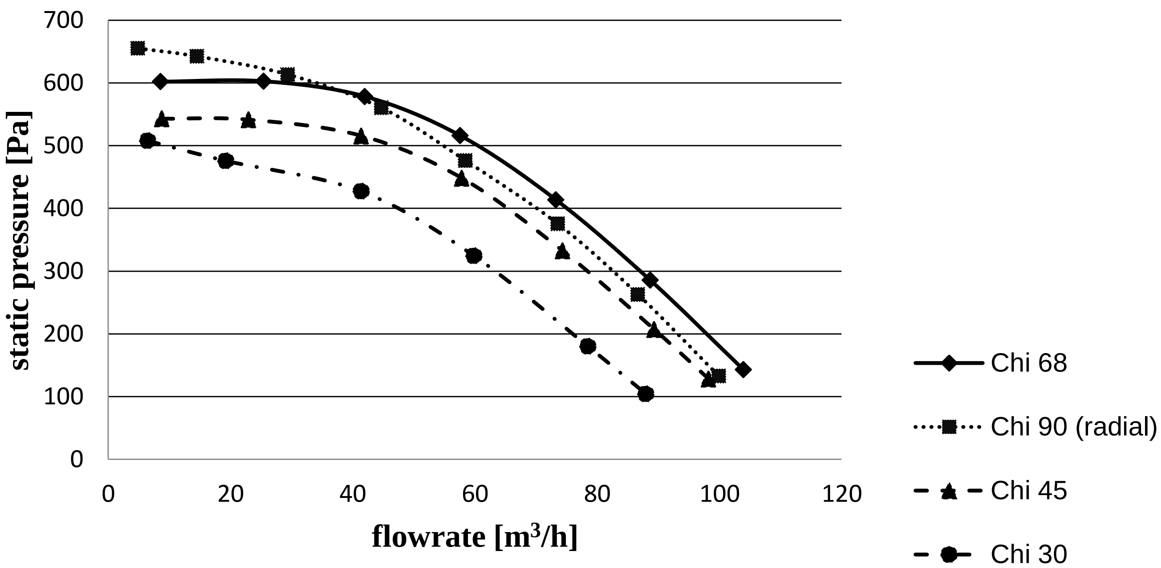
Figure 8a-d: Mixed flow fans with four different downstream angles χ, a) χ=90° (centrifugal), b) χ=30° (almost axial), c) χ =45° and d) χ=68°

Experimental results of different downstream angles

The measured characteristic curves are plotted in Figure 9. The variation of the meridional section with χ = 45° (Figure 9, dashed line) does not increase flowrates, comparable to axial fan types. Even reduced pressure increases at almost constant volume flows were measured.

Due to the lower downstream angle χ = 45°, the centrifugal forces are reduced in the impeller and the pressure rise was decreased. From lower downstream angles with χ = 30° a significant decrease of the characteristic curve, with deteriorated efficiencies has been measured. The characteristic curve is far off the theoretical design point. The blade design, which was adapted from the radial method (Staerk, 2013), is no longer suitable for fans with downstream angles smaller than χ < 45°.The best results with regard to the flowrates, pressure rise and efficiency are measured at the mixed flow fan with the downstream angle of χ = 68° with a static efficiency of 38% (Figure 9, black line).

As the basis for the efficiency calculation for the mixed flow fans, the static pressure was taken. The dynamic pressure caused by the airflow can be converted into static pressure by a volute or a diffusor. The mixed flow fan was analyzed without additional components downstream, therefore the dynamic part of the pressure is fully lost. The share of the dynamic pressure is much higher for axial fans with static pressure typically below 300 Pa, than for centrifugal fans (Radgen and Cory, 2008). 413 products of axial fans with static pressure < 300 Pa have been analysed by Radgen and Cory and overall static efficiency points from 20-50% were published. The overall static efficiencies vary up to ±30 %-points. This can lead to the conclusion that large efficiency improvement potentials are existent in currently used small axial fan products (Radgen and Cory, 2008).



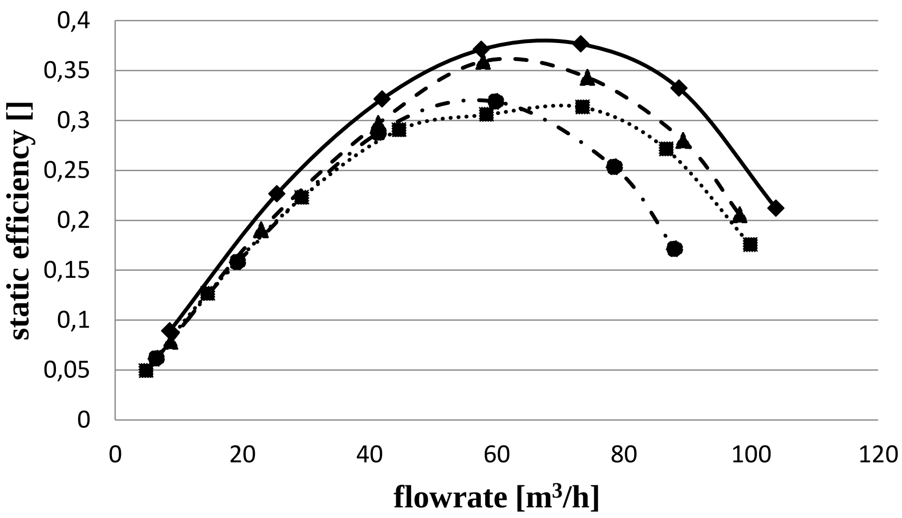


Figure 9: measured characteristic curves of four different mixed flow fans with downstream angles of χ= 30° (dashed dotted line), χ = 45° (dashed line), χ = 68° (black line) and χ = 90° (dotted line)

CONCLUSIONS

Main development objectives with the growing environmental awareness are still the increase of profitability and operational reliability and thus also increased requirements for the design method, efficiency rates and reduced noise. Axial and centrifugal fans are represented in a wide variety of different shapes on the market. However, mixed flow fans are still a rarity.

Since no empirical factors are available in the existing literature for mixed flow fans, the existing correction factors for centrifugal flow machines (Pfleiderer and Petermann, 2005) were used for the calculation of the outlet blade angle β_2. A very good agreement to the theoretical design point, with a slightly displaced best efficiency point (BEP) towards low load range, resulted in the experimental studies with a downstream angle χ = 68°. In the investigated mixed flow fan with a smaller downstream angle of χ = 30° (almost axial) large deviations from the design point were determined. Therefore, the adapted radial blade design methodology is apparently not applicable for mixed flow fans with downstream angles smaller $\chi < 45°$.

The impact of various downstream angles χ in a mixed flow fan with hub and shroud was measured. The investigations revealed that changing the downstream angle to smaller χ is decreasing the flow rates with smaller pressure rises. A too large flow deflection with a shroud radius of R80, cause flow separation at the shroud. While a large radius of R180 results in increased friction losses and deteriorated efficiencies. A shroud with a radius R130 is recommended for the investigated fan.

In axial fans the stalling dip occurs, when the flow in the partial load range detaches from the blades due to too large angles of attack. Axial fans should never be operated in the stalling range, increased turbulences and poor flow guidance requires low efficiencies and causes high noise levels. There were no stalling dips measured in the characteristic curves of the mixed flow fans with hub and shroud. The mixed flow fan can be integrated in fluid systems with axially parallel inlet and outlet flow as well as axial fans and already achieve a higher pressure rise in a single stage without stalling range.

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