EXPERIMENTAL INVESTIGATIONS OF MIXED FLOW FANS WITH VARIOUS DOWNSTREAM ANGLES

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SUMMARY

If the fan design point requires large flow rates with a sufficiently large increase in pressure, the recommended fan type is within the range between the axial and radial fan type, in the area of mixed flow fans. Mixed flow fans are still a rarity on the market and only a very few design recommendations are available. The impact of various downstream angles χ = 30° (almost axially) to χ = 90° (centrifugal) was evaluated for the flowrate and the pressure increase. Mixed flow fans can be easily integrated in fluid systems with axially parallel inlet and outlet flow, as well as radial outlet flow and represent a serious alternative in the application field between axial and radial fans.

INTRODUCTION

A fan moves airflow through a system, with the task of overcoming the existing friction and pressure resistance. Therefore, the respective operating point of the fan with a certain flow rate Q at the required pressure increase Δp depends on the system situation. Various system requirements are covered in a wide range by axial and radial fans on the market.

Centrifugal fans can often meet the requirements for the pressure increase in a single stage to overcome the resistance occurring in the system. Disadvantageous are the relatively large impeller diameter and the deflected radial outlet mass flow. Axial fans have axial inlet and also axial outlet mass flows, but sometimes need two or more impeller stages to meet the required pressure increase, similar to those of centrifugal fans. This increases the costs and also the installation space.

An obvious conclusion is the demand for a diagonal fan type that generates flow characteristics with relatively large flow rates and pressure increases, thus combining the radial and axial machines.

There exist good and established blade design methods for axial and radial fan blades. For a diagonal blade design, there are only a few recommendations given in the literature. 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.

Geometry parameters and their effects on the impeller flow in a mixed flow fan were systematically investigated in the scientific work of Anschütz and Felsch [1], [2]. Felsch is particularly concerned with the determination of the flow areas in the meridian section as well as the determination of the blade geometry [2].

In 1952 Wu published a quasi-three-dimensional design method for mixed flow fans [3]. 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 [4].

Wang extended the quasi-three-dimensional method to a full three-dimensional method by considering the two stream surfaces as variable [5]. Füzy describes a method to calculate the blade profile geometry on curved stream surfaces, assuming a friction-free and incompressible flow field [6]. Detailed flow investigations were carried out by Carey and Mizuki in the wake of mixed flow impellers [7], [8]. How the characteristic curves of a mixed flow fan are influenced by a change in the meridian contour is presented in this paper.

MIXED FLOW FAN CONTOUR

The investigated mixed flow fan was used for cooling large engines and is installed directly on the motor shaft. Figure 1 shows the mixed flow fan as a sectional view. The hub and shroud contour are each designed as a circular arc with radius R80 (shroud) and R150 (hub).

The fan was operated at a rotational speed of $n = 1500\ min^{-1}$. For the blade design, the fan speed n and the geometric dimensions of the hub diameter D.n, the outer diameter of the suction side D.s, the outer diameter of the blade leading edge D.1a, the outer diameter of the blade trailing edge D.2a and the blade width b.2 (specified in Figure 1) were determined due to the limited space inside the motor housing and were kept constant during the investigations. The calculated values for the pressure and flow coefficients are shown in Table 1.

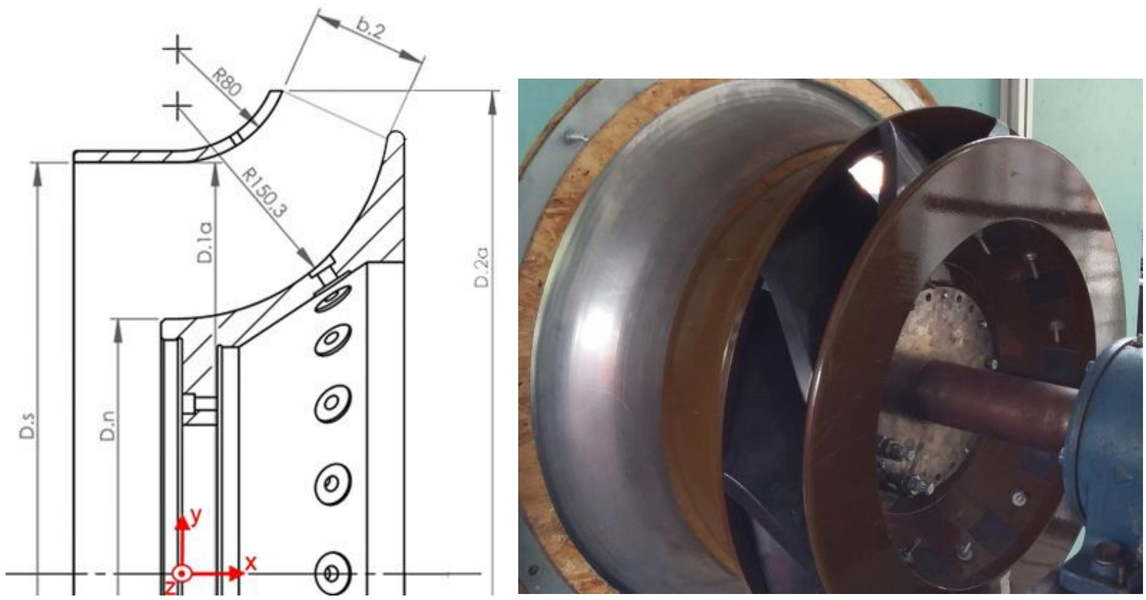


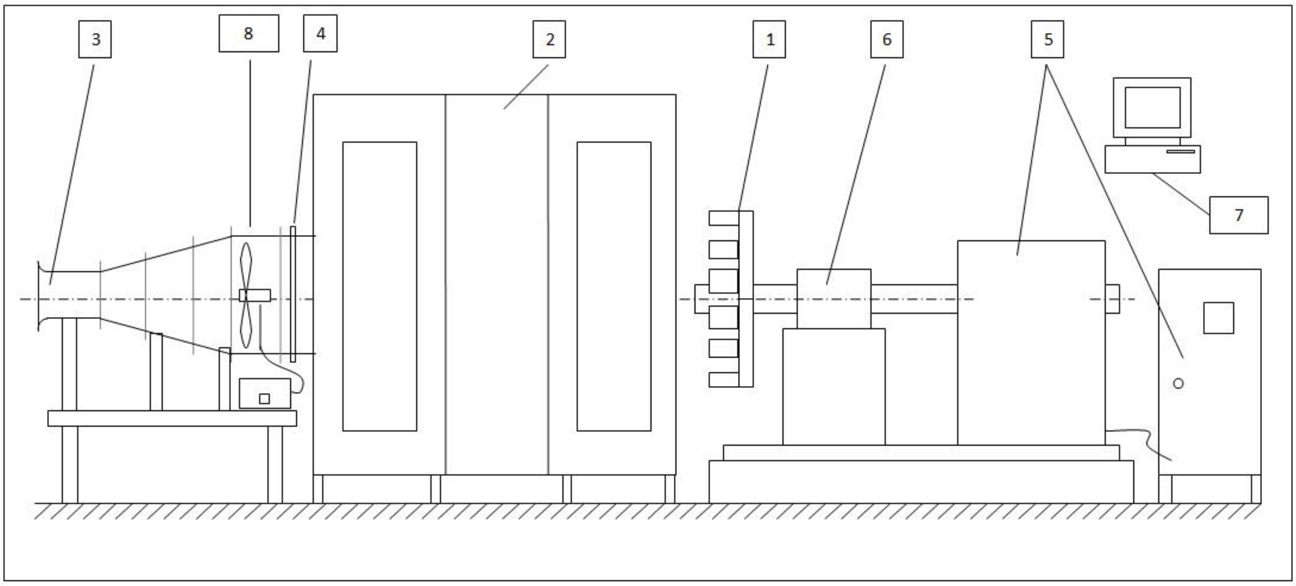
Figure 1: Geometric dimensions of the hub and shroud contour of a mixed flow impeller (left), mixed flow fan installed on the test bench (ISO 5801)

Table 1: design data and fan geometry of the investigated mixed flow fan

	symbol	**value**	**unit**
rotational speed	n	1500	min^{-1}
pressure rise	Δp	500	Pa
flow rate	Q	2	m^3/s
speed ratio	σ	0.8	[-]
diameter ratio	δ	2.3	[-]
specific speed	n_q	129.5	[-]
pressure coefficient	ψ	0.28	[-]
flow coefficient	φ	0.1	[-]

How the radial respectively the axial blade design methodology can be adapted to the blade design of mixed flow fans has been investigated at the Fluid System Dynamics Department of the TU Berlin [9]. The adapted radial blade design according to the method of Pfleiderer [10] showed the best agreement of the theoretical design point with the characteristic measurement results.

Experimental setup



1	Fan	3	Induction nozzle	5	Drive with frequency converter	7	Computer
2	Measurement chamber	4	Orifice	6	Bearing unit	8	Supporting fan with frequency converter

Figure 2: schematic illustration of the ventilator test stand corresponding to EN ISO 5801

The experimental investigations took place on a large and a smaller ventilator test stand corresponding to EN ISO 5801, schematically illustrated in Figure 2. To plot the characteristic curves the flow rate Q, the pressure rise Δp (total to static) and the efficiency η (total to static) have been measured with piezoresistive pressure transducers, temperature sensors, an air moisture analyzer, an incremental encoder for speed and a force sensor for determining the torque on the pendulum motor. The control of the test stand is supported via the measurement computer assuring a timely processing of measurement data.

Gap flow rate

A suitable inlet nozzle was constructed to allow a flow-favored transition from the measuring chamber to the inlet of the mixed flow fan. The installation of the mixed flow fan on the test bench according to ISO 5801 is shown in Figure 3. Due to the existing gap between the rotating impeller and the fixed inlet nozzle, an additional volume flow Q_{gap} must be taken into account in the blade design process.

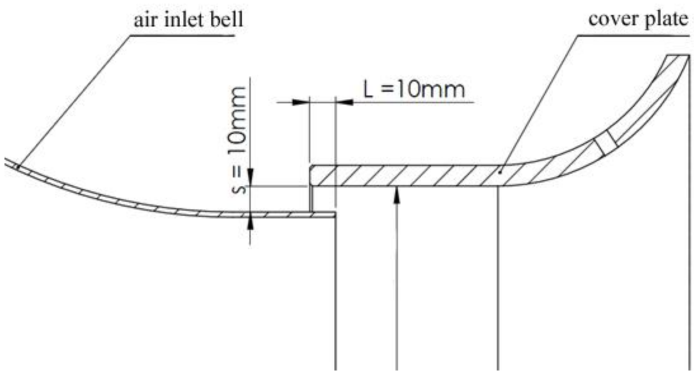


Figure3: mixed flow fan on the test bench (ISO 5801), installed with inlet bell

There are several analytical approaches for the calculation of the gap flow rate. Felsch [2] proposed a gap flow calculation especially for mixed flow fans (see Formula 1 below). As confirmed previously by Anschütz [1], the present investigations also achieved a good agreement of the estimated value according to Formula 1 and the measured gap flow rate via hot wire anemometer with a deviation of less than 0.2 %.

$$\frac{Q_{gap}}{Q} = \frac{4s}{D_s}\sqrt{\frac{\phi}{\varphi^2}\cdot\left(\frac{D_s}{D_{2a}}\right)^4 + \frac{1}{\left(1-\frac{2s}{D_s}\right)^4}} \qquad (1)$$

Assumptions for blade design

To design the mixed flow blades according to Pfleiderer [10], the hydraulic efficiency η_h can be usually estimated on the basis of comparative values of existing turbomachines. But no empirical data for mixed flow fan design is yet available. Based on the preliminary investigations [9], the hydraulic efficiency is assumed to be $\eta_h = 0.65$.

The iterative blade calculation has revealed a recommended blade number of z = 12. With regard to the data comparability with the preliminary investigations, the number of blades was set as unchanged parameter with z = 10. The blade thickness d was determined in terms of a sufficient material strength. The required minimum of the blade thickness d = 12 mm was used, due to screw connection of the blades between the hub and shroud contour.

For the calculation of the suction side, Pfleiderer [10] recommends an approximately equal static moment of the inner and outer streamline. The blade angles β_1 at the suction side are calculated with the criterion of the impact-free entry.

The approach of the reduced performance according to Pfleiderer [10], due to the finite number of blades, is used to calculate the blade outlet angles β_2. After determining the blade angles on the suction and pressure side, the distribution between β_1 and β_2 is determined. A backward curved blade shape was selected with a pointwise calculation of the blade angle distributions as a function of the radius $\beta(r)$. The blades were manufactured using the vacuum casting process.

VARIATION OF DOWNSTREAM ANGLES

In ISO 13349 [11] fan types are categorized into radial, axial and mixed flow fans as verbal definition according to the airflow through the impeller. In Table 2 these definition of the flow directions are also given by using the downstream angle χ.

In the investigated mixed flow fans the downstream angle χ is defined between the rotation axis and the straight line of the outflow. This straight line is the orthogonal of the connecting line of the two points D_{2a} (outer edge of the shroud at the outlet) and D_{2i} (outer edge of the hub at the outlet), see Figure 5. Due to the large possible range of mixed flow fans (χ = 1°... 89°), a further division of this fan type based on the downstream angle χ is advisable.

Table 2: categorization of fans according to the downstream angle

fan type	**downstream angle**	
radial	$\chi = 90°$	axial inlet and radial outlet mass flow
axial	$\chi = 0°$	axis-parallel inlet and outlet mass flow
mixed flow	$0° < \chi < 90°$	axial inlet and diagonal outlet mass flow

Previous investigations on the influence of the shroud radius were carried out on the mixed flow fan with a downstream angle of χ = 68°, pictured in Figure 4. The recommended shroud radius of R130 was taken from the preliminary investigations [12]. For each of the modified meridian contours of the hub and shroud, a customized blade design was carried out, depending on the downstream angle.

The four investigated mixed flow fans with downstream angles of χ = 30°, χ = 45°, χ = 68° and χ = 90° are shown in Figure 5. The measurements were made as a model study using the fan laws. The transferability of the measurement results of mixed flow fans of different size was confirmed in preliminary investigations [13]. The fan prototypes were fabricated by selective laser sintering.

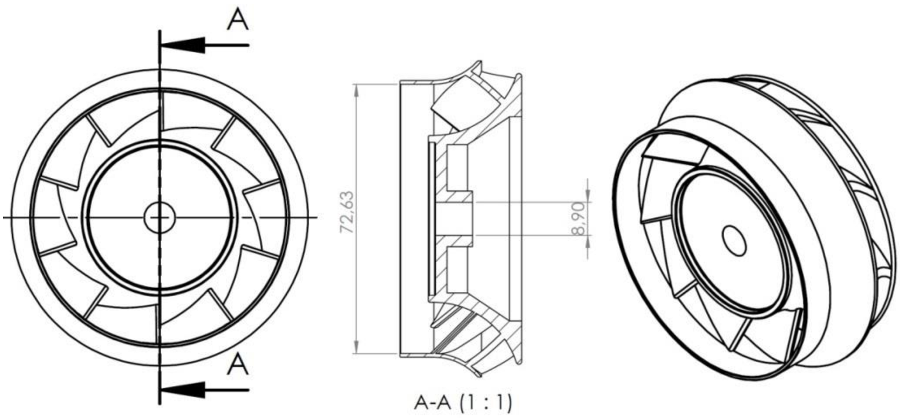


Figure 4: CAD drawing of the mixed flow fan with shroud radius R130 and downstream angle χ=68°

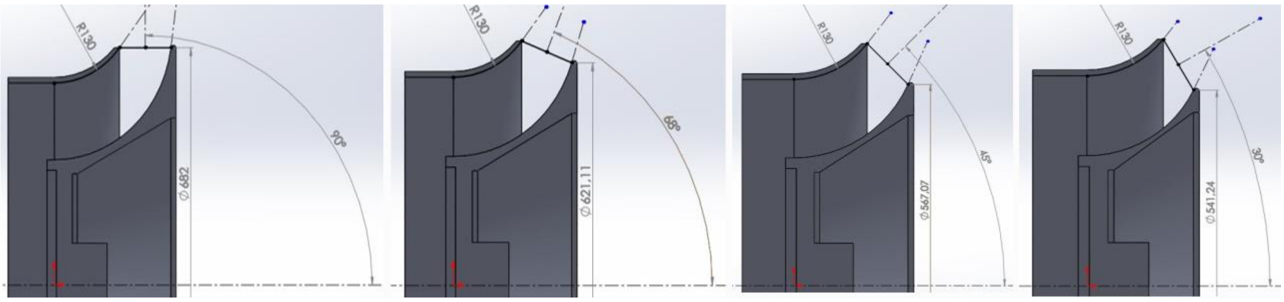


Figure 5: Mixed flow fan with 4 different downstream angles, from left to right: χ=90°, χ=68°, χ=45°and χ=30°

Measurement results

The characteristic measurements of the mixed flow fans with different downstream angles χ were performed on a test bench according to ISO 5801 at the Fluid Systems Dynamics Department of the TU Berlin. The measured dimensionless characteristic curves are plotted in Figure 6.

The design point (DP) with a flow coefficient of $\varphi = 0.1$ and a pressure coefficient of $\phi = 0.28$ (see Table 1) is pictured in the diagram as a dashed line. The investigated mixed flow fan in original size of $D_{2a} = 0.682$ m and a downstream angle of $\chi = 68°$ was measured in good correlation with the design point. The displayed measurement results of the smaller fan model with $D_{2a} = 0.085$ m and $\chi = 68°$ (see Figure 6, red line) is slightly below the design point (DP). The best efficiency point (BEP) is shifted into the range of smaller flow rates.

Reducing the downstream angle to $\chi = 45°$ (see Figure 6, light blue line) is decreasing the maximum flow and pressure coefficients. The measured values are below the theoretically design point. In the further reduction of the almost axially downstream angle to $\chi = 30°$ (see Figure 6, black line), a significant decrease in the pressure rise and the maximum achievable flow rate was measured. The blade design method for mixed flow fans, adapted by the radial design method, showed significant deviations between the theoretical design point and the measured characteristic curves for the mixed flow fans with downstream angles of $\chi = 30°$ and $\chi = 45°$.

The centrifugal fan $\chi = 90°$ (see Figure 6, blue line) achieves increased pressure rise in the range of smaller flow rates. The relatively large cover plate (radius R130), with also large blade widths b, cause a large impeller channel surface. The surface is also relatively rough, due to the manufacturing process by selective laser sintering and contributes to increased friction losses.

The best measurement results in terms of flow rate, pressure rise and efficiency were determined with the mixed flow fan with a downstream angle of $\chi = 68°$ (see Figure 6, red line).

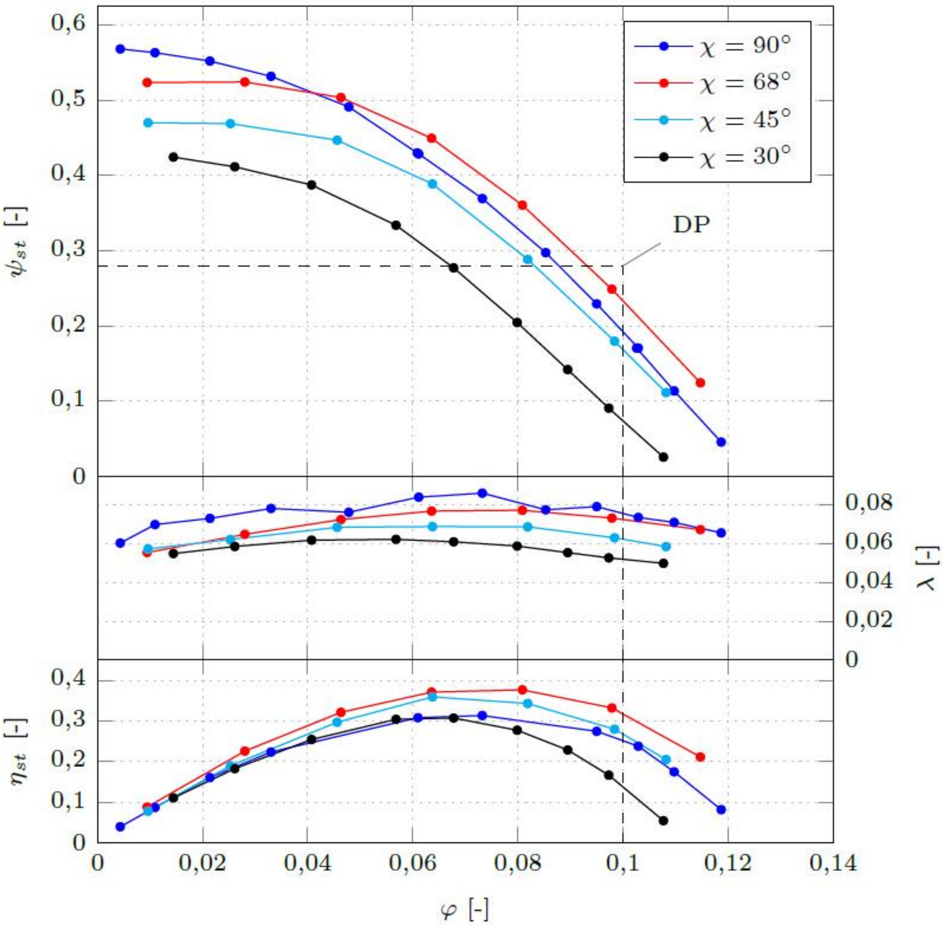


Figure 6: measurement results of 4 mixed flow fans with different downstream angles, χ=90° (dark blue line), χ=68° (red line), χ=45° (light blue line), χ=30° (black line),

CONCLUSIONS

The mixed flow fan combines large flow rates with relatively high increase of pressure, thus combining the characteristics of the axial and radial fans. For the investigated mixed flow fan with a downstream angle of χ = 68°, a diagonal blade was designed with the adapted radial method according to Pfleiderer [10]. The measured operating point of ten backward curved blades, with a linear blade angle distribution β(r), is in a good agreement with the theoretical design point in the original size of D.2a = 0.682 m. The measurement results of the smaller fan model with D.2a = 0.085 m are shifted slightly below the design point (DP). The blade design requires a correct estimation of the gap flow rate Q_{gap}. For mixed flow fans with an existing shroud contour, the equation (see Formula 1) according to Felsch and Winkler [2] is suitable.

The performance curves of axial flow fans are characterized by an operating range and also a stalling range at small flow rates. The measured stalling dip in the characteristic curve is more intense with increasing hub ratios. Axial fans should not be operated in the stalling area, due to increased turbulence and noise emissions with also reduced efficiencies. In the investigated mixed flow fan with hub and shroud contour, no stalling dip has been measured. This is advantageous, especially in applications requiring large hub ratios.

The method used to design the mixed flow fan blades depends on the downstream angle χ. Reducing the angle from χ= 68° to χ= 45° and χ= 30° resulted in reduced flow and pressure coefficients. For smaller downstream angles (χ= 30° and χ= 45 °), an adapted axial blade design method will possibly yield to smaller deviations between the measurement results and the theoretical design point.

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