AXIAL IMPELLER-ONLY FANS WITH OPTIMAL HUB-TO-TIP RATIO AND ADAPTED BLADES

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SUMMARY

This study targets at determining impeller-only axial fans with optimal hub-to-tip ratio for highest achievable total-to-static efficiency. Differently from other studies, a holistic approach is chosen. Firstly, the complete class of this fans is considered. Secondly, the radial distribution of blade sweep angle, stagger angle, chord length and camber are varied to adapt the blades to the complex flow in the hub- and tip region. The tool being used is an optimization scheme with three key components: (i) a database created beforehand by REYNOLDS-averaged NAVIER-STOKES- (RANS-) predicted performance characteristics of 14,000 designs, (ii) an artificial neural network as a metamodel for the fan performance as a function of 26 geometrical parameters, (iii) an evolutionary algorithm for optimization, performed with the metamodel.

In general, the hub-to-tip ratios for the class of axial impeller-only fans proposed by the optimization scheme are smaller than those obtained applying the classic rules. A second outcome are the shapes of adapted blades which deviate substantially from the classic and even the state-of-the-art “swept-only” or “swept with dihydral” design. Chord length, stagger and sweep angle are distributed from hub to tip in a complex manner. The inherent reason is that the scheme tries to minimize not only the dynamic exit loss but also frictional losses due to secondary flows in the hub and tip region which eventually results in the maximum achievable total-to-static efficiency.

Upon request the authors will provide the full geometry of four impellers analyzed in some detail in this study to any individual for experimental validation or further analysis of their performance.

Keywords: axial fan, impeller only, hub-to-tip ratio, optimization

INTRODUCTION

For several applications the impeller-only axial fan in a duct-type casing is the preferred choice. Examples are dry cooling towers in thermal power plants, installations for locomotive and automotive engine cooling, railway and automotive air conditioning systems, heat pumps, etc. The appropriate hub-to-tip ratio of the fan impellers (Fig. 1) has been discussed for many years. A historic example is the graph in Fig. 2 from the last edition of the classic text book by Eck [1], 1972, where the recommend specific speed and diameter[1] as well as the smallest hub-to-tip ratio of such fan impellers are indicated in a so-called Cordier-diagram.

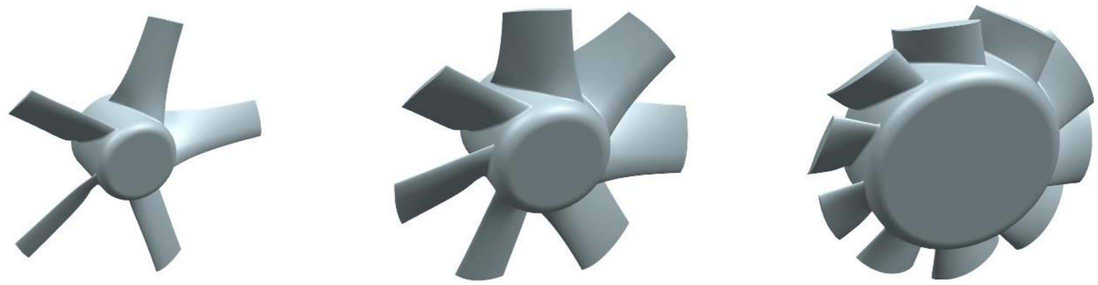


Figure 1: Typical axial impellers with increasing hub-to-ratio (classic blade shape); duct-type casing is not show.

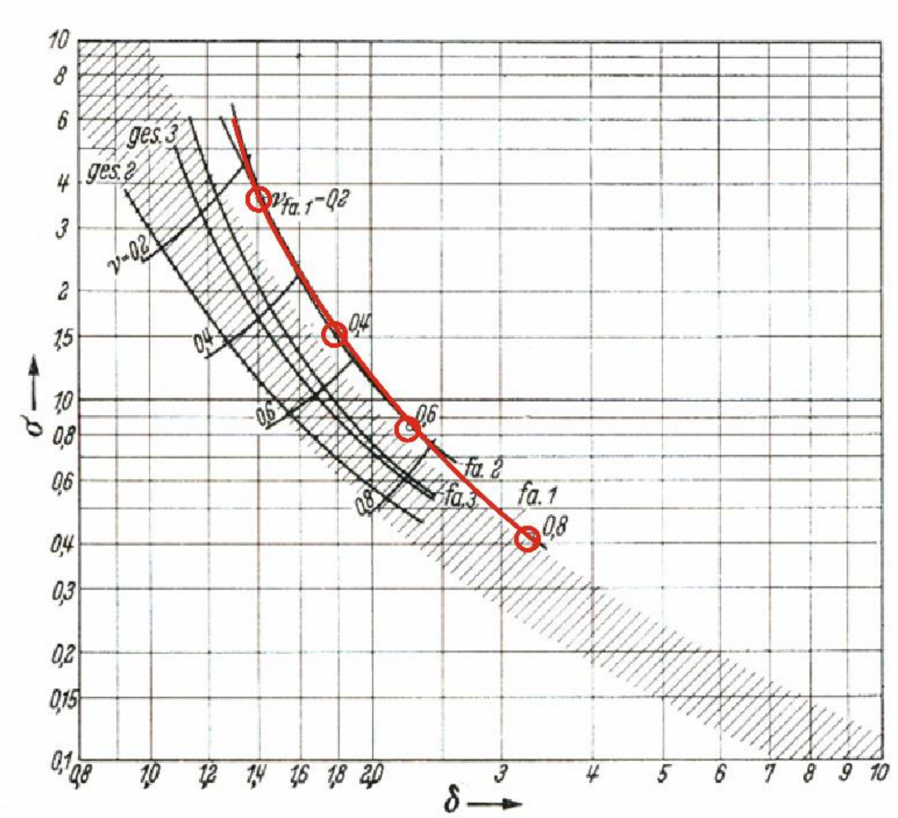


Figure 2: Axial fans and minimum hub-to-tip for different fan assemblies according to Eck [1], 1972; the curve *fa.1*, highlighted in red, is valid for free-exhausting impeller-only fans which are of interest in this study; $\nu_{fa.1}$ is the recommended hub-to-tip ratio; *ges.2* and *ges.3* refers to assemblies with outlet guide vanes and outlet guide vanes plus diffuser and the overall efficiency.

[1] Here and throughout this study, all values of non-dimensional coefficients refer to the optimal point of operation, i.e. the operating point on the performance characteristic, where the aerodynamic efficiency is maximal. The frequently used index "*opt*" is omitted for brevity.

From fundamental aerodynamic findings, hints for the minimum hub-to-tip ratio had been established by Strecheltzky (see Horlock [2]), de Haller [3] and Schiller [4]. Nevertheless, designers frequently try to reduce the hub-to-tip ratio further. One driver is cost - a larger hub can be expensive. Another reason is reducing the blocked area of an impeller at stand-still. In automotive cooling units, for instance, the fan is mostly not running, since the outer airstream provides the forced cooling at moderate and high speeds of the vehicle. A third argument is that too large a hub shades the cooler matrix from the fan flow. This is frequently incorrect. The near-hub sections of blades on an inappropriate hub may experience or even generate a substantial back flow. This can increase substantially the area of a downstream heat exchanger *without* through flow.

Several authors worked on methods for designing impellers with very low hub-to-tip ratio. For instance, Lindemann *et al.* [5] suggested a design method for small hub-tip-ratio with swept blades, based on an empirical axial and tangential velocity distribution at the impeller. In a more recent paper Wang and Kruyt [6] studied small hub-to-tip ratio fans. Among others, they analysed the influence of non-aerodynamically shaped parts of the blades and showed “that the presence of non-airfoil sections near the root has a minor influence on the pressure coefficient and hence on the total-to-static efficiency (of the fan), due the formation of a vortex upstream from the blades near the hub. Overall, the ‘main blade’ part well represents the aerodynamic performance.” The potential drawback in efficiency was not discussed in detail. Nevertheless, the idea in their subsequent paper [7] was to quantify the potential of an overall blade sweep, dihedral, and skew on the aerodynamic performance of such fans. They found only minor benefits.

This study targets at determining highly efficient impeller-only axial fans with minimum hub-to-tip ratio. The efficiency considered is the total-to-static efficiency. Differently from other studies, a holistic approach is chosen. Possible design points cover the complete range common for this class of fans. The radial distribution of blade sweep angle, stagger angle and chord length are varied to adapt the blades to the complex flow in the hub- and tip region. The tool being used is an optimization scheme developed and validated by Bamberger [8], see also Bamberger & Carolus [9].

METHODOLOGY

Dimensionless coefficients used

In this study the common definitions of non-dimensional coefficients are used [10]. Q is the volume flow rate, Δp a pressure rise, n the rotational speed of the impeller, D_{tip} the rotor outer diameter (and approximately the clear diameter of the casing), ρ the (constant) density of the gas, P the shaft power. The volume flow and pressure rise coefficients and the efficiency are

$$\phi \equiv \frac{Q}{\frac{\pi^2}{4} D_{tip}^3 n}, \; \psi \equiv \frac{\Delta p}{\frac{\pi^2}{2} D_{tip}^2 n^2 \rho}, \; \eta \equiv \frac{Q \Delta p}{P}, \tag{1, 2, 3}$$

and the specific diameter and speed

$$\delta \equiv \frac{D_{tip}}{\left(\frac{8}{\pi^2}\right)^{\frac{1}{4}} \left(\frac{\Delta p}{\rho}\right)^{-\frac{1}{4}} Q^{\frac{1}{2}}}, \; \sigma \equiv \frac{n}{\left(2\pi^2\right)^{-\frac{1}{4}} \left(\frac{\Delta p}{\rho}\right)^{\frac{3}{4}} Q^{-\frac{1}{2}}}. \tag{4, 5}$$

Depending on the case of application, ψ and η can be defined using either the total-to-total (index "*tt*") or the total-to-static (index "*ts*") pressure rise.[2] The total-to-total pressure rise Δp_{tt} is a measure for the total energy transferred from the shaft to the fluid and equals the difference between the total pressure downstream of the fan and the total pressure upstream of the fan. In many applications, however, the kinetic energy downstream of the fan dissipates in the surroundings. In that case, the total-to-static pressure rise Δp_{ts} (i.e. Δp_{tt} diminished by the so-called exit loss) is more adequate to describe the design point and ψ_{ts} and η_{ts} become the relevant dimensionless quantities. In contrast, σ and δ are always defined with Δp_{tt}, with most probably an exception in Fig. 1, which will be discussed below.

The geometrical quantity which is most relevant in this paper, is the hub-to-tip ratio

$$\nu \equiv \frac{D_{hub}}{D_{tip}} \quad . \tag{6}$$

Optimization scheme

A short summary of the optimization scheme is given in this section. Details can be found in [8, 9]. Three key components are essential: (i) a database of performance characteristics of 14,000 different axial fan impellers, (ii) a metamodel for the fan performance as function of the geometrical parameters varied, (iii) an evolutionary algorithm as the optimization method.

i) The database was created beforehand by Reynolds-Averaged Navier-Stockes- (RANS-) predicted performance characteristics of 14,000 individuals in the class of axial fan impellers. The CFD model comprises only one blade channel using periodic boundary conditions at the lateral surfaces. The other boundary conditions are: Fixed mass flow at the inlet, ambient pressure at the outlet and no slip at the walls. The inlet in placed one fan diameter upstream of the impeller and the outlet is placed two fan diameters downstream of the impeller. The turbulence was modeled using the k-ω SST model. The solver is Ansys CFX™. Numerical grid-generation and evaluation of the RANS-simulations were automated. Combinations of in total 26 geometrical parameters were determined systematically with a method of Design of Experiment (DoE). Two minor constraint are

- All RANS-simulations were performed for a 0.3 m diameter impeller running at 3000 rpm. This leads to a typical chord-based Reynolds number of 200.000. If the results of this study will be applied for fans with substantially smaller or larger Reynolds-numbers, at least the absolute value of the predicted efficiency could be scaled with Reynolds scaling laws.
- The blades are composed of 4-digit NACA airfoil sections with the parameters “maximum thickness”, “maximum camber” and their respective “chordwise positions” .

ii) The metamodel is based on an artificial neural network. It enables predicting the performance characteristics including efficiency and the circumferentially averaged flow velocity in the impeller exit plane of any impeller made of a reasonable combination of the 26 geometrical parameters. The neural network type selected is the multi-layer perceptron (MLP). MLPs consist of the input layer, an arbitrary number of hidden layers and the output layer. The number of hidden layers and the number of neurons in each of them determines the model complexity. A too simple model will lead to large errors because of insufficient flexibility. On the other hand, a too complex model will suffer from over-fitting effects, i.e. the error will be small on the training data but high on fresh data that

[2] In ISO 5801 [11] the total-to-total pressure rise is simply called the "fan pressure” with the symbol p_f. The total-to-static pressure rise is called "fan static pressure” p_{fs} which must not be mistaken for the true static pressure rise of the fan.

was not used for the training. Therefore, the available data was split into training and test data and the model complexity was optimized aiming at a minimal error on the test data.

iii) The actual optimization is performed with the metamodel. For optimization an evolutionary algorithm is implemented. One essential advantage of evolutionary algorithms is the ability to find the global optimum which is considered important for the present work. The main disadvantage compared to local optimization algorithms (e.g. gradient based) is the high number of function evaluations that is required for convergence. Due to the extreme quickness of the metamodels, however, this disadvantage is less relevant for the present study. The main optimization target is always the maximization of η_{ts} with the constraint that the targeted design point must be fulfilled.

The complete scheme is implemented in Matlab™ and requires standard PC computer resources only.

Parameters varied and fixed

Fig. 3 shows the geometrical parameters involved. In Tab. 1 the parameters are compiled, which are varied throughout the optimization. The maximum blade thickness is fixed to 5 % of the chord length. Bamberger in [8] identified the sensible range of aerodynamic design parameters ϕ and ψ_{ts} of axial impeller-only fans which were designed in the same way except the maximum blade thickness was limited to 12 % instead of 5 %. This design space is considerably broader than suggested by the conventional Cordier-band. Fig. 4 visualizes Bamberger's space, as it is more or less adopted for this study, Tab. 2.

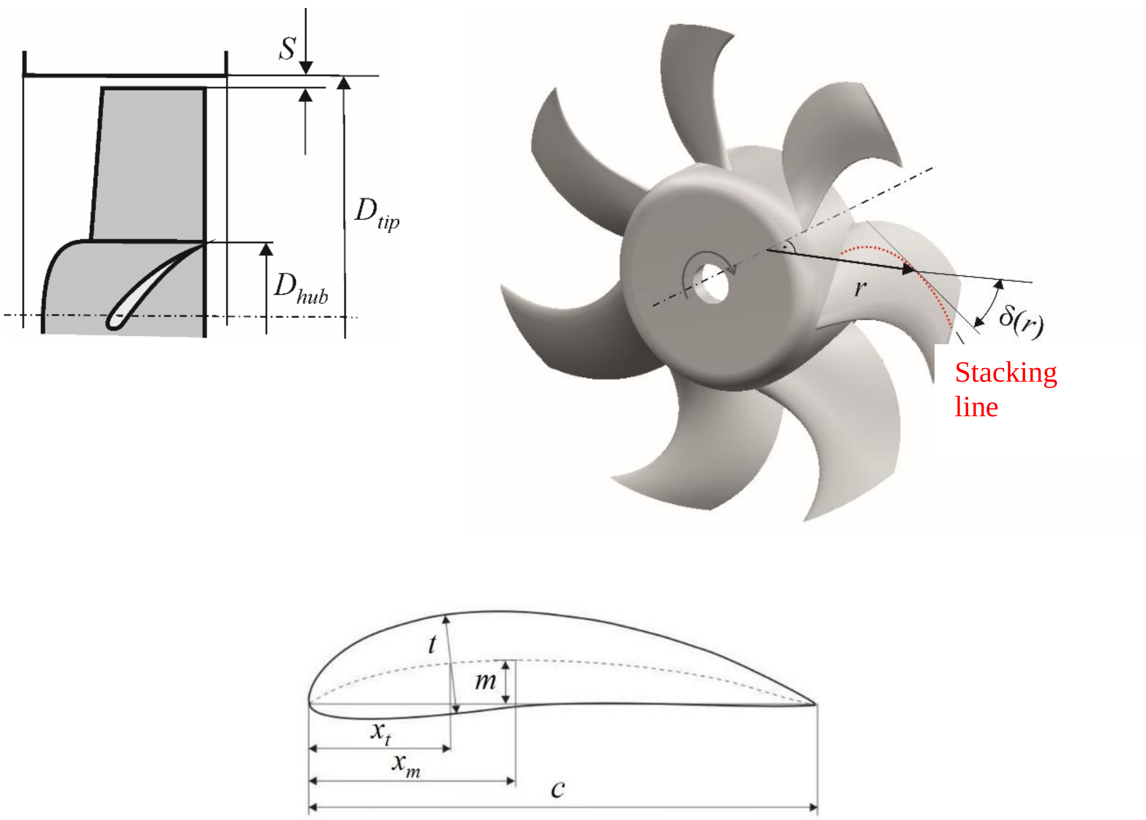


Figure 3: Illustration of geometrical parameters. Top left: Impeller in the duct-type casing. Top right: Definition of the sweep angle. Bottom: 4-digit NACA section.

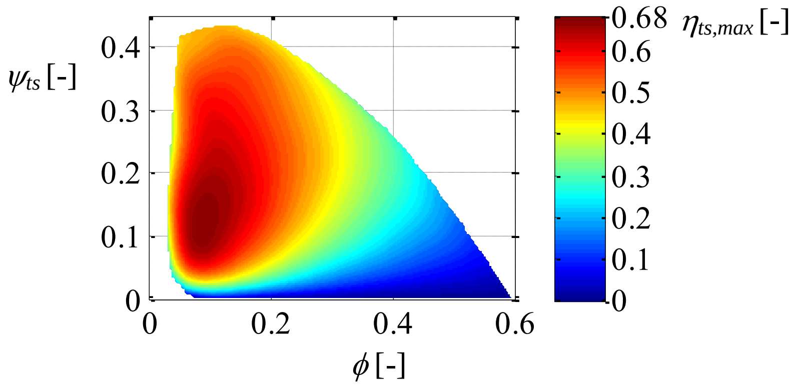


Figure 4: Performance of the class of single-stage impeller-only axial fans; from BAMBERGER [8].

Table 1: Geometrical parameters varied

Parameter		Range	Comment
Hub-to-tip ratio	ν	0.3 - 0.7	
Number of blades	z	5 - 11	Only integers
Chord-length ratio	c / D_{tip} [a]	0.13 - 0.33	
Maximum camber	m / c [a]	0 - 0.15	
Pos. of max. camber	x_m / c [a]	0.1 - 0.7	4-digit NACA sections
Pos. of max. thickness	x_t / c [a]	0.1 - 0.5	
Blade sweep angle	δ [a]	-60° - +60°	

[a] defined at three equidistant locations between hub and tip

Table 2: Aerodynamic design parameters varied

Parameter		Range
Pressure rise coefficient (at design point)	ψ_{ts}	0.1 - 0.4
Volume flow rate coefficient (at design point)	ϕ	0.1 - 0.45

RESULTS AND DISCUSSION

A systematic design of impellers in the design space yields the hub-to-tip ratios plotted in the upper plot of Fig. 5. It is important to remember that the lower and upper bounds for ν were set 0.3 and 0.7, respectively. This means that for fans designed for large values of ϕ and low to moderate of ψ_{ts} a lower hub-to-tip ratio could be in principle feasible. Such a design, however is impossible with the scheme utilized, because the initial training data for the metamodel were deliberately confined to the range of ν = 0.3 - 0.7. The maximum achievable total-to-static efficiency for each design is depicted in the lower plot of Fig. 5. Clearly, the fans with the highest efficiencies are those designed for pairs ϕ / ψ_{ts} in the region of the lower left corner with a hub-to-tip ratio $\nu \approx$ 0.3 - 0.4 accordingly.

Selected fans (i.e. pairs of ϕ / ψ_{ts}) in Tab. 3 illustrate the results. In contrast to conventional designs the optimization suggests blade shapes with an unexpected radial distribution of chord length, stagger angle and especially sweep angle.

The metamodel also yields the resulting circumferentially averaged flow field at the impeller exit, Fig. 6. Seemingly the optimal blade shape causes a meridional velocity component at the fan exit c_{m2}, with a maximum in the middle or outer blade section. This means that the volume flow rate is not evenly distributed in the bladed annulus of the impeller: In the critical hub region the through flow is reduced. The tangential velocity c_{u2} which is a measure of energy transfer to the fluid, is shown as well. In all graphs the velocities are normalized with the blade tip speed $\pi D_{tip} n$, indicated by a star as a superscript.

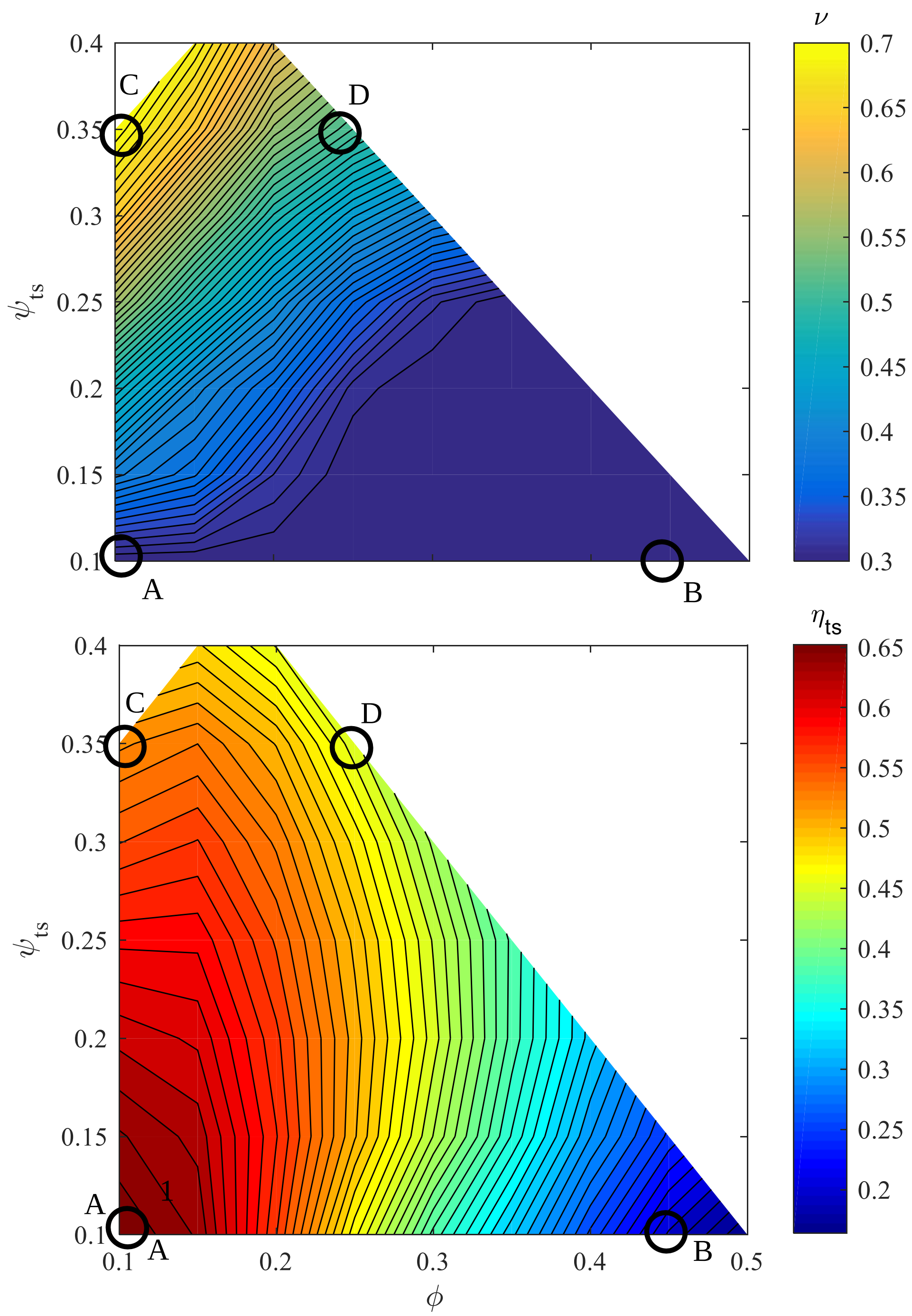
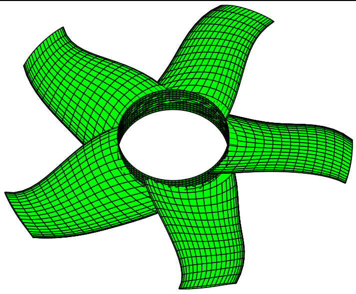
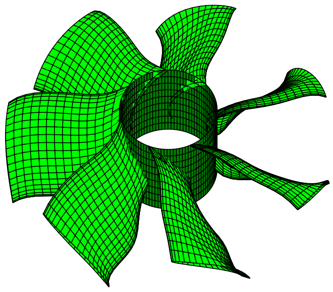
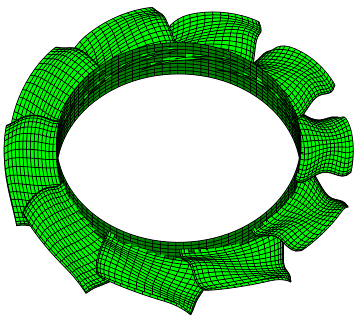
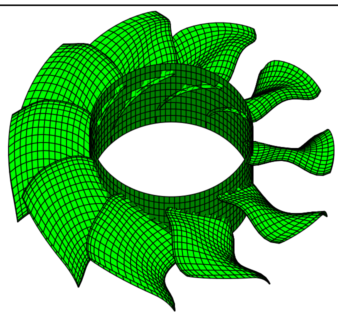


Figure 5: Upper: Hub-to-tip ratio of optimal impellers, lower: total-to-static efficiencies of these impellers. The design points denoted with letters refer to Fig. 6.

Despite a small region of backflow the most efficient fan is #A. For the same design point an impeller was designed with the standard text book method, choosing a free vortex, i.e. rc_{u2} = const and c_{m2} = const., in agreement with the requirement of radial equilibrium. The respective distributions are compared in Fig. 7. The c_{u2}-distributions are similar, but c_{m2} is different. However, the chosen hub-to-tip ratio $\nu = 0.31$ for the text book design is by far below the classic limits according to Strecheltzky, de Haller and Schiller. The latter is a general observation: The hub-to-tip ratios suggested by the optimization scheme are always smaller than those obtained applying the classic rules.

This hold true for the initially shown values in Ecks' Fig. 1[3] as well. Of course, highly efficient fans with the smaller hub-to-tip ratios require an adapted blade design which deviates substantially from the classic and even the state-of-the-art swept-only- or swept with dihydral-design. The decisive parameters chord length, stagger and sweep are distributed in a complex way which is an outcome of the optimization scheme. Ultimately the scheme tries to minimize not only the dynamic exit loss but also frictional losses due to secondary flows in the hub and tip region.

Tab. 3 Selected fans.

#	ψ_{ts}	ϕ	ν	η_{ts}	
A	0.1	0.1	0.31	0.66	
B	0.1	0.45	0.30	0.20	
C	0.35	0.1	0.70	0.52	
D	0.35	0.25	0.50	0.45	

[3] Some caution is necessary when interpreting Fig. 1. Most probably the *fa.1* curve is obtained when defining σ and δ with Δp_{ts} and not Δp_{tt}. Otherwise it does not make sense to present different curves for different fan assemblies.

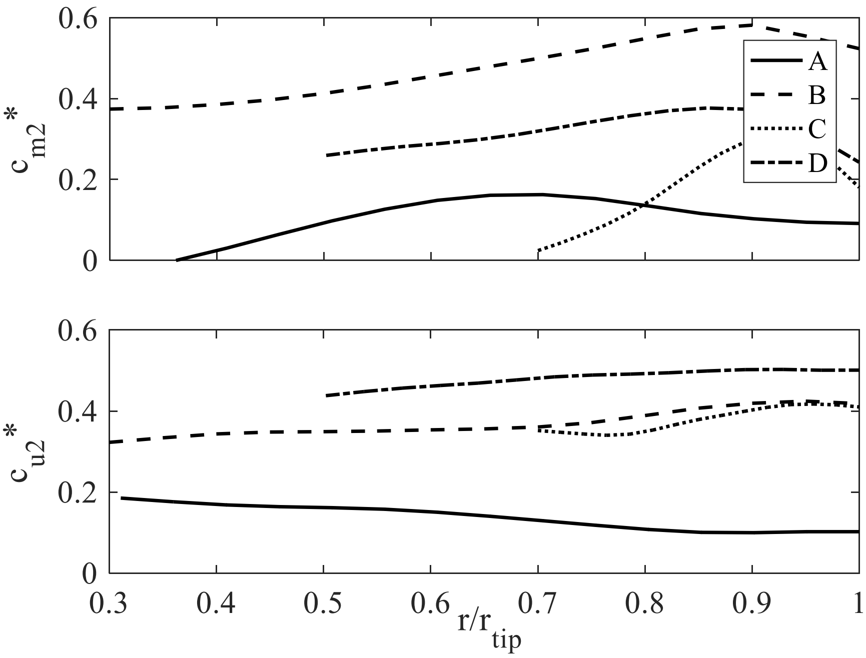


Figure 6: Distributions of the meridional (through flow) and tangential flow velocities in the impeller exit plane.

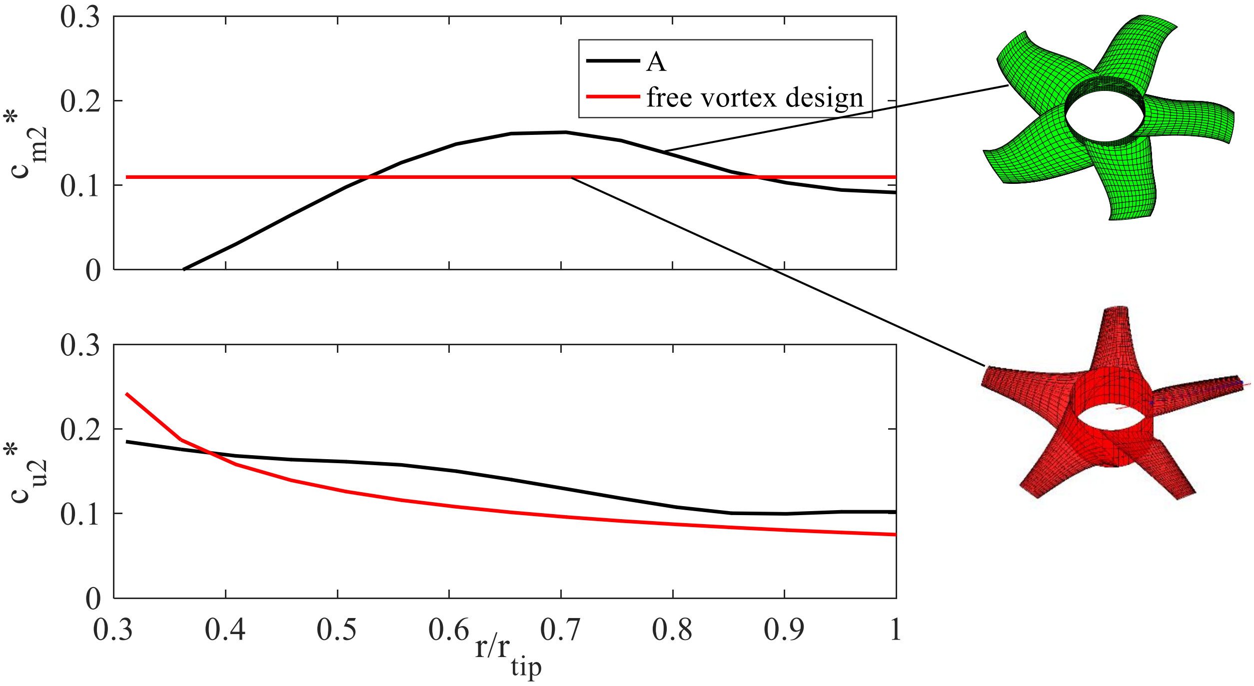


Figure 7: Comparison of optimal fan impeller #A with a standard text book design (i.e. free vortex, rc_{u2} = const and c_{m2} = const. according to the requirement of radial equilibrium).

SUMMARY AND CONCLUSIONS

Objective of this study was the design of impeller-only axial fans with optimal hub-to-tip ratios for highest achievable total-to-static efficiency. This in contrast to other studies where a primary design target is a very small hub, with the consequence that aerodynamically shaped blades in the hub region are obsolete and the flow in the hub is disturbed substantially by secondary flow.

Differently from other studies, a holistic approach was chosen. Firstly, the complete class of these fans is considered, not only one particular design case. Secondly, an optimization method is applied, which allows determining the optimal hub-to-tip ratio, the radial distribution of blade sweep angle, stagger angle and chord length. The classic hub-design rules by by Strecheltzky, de Haller and Schiller need not to be applied.

In general, the hub-to-tip ratios for the class of axial impeller-only fans proposed by the optimization scheme are smaller than those obtained applying the classic rules. A second outcome are the shapes of adapted blades which deviate substantially from the classic and even the state-of-the-art “swept-only” or “swept with dihydral” design. Chord length, stagger and sweep angle are distributed from hub to tip in a complex manner. The inherent reason is that the scheme tries to minimize not only the dynamic exit loss but also frictional losses due to secondary flows in the hub and tip region. The exit flow distributions deviate for instance from the free vortex design with standard text book methods.

Essentially, the optimization method used is based on RANS-simulations. The underlying RANS-method have been validated in the past for several examples. Nevertheless, upon request the authors will provide the full geometry of the four impellers analyzed in some detail in this study (impeller A, B, C, D) to any individual for experimental validation or further analysis of their performance[4].

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