Aerodynamic and aeroacoustic optimization of a small centrifugal fan with backward-curved blades by means of inverse design

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Abstract

Centrifugal fans with backward-curved blades are characterized by a relatively high efficiency compared to other types of centrifugal fans. In contrast, there is an overall louder, tonal sound and higher rotational speeds. Therefore, in home appliances such as clothes dryers, rather squirrel cage fans are used. However, the influence of the fan unit increases with the need of a continuously rising efficiency of the overall device. Hence, the use of centrifugal fans with backward-curved blades for operation in commercial tumble dryers with respect to aerodynamic and aeroacoustic properties will be examined. In the first steps, a reference fan has been designed using classical design guidelines. The fan is numerically studied with a hybrid Computational Fluid Dynamics/Computational Aeroacoustics approach and the results are experimentally validated with the in-duct method according to ISO 5136. The reference fan will be further optimized in terms of its aerodynamic and aeroacoustic properties. For this purpose, the "inverse design" method is used to extend the scope of classical design rules. Using design of experiments (DoE) and genetic algorithms, various parameters and their influence on fan efficiency and sound radiation are examined. In further project steps, promising fan designs will again be investigated numerically and subsequently validated experimentally.

Keywords: Flow Acoustics, Fans, Inverse Design, Optimization

1 INTRODUCTION

Modern home appliances such as tumble dryers are characterized not only by an attractive design and high functionality. Environmental issues and increased awareness of sustainability among the consumers are central to the development of tumble dryers and other HVAC-systems (Heating, Ventilation & Air Conditioning). The energy efficiency class, which is an indication of the efficiency of the overall system, and the sound emission indication on the EU energy label serve as an important tool for many costumers when buying home appliances. Accordingly, manufacturers are interested from an economic as well as ecological point of view to continuously improve these aspects. Since the overall devices become more and more efficient, the influence of smaller subsystems become apparent. For a further increase in efficiency, these must also be taken into account.

The project HELNoise (High Efficiency Low Noise Heatpump Dryer - FKZ 13FH014PA5), funded by the German Federal Ministry of Education and Research (BMBF), aims to improve fans for process air generation in heat pump clothes dryers. Process air fans in heat pump dryers are usually designed as squirrel cage fans. This type of fan is characterized by its compactness and broadband operational sound. However, due to the high degree of secondary flows and flow separation zones inside the blade channels, squirrel cage fans achieve lower peak efficiencies than other centrifugal fan types. Centrifugal fans with backward-curved blades can reach higher efficiencies, but tend to have an overall louder and tonal sound along with higher rotational speeds required.

The aim of this work is to use the inverse design method to design and optimize a centrifugal fan with backward-curved blades. Based on an existing reference model [1], a fan with identical specifications has been designed using this method and the efficiency as well as the sound radiation should be improved to meet the criteria for use in heat pump clothes dryers. Evolutionary algorithms will be used to provide faster and more efficient optimization than just rely on empirical investigations. In the following, the inverse designed fan will be used as a comparison for the further optimizations and is referred to as the baseline model.

2 INVERSE DESIGN METHOD

The Inverse Design method is based on the three-dimensional, compressible design approach of M. Zangeneh [2], which is implemented in the program package TURBOdesign Suite (TDsuite) by Advanced Design Technology (ADT). In classical or direct design strategies, the main dimensions of the turbomachine are calculated using empirical or semi-empirical relationships and the blades are designed with regard to various factors. Centrifugal fans often use simple circular arc blades or two-dimensional pointwise calculated blades from a manufacturing point of view. The flow field thus results from the blade shapes determined in this way. With Inverse Design, the main dimensions are determined analogously to the direct methods. By contrast, the blade shape is calculated iteratively for a given flow field. By specifying the load on the blade, an optimum blade design can therefore be determined for a specific flow distribution. The blade loading is specified by the circumferentially averaged mean tangential velocity $r\overline{V}_\theta$. The derivative with respect to the meridional coordinate m is directly related to the pressure loading on the blade. This results in the following relationship for incompressible potential flow [2]

$$P^+ - P^- = \frac{2\pi}{B}\rho\overline{W}_{mbl}\frac{\partial r\overline{V}_\theta}{\partial m} \tag{1}$$

Here, P^+ and P^- represent the pressure on the upper and lower surface of the blades respectively, B the number of blades, and $\overline{W}_{mbl}$ the mean meridional relative velocity of the blade surface. The preliminary design of the fan by TURBOdesign Suite provides the meridional geometry as well as a normalized form of the Euler work between the inlet and outlet of the blade passage. For centrifugal fans this work is represented by [3]

$$rVt^* = \frac{\Delta p_{tot}}{\rho U_2^2} \tag{2}$$

where Δp_{tot} represents the fan total pressure rise including all loss terms and U_2 the circumferential speed at the blade trailing edge. The work rVt^* to be done by the fan is applied between the inlet and outlet where a swirl-free inflow is assumed. The rVt^* value is hence set to zero at the blade leading edge. Using the expression $\partial r\bar{V}_\theta/\partial m$ the blade loading distribution is applied to different sections along the meridional coordinate. Subsequently the loading between hub and shroud is interpolated onto a numerical grid and the blade contour is calculated iteratively until specific criteria is met. Figure 1 shows an example of the rVt^* and blade loading distribution for the baseline fan and the corresponding blade shape.

3 BLADE LOADING OPTIMIZATION

3.1 Methodology

The shape of the blade loading distribution is in the simplest case given by two parabolic curves and an intermediate straight section, as shown in figure 1. This can be controlled for the hub and shroud by four parameters each. **1** Start point of the straight section, **2** End point of the straight section, **3** Slope of the straight section, **4** Leading edge loading.

These eight parameters are used in the optimization process to investigate their influence on the efficiency as well as the sound radiation of the fan. For the optimization, the elitist multiobjective genetic algorithm NSGA-II is used in conjunction with the TURBOdesign Suite in a direct approach. NSGA-II uses non-dominated sorting and elite selection to ensure that the best variants of a population are preserved, which makes it a very fast and efficient algorithm [4]. In the direct approach, no computational fluid dynamic (CFD) simulations are carried out during the optimization process, and instead the search for the best variants is done analytically by means of

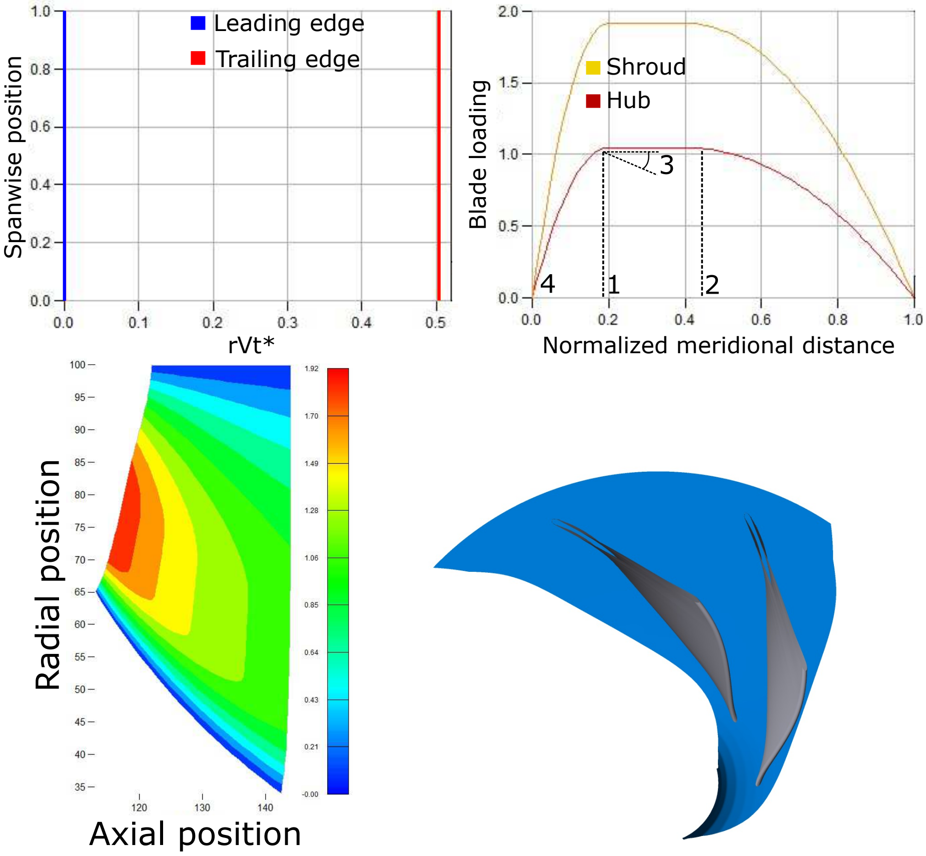

Figure 1. Top left: Normalized Euler work at leading and trailing edge between hub and shroud. Top right: Blade loading at hub and shroud per meridional distance. The loading shape can be controlled by four parameters. Bottom left: Interpolated and integrated blade loading contours. Bottom right: Corresponding blade shape for the baseline fan.

Inverse Design. Subsequently, promising designs are checked via CFD simulations. The optimization is carried out in two independent runs, on one hand in terms of high efficiency and on the other hand to a low sound radiation.

3.2 Optimization targets

The efficiency of a turbomachine is influenced by a large number of parameters. As targets of the efficiency optimization of the fan, the secondary flow factor (SF factor) and the total profile loss were chosen. The SF factor represents a measure of meridional secondary flow tendency on the blade suction side via gradients of the reduced static pressure coefficient C_p. It has been shown by Zangeneh et al. [5] that there is very good correlation between gradients of C_p and the intensity of secondary flows. Therefore by controlling C_p at specific locations in the fan the formation of secondary flows can be minimised. The reduced static pressure coefficient is given by

$$C_p = \frac{p^* - p_r}{1/2 \rho U_2^2} \tag{3}$$

In equation (3) p^* represents the rotary stagnation pressure and p_r the reduced static pressure. According to Johnson and Moore [6] the rotary stagnation pressure and the reduced static pressure are given by $p^* = p + 1/2\rho W^2 - 1/2\rho\omega^2 r^2$ and $p_r = p - 1/2\rho\omega^2 r^2$ respectively, where p is static pressure, W the relative velocity, ω the angular velocity and r is the radius. The total profile losses result from the summation of the blade suction and pressure side losses which are calculated from the blade surface velocity distribution [3]. When both of the target variables are minimized, an increase in efficiency is to be expected due to lower flow-induced losses.

The main objectives of the optimization with regard to low sound radiation are the total noise at a specific observer position and the total profile loss. The loss of the profile as a target is supposed to ensure that as little efficiency as possible is lost while reducing the sound radiation of the fan. The total noise is the sum of the sound generation due to fluid displacement and an acceleration of the force distribution around the blade. These two mechanics are referred to as thickness and loading noise respectively. To determine the total noise, the Ffowcs Williams-Hawkings (FW-H) model is used. The FW-H equation is an inhomogeneous wave equation and can be derived as an extension of the Lighthill analogy for moving surfaces from the basic equations of fluid dynamics. It can be represented in a very compact way as [7]

$$\Box^2 p' = \frac{\partial}{\partial t}[\rho_0 v_n \delta(f)] - \frac{\partial}{\partial x_i}[(p-p_0)n_i\delta(f)] + \frac{\partial^2}{\partial x_i \partial x_j}[H(f)T_{ij}] \tag{4}$$

Here, $\Box^2$ is the wave or d'Alembert operator, p' is the acoustic pressure fluctuation in the far field, ρ_0 is the density of the undisturbed air, v_n is the local normal velocity on the blade surface, $\delta(f)$ the Dirac delta function, p and p_0 the local and undisturbed pressure respectively, $H(f)$ the Heaviside function and T_{ij} the Lighthill stress tensor. The moving surface is represented by $f(\vec{x},t) = 0$, so that $\nabla f = \vec{n}$ where $\vec{n}$ is the outward normal surface vector. In equation (4) the viscous shear forces over the blade surfaces are neglected in the second term on the right hand side because their contribution to the generation of sound often is minimal [8]. The first term on the right hand side of the FW-H equation describes the sound generation due to fluid displacement or thickness noise

$$\Box^2 p'_T = \frac{\partial}{\partial t}[\rho_0 v_n \delta(f)] \tag{5}$$

The second term describes the sound generation due to acceleration of forces on the moving surface or loading noise

$$\Box^2 p'_L = -\frac{\partial}{\partial x_i}[(p-p_0)n_i\delta(f)] \tag{6}$$

The third term includes non-linearities such as shocks and turbulence effects which have the character of quadrupoles and will be neglected in this case because of the high numerical requirements. This separation of the three sound source terms is one of the biggest advantages of the FW-H equation. In this way the contribution of each source on the total sound radiation can be determined. Equations (5) and (6) can be reshaped using Green's function of the free-space wave equation, yielding a special solution of the FW-H equation for surface sources only when the surface moves at subsonic speed. This is called Formulation 1A (or 1) of Farassat and can be used to determine the sound pressure at any point in the far field [7].

4 RESULTS

Both optimization runs are calculated with a population size of 50 with 50 generations, leading to 2500 designs each. Figure 2 shows the different designs over the target functions and the resulting Pareto fronts. A particular design is at the Pareto front if it is being chosen as optimal, which means none of the objectives can be improved without sacrificing another. For both runs one design has been chosen to be investigated in more detail. These designs have been labeled with Low Noise and High Efficiency respectively.

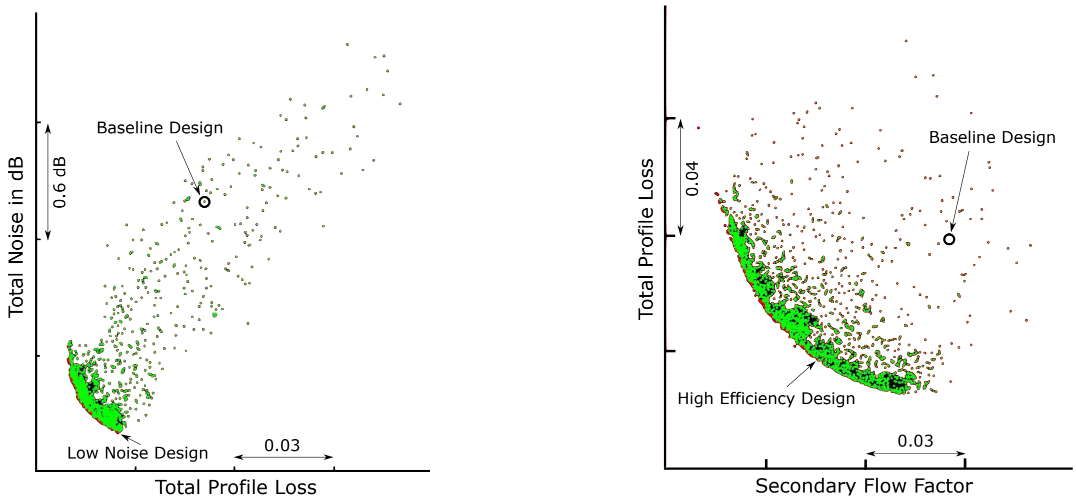

Figure 2. Different designs resulting from two independent optimization runs. The left aims for low sound radiation with the lowest possible loss in efficiency of the fan. The right aims for maximum gain in efficiency. Every point represents a blade loading distribution and a corresponding blade shape.

The low noise as well as the high efficiency design will be checked by means of stationary RANS (Reynolds-averaged Navier-Stokes equations) simulations (Ansys CFX, $k\omega$-SST model). The high efficiency design is located approximately in the middle of the Pareto front and thus represents a good compromise between secondary flow factor and total profile loss. Figure 3 shows the corresponding blade loading distributions and the resulting blade shapes. It can be clearly seen that, with an optimization with regard to the generation of noise, the blade is subjected to a greater load in the front area of the blade both on the hub and shroud section leading to a higher blade angle near the leading edge and lower blade angles especially at the hub for the rest of the blade compared to the baseline model. The optimization with the aim of high efficiency results in a higher loading in the direction towards the blade leading edge for the shroud section whereas the hub section tends to have higher loading towards the trailing edge resulting in a slightly larger blade angle at the hub in the back part of the vane. The resulting target values are summarized in table 1 in relation to the baseline model. As a measure of the sound radiation, the sound pressure level (SPL) at 500 Hz directly behind the blade leading edge is used.

Table 1. Target values of the investigated designs related to the baseline model

Model	Target Value			
	Secondary Flow Factor	Total Profile Loss	η/η_{Basis}	ΔSPL
Baseline	1	1	1	-
High Efficiency	- 31.8 %	- 2.7 %	+ 1.5 %	+ 0.7 dB
Low Noise	+ 24.4 %	- 2.4 %	+ 0.1 %	- 1.3 dB

The high efficiency design could achieve a 1.5 % higher efficiency at the design point compared to the baseline model, whereas the sound pressure level is slightly increased. The gain in efficiency might come from the greatly reduced SF factor. From figure 4 it can be seen that, for the High Efficiency design, there is less

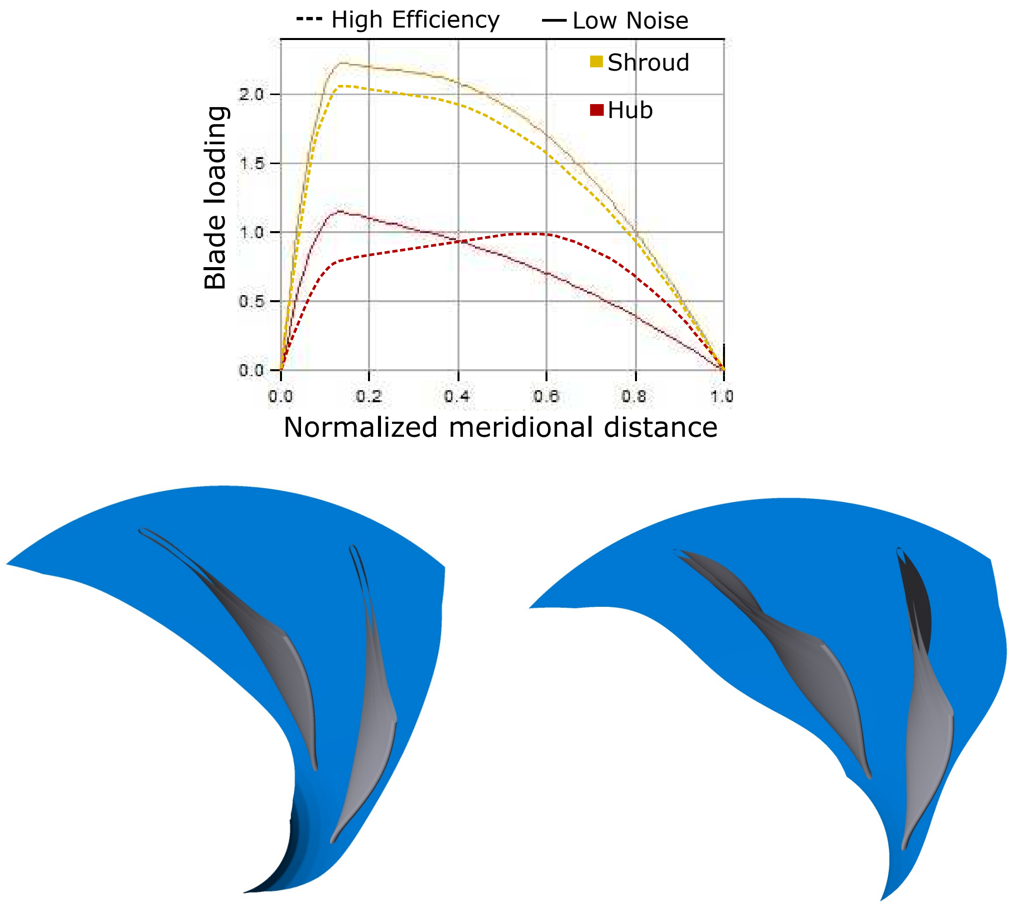

Figure 3. Blade loading distributions and corresponding blade shapes for the two optimized designs which are investigated further. Left: High Efficiency design Right: Low Noise design.

reverse flow on the blade suction side especially in the back part of the blade and the circulating region in the middle is smaller compared to the baseline model. The noise optimized design shows a more than 1 dB lower sound pressure level at the reference frequency with an almost identical efficiency compared to the baseline model. Figure 4 shows larger regions of reverse flow for this design which can be related to the increased SF factor. The almost-constant efficiency therefore results from the lower profile loss compared to the baseline design.

Figure 5 shows the difference in sound pressure level of both designs over a wider frequency range compared to the baseline model. The reduction of the sound radiation is clearly visible.

5 CONCLUSIONS

The impact of varying the blade loading on sound radiation and efficiency of a centrifugal fan has been investigated by means of Inverse Design together with multi-objective genetic algorithms in a direct approach. Varying only the blade loading, the efficiency of the fan could be improved by about 1.5 % compared to a non-optimized baseline model whereas the sound radiation was slightly increased. Another optimization led to a fan with an average of 1 dB less noise over the investigated frequency range while the efficiency could be kept constant. In further steps of the project, the objectives of high efficiency and low noise are to be combined in

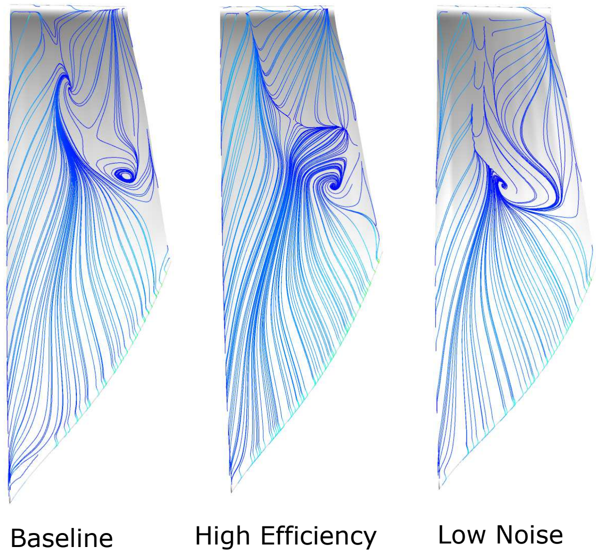

Figure 4. Surface streamlines on the suction side of the blades indicating areas of reverse flow.

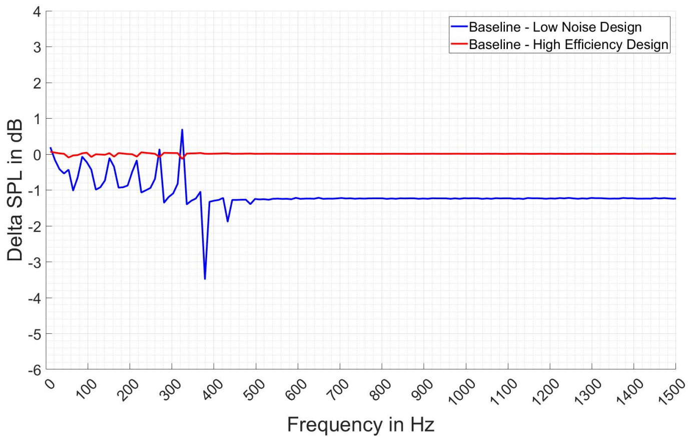

Figure 5. Difference in sound pressure level of the two investigated designs in relation to the baseline model.

one optimization and, in addition to the blade loading, further parameters such as rotational speed, number of blades or the meridional geometry are varied. Subsequently, promising designs will be investigated by means of scale-resolving simulations in combination with methods of numerical aeroacoustics [9, 10, 11] and prototypes will be investigated experimentally.

ACKNOWLEDGEMENTS

The authors would like to thank the German Federal Ministry of Education and Research, Advanced Design Technology Ltd as well as the project partners ANSYS Germany, B/S/H/ and GRONBACH for the support.

REFERENCES

[1] Eisenmenger, C.; Frank, S.; Dogan, H.; Ochmann; M. Aerodynamische und aeroakustische Untersuchungen an Radialventilatoren mit rückwärts gekrümmten Schaufeln für Haushaltsgeräte. Fortschritte der Akustik - DAGA 2018, Munich, March 19-22, 2018, pp 1208-1211.

[2] Zangeneh, M. A Compressible Three-Dimensional Design Method for Radial and Mixed Flow Turbomachinery Blades. International Journal of Numerical Methods in Fluids, Vol 13, 1991, pp 599-624.

[3] Advanced Design Technology Ltd: TURBOdesign1, 1998-2018

[4] Deb, K.; Pratap, A.; Argawal, S.; Meyarivan, T. A Fast and Elitist Multiobjective Genetic Algorithm: NSGA-II. IEEE Transactions on Evolutionary Computation, Vol 6 (2), 2002, pp 182-197.

[5] Zangeneh, M.; Goto, A.; Harada, H. On the Design Criteria for Supression of Secondary flows in Centrifugal and Mixed Flow Impellers. ASME Journal of Turbomachinery, Vol 120 (4), 1998, pp 723-735.

[6] Johnson, M. W.; Moore, J. The Influence of Flow Rate on the Wake in a Centrifugal Impeller. ASME Journal of Engineering for Gas Turbines and Power, Vol 105 (1), 1983, pp 33-39.

[7] Farassat, F. Derivation of Formulations 1 and 1A of Farassat. National Aeronautics and Space Administration, Langley Research Center, Hampton, Virginia, 2007.

[8] Brentner, K.S.; Farassat, F. Modeling Aerodynamically Generated Sound of Helicopter Rotors. Progress in Aerospace Sciences, Vol 39 (2-3), 2003, pp 83-120.

[9] Dogan, H.; Ochmann, M.; Eisenmenger, C.; Frank, S. Hybrid CFD/BEM calculations for the aeroacoustic noise radiated from a radial fan. Proceedings of Euronoise, Hersonissos, May 27-31, 2018, pp 297-302.

[10] Dogan, H.; Ochmann, M.; Eisenmenger, C.; Frank, S. A hybrid CFD/BEM method for the calculation of aeroacoustic noise from a radial fan. Fortschritte der Akustik - DAGA 2018, Munich, March 19-22, 2018, pp 493-496.

[11] Dogan, H.; Ochmann, M.; Eisenmenger, C.; Frank, S. Prediction of the aeroacoustic noise of a radial fan using Lighthill's Analogy in frequency domain. Fortschritte der Akustik - DAGA 2019, Rostock, March 18-21, 2019, pp 414-417.