Design and Optimization of Contra-Rotating Fans

Christian Friebe
ILK Dresden
Bertolt-Brecht-Allee 20
01309 Dresden, Germany
christian.friebe@ilkdresden.de

Ralph Peter Mueller
CFturbo Inc.
NEWLAB, 19 Morris Avenue
Bld. 128, Brooklyn Navy Yard
Brooklyn, NY 11205, USA
ralph-peter.mueller@cfturbo.com

Oliver Velde
CFturbo GmbH
Unterer Kreuzweg 1
01097 Dresden, Germany
oliver.velde@cfturbo.de

Abstract

Axial, contra-rotating fans can be one solution for compact, efficient, powerful air movement technology especially if the speed of each Impeller can be controlled independently. No additional bladed stator is required. Also, this concept has potential for better acoustic behavior. Nonetheless the design of contra-rotating fans is a challenging task since performance and acoustic depended on various geometric parameters as well as of their matching. Using virtual methods including automatic optimization in the design process allows a lot of different design considerations in a reasonable time that would not be possible if each design was tested in a tested rig.

Keywords

Optimization, Design of contra-rotating fans, measurements of contra-rotating fan prototypes

Nomenclature

c	absolute velocity
u	circumferential velocity
w	relative velocity
g	acceleration of gravity
l	chord length
LE	leading edge
n	rotational speed
m	meridional
Q	volume flow
p	pressure
P	Power
T	Torque
z	number of blades
Δ	any difference
λ	stagger angle
η	efficiency
ρ	density

1. Introduction

Compact high-performance machines e.g., for highly integrated electronics in the fields of IT, telecommunications, network technology and renewable energy require cool air that reliably reaches internal components which need to be cooled. For some cases conventional single impeller fans are too weak whereas radial blowers with volutes may not be applicable because of limited space, or a preferred axial-throughflow concept.

Design, simulation and optimization methods described in this paper will require parametric geometry generation for turbomachinery components including conceptual design, simulation techniques like 3D-CFD, as well as mathematical optimization methods.

It has been an introductory optimization project with some potential to vary further contra-rotating fan configurations.

2. Design configuration

Figure 1 shows the sketch of a driving system of a contra-rotating fan. Here the first Impeller is mounted on the axis whereas the second one is connected to the rotor of the driving motor. If this kind of setup is used the torque and power distribution respectively is imposed by the aerodynamic design of the Impellers.

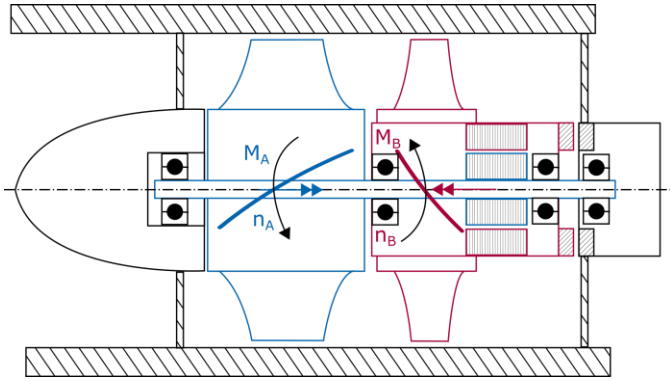

Figure 1 Gearless single-motor contra-rotating fan example

The design point for the given example consists of the pressure difference Δp, the flow Q and the speed n. The power distribution between two Impellers can be defined by either of:

- Pressure difference and speed: $\Delta p_{tot,I}$, n_I and $\Delta p_{tot,II}$, n_{II}
- Torque and speed: T_I, n_I and T_{II}, n_{II} (with $T \cdot n = \Delta p_{tot} \cdot Q$).

2.1. Design fundamentals

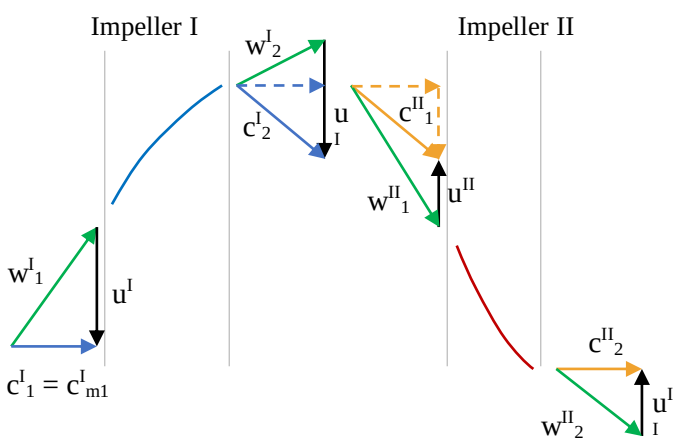

Figure 2 Velocity triangles

By principle of contra-rotating fans the first Impeller generates a negative pre-swirl for the second Impeller. Euler equation of turbomachinery reads as follows for two subsequent contra-rotating Impellers:

$$\frac{1}{\rho}\left(\Delta p_I + \Delta p_{II}\right) = c^I_{u2}\left(u^I + u^{II}\right), \qquad \text{(Eq. 1)}$$

which means that the pressure difference produced by the contra-rotating fan depends on the swirl component cIu2 generated by the first Impeller, the Impeller diameter, and the respective Impeller speeds (under the assumption of zero pre-swirl and post-swirl). This suggests a simple way to choose a combination of two different speeds to gain a certain momentum ratio imposed by the Impellers.

2.2. Impeller design process

The applied design process is modular both for each component but also within each component for different design steps [5]. While being conceptual it takes operating point and turbomachinery theory into account rather than pure geometric information.

The starting point is the definition of the design point consisting of pressure difference, flow rate and speed as well as fluid properties. For the applied design approach, the gas density is needed only since the flow can be considered as being incompressible and since the viscosity is not a design parameter within the design approach applied.

The design point is defined as shown in following table:

	fan	Impeller I	Impeller II
Flow Q [m^3/h]	540	540	540
Pressure diff. Δp [Pa]	1000°	690	310
Speed n [rpm]		8000	-5000

Table 1 Design point definition

The Impeller design is done sequentially:

- Total pressure definition as part of the overall pressure difference, speed definition if differing from design point definition, contra-rotating option applied or not
- Definition of main dimensions consisting of hub and shroud diameter
- Blade properties: setup of kinematics, definition of number of blades, definition of airfoil profiles plus stagger angle and chord length
- 3D-generation of the blades by scaling (with chord length) and staggering the chosen airfoils and wrapping them onto their respective span's radius
- Sweeping of the blades by shifting the center of gravity of each airfoil towards an acoustically beneficial position

The design process within CFturbo® is interactive. Initial settings are proposed by CFturbo® software based on latest turbomachinery design theory. To this end balance equations are solved together with the application of empirical co-relations. Often those co-relations are given with respect to the specific speed that is defined by:

$$n_q = n\left[\min^{-1}\right] \cdot \frac{Q\left[m^3/s\right]^{1/2}}{\left(\frac{\Delta p[Pa]}{\rho \cdot g}\right)^{3/4}}, \qquad \text{(Eq. 2)}$$

Informational values are provided once parameters have been chosen. They can be used to check and revise the draft of the fan during the interactive design. The input panel shows a first Impeller that shall generate 67% of the overall pressure difference. Its specific speed value is given with respect to this portion. Based on the design point and Impeller diameter, mid span estimations are provided for all velocity components.

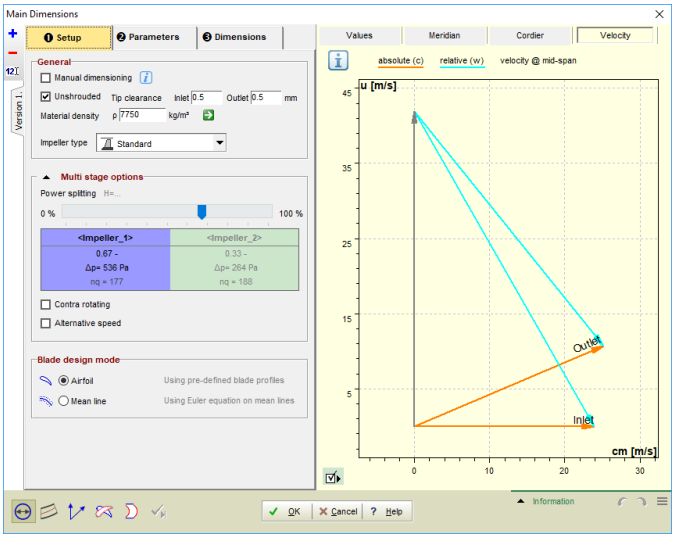

Figure 3 Main dimensions of Impeller I

Design step 3 is most important since it defines size and staggering of the blade profiles at each span which determines the power transmission in the fan. A beneficial post blade radial distribution of the absolute velocity components can be adjusted. Therefore, a radial equilibrium is solved in CFturbo considering both the design point flow rate as well as the Impeller's design point pressure difference.

Due to its high specific speed a low-pressure axial fan should be designed by the blade element momentum theory, see [6]. A blade profile was chosen with appropriate aerodynamic properties relevant for the Reynolds-number range (i.e. NACA 6508). Combining the above mentioned absolute velocity component distribution and the aerodynamic properties (lift and drag coefficient) of the airfoil within the blade element momentum theory yields suggestions for a reasonable set of stagger angle and chord length for each span of the blade.

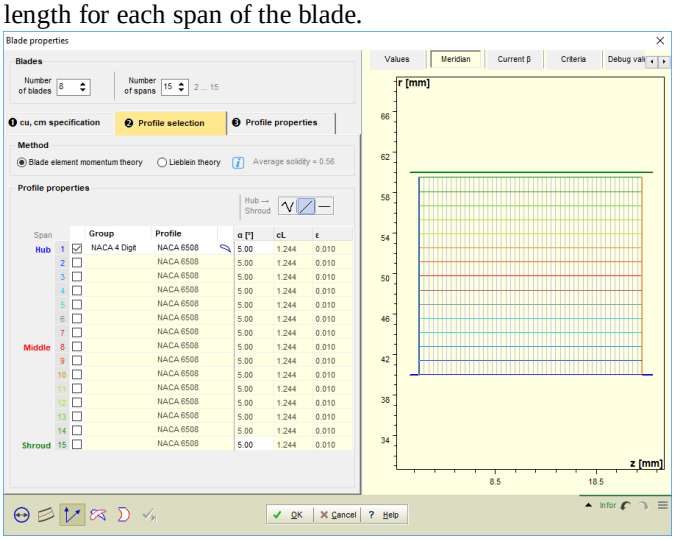

Figure 4 Design step blade properties, Impeller I

All design steps must be done for both Impellers. The complete geometric representation of the fan is parametric. This allows for any parameter change followed by an entire 3D geometry update:

2.3. Initial design

The actual design point is given in the following table:

	Impeller I	Impeller II
Number of blades z	8	6
Profile	NACA 6508	NACA 6508
Stagger angle λ [°]	46.7 .. 36.1	45.0 .. 36.3
Chord length l [mm]	32 .. 24	32 .. 26

Table 2 Initial design

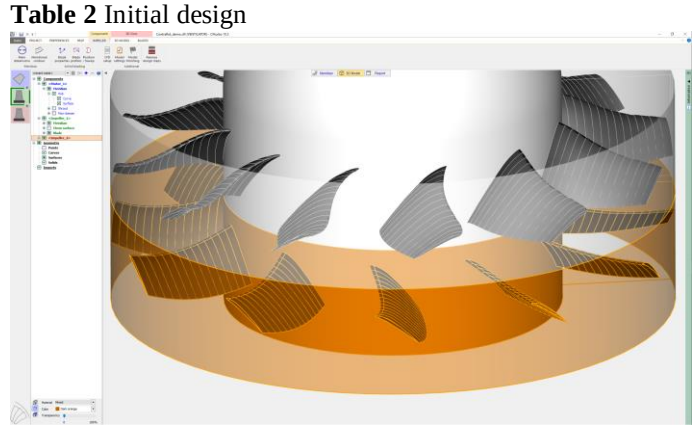

Figure 5 Contra-rotating fan configuration in CFturbo®

3. Simulation

3.1. Computational model

To validate each new fan design 3D-flow simulation (CFD) has been performed to compute integral values like pressure difference, efficiency, torque and power. This was done for a set of operation points to get characteristic curves.

Some simplifications have been performed to save computational resources especially with respect to the optimization process, see chapter 0. Those simplifications are:

- Simulation of a segment rather than of the complete 350° model, application of periodic b.c.
- Steady simulation, application of a mixing plane interface

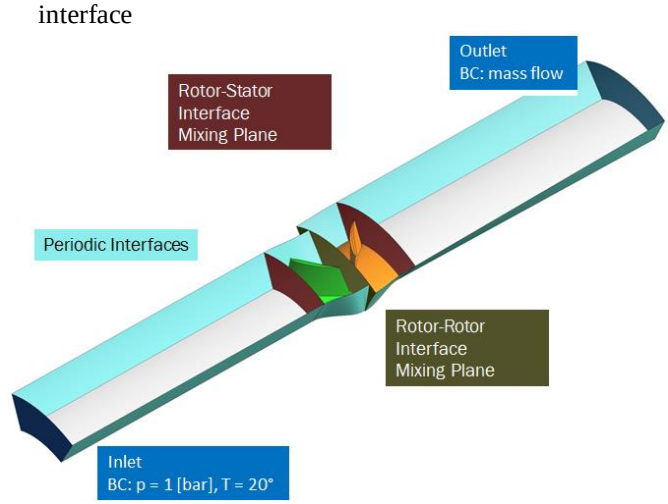

Figure 6 Computational model

For meshing and simulation ANSYS-TurboGrid and ANSYS-CFX respectively were applied. An example of a mesh is displayed in the picture below.

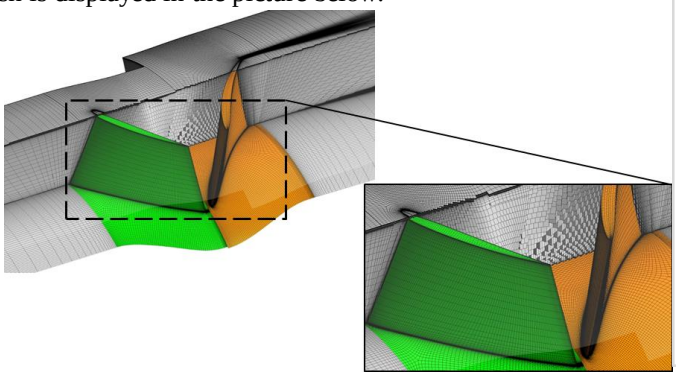

Figure 7 Mesh of the flow passage

3.2. Model setup

The following settings were used for the actual run of the simulations:

- Boundary conditions, see Figure 6
- Rotational speed, see Table 1
- Fluid: Air as perfect gas
- Steady state simulation with heat transfer
- SST turbulence model
- High resolution difference scheme

3.3. Results

Integral values such mass flow averaged pressure where calculated based on the CFD results. The same applies to the determination of the torque which is used to calculate the coupling power by:

$$P_c = T \cdot 2\pi \cdot n \,. \qquad \text{(Eq. 3)}$$

With the help of the coupling power and the mass flow averaged total pressure difference, the efficiency can be determined by:

$$\eta = \frac{\Delta p_{tot} \cdot Q}{P_c} \,. \qquad \text{(Eq. 4)}$$

The planes used for the analysis of those integral values are shown in the figure below:

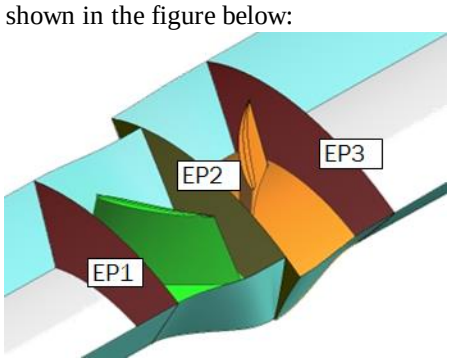

Figure 8 Evaluation planes (Control sections)

Total pressure differences and efficiencies have been calculated for a set of flow values (Q = 400 m^3/h to 650 m^3/h). The characteristics for these values are given below for the initial design:

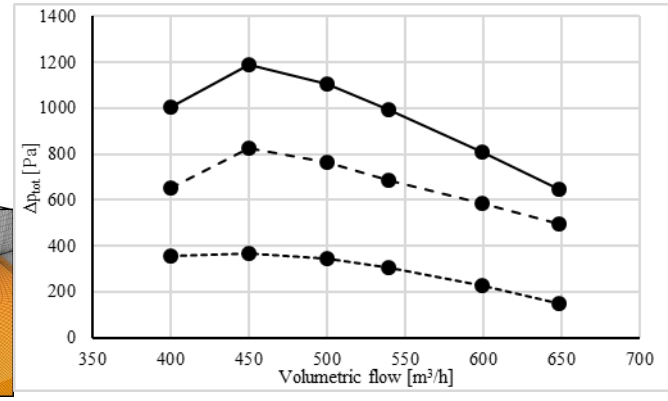

Figure 9 Total pressure difference vs. flow

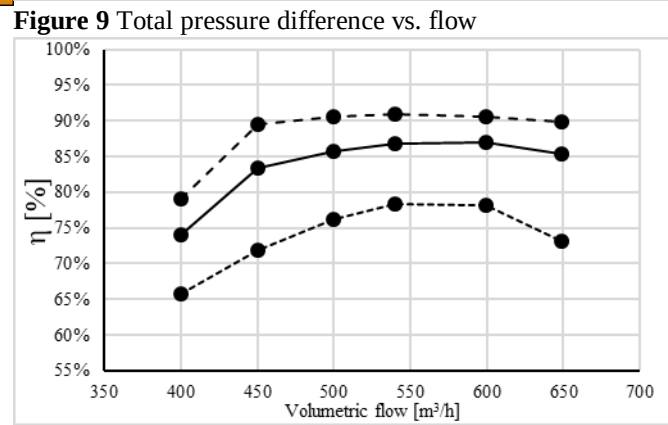

Figure 10 Efficiency vs. flow

4. Optimization

4.1. Setup

A new design method has been developed to design and validate all-new contra-rotating fans from scratch explained in chapters 2 and 3. This methodology must become part of an automatic optimization process whereas an optimization software will control geometric impeller parameters to achieve the design objective [1].

For each individual design modification done in CFturbo® there must be an ability to interact robust and smoothly with the optimizer software as well as with meshing and simulation codes. If not done well promising designs might get lost.

The optimization workflow is depicted in Figure 11 as shown below. The optimizer (HEEDS) is the central control unit which applies specific optimization strategies to modify and to adjust geometric parameters within the geometry engine (CFturbo®). Decisions made are based to the first on design objectives defined by the user. Secondly geometry changes are based on simulation results provided by the 3D-CFD-code (ANSYS-CFX).

The goal for this initial contra-rot design project was to improve the efficiency of Impeller II. The initial design of Impeller I was taken to be satisfactory due to its excellent performance and efficiency.

Figure 11 Optimization workflow

4.2. Optimization results

The initial design of Impeller I yielded a high efficiency of 91% at the Best Efficiency Point (BEP). To keep the number of geometry parameters to be changed in a reasonable range only the model of Impeller II should be modified.

Amongst those parameters are stagger angles, chord length, axial position of the leading edge, number of blades and some others:

Parameters	Range
Number of blades z	4 .. 10
Stagger angle λ_{hub}	20° .. 60°
Stagger angle λ_{tip}	20° .. 60°
Chord length l_{hub}	32 mm (const.)
Chord length l_{tip}	20 .. 32 mm
LE position Δz_{hub}	1 .. 8 mm

Table 3 Geometric parameters controlled by optimizer

The efficiency of the second Impeller could be improved by 3% increments from initially 78% to 81% at the design point (BEP). At part load the improvement was substantially higher. Measurements showed good agreement with the results provide by 3D-CFD-simulation. While the initial conceptual design for both impellers had reasonably good aerodynamic quality an excellent starting point for whole optimization process of the fan was given. Such an advantage can dramatically reduce the number of design loops in similar projects especially when the sensitivity of design changes can be anticipated before running an optimization.

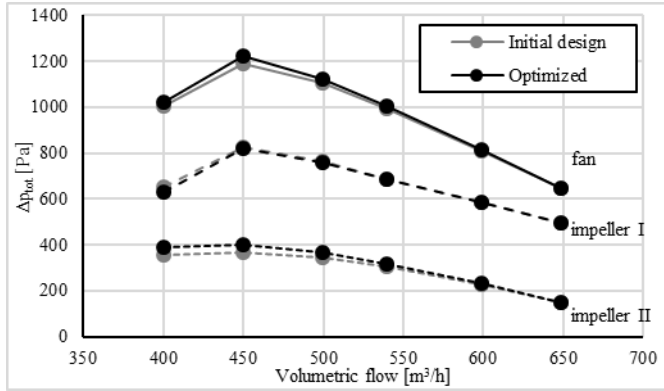

Figure 12 Pressure difference vs. flow

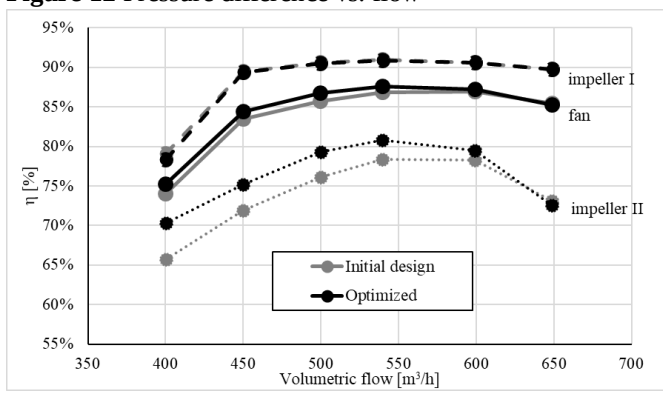

Figure 13 Efficiency versus flow

5. Measurements

Additive manufacturing technology was used to build physical prototypes of the most promising fans created in the optimized design process. Those have been tested subsequently on Aerodynamics as well as on Aeroacoustics. The measurement showed good agreements with the simulation results and proofed the concept of the optimization process applied.

6. Possible applications for contra-rot fans in IT industry

Different possibilities to use contra-rotating fans for air movement tasks in the IT-industry can be investigated. Key factors for a fan concept decision are efficiency, space, noise and total-cost-of ownership.

According to the authors contra-rotating fans could be made valuable to all lager “form factors” like data centers, cabinets, servers, racks, as well as for workstations. There is a promising possibility for independent speed control when each impeller is driven by its own electrical motor. For each application and size (form factor) different fan concepts should be developed and optimized considering aerodynamic and mechanical design, acoustics as well as manufacturing and financial constraints.

7. Conclusion and outlook

In this introductory project the combination of automated design, simulation and optimization techniques enabled very short turnaround times to get a new, efficient contra-rotating fan design from scratch. Approximately 300 designs with complex blade shapes using highly cambered and staggered

profiles were generated and numerically validated. Those blades have been swept to gain acoustic benefit.

A small group of well-designed contra-rotating fans as found during the optimization process. Some of them were chosen for prototyping and experimental investigation.

Ongoing and planned development work for different sizes of contra-rotating fans will cover performance map optimization by independent speed control, detailed acoustic investigations, rapid prototyping and manufacturing issues.

References

1. S. Stuebing, G. Kreuzfeld, R.P. Mueller, S. Marth, M. Schimmelpfennig – Initial Design and Optimization of Turbomachinery with CFturbo and optiSLang, WOST - Optimierungs- und Stochastiktage 2014, Weimar Germany, 2014
2. Ch Friebe, O. V, R. Krause, K. Hackeschmidt – Design and investigation of a multistage axial contra-rotating fan, FAN2018, Darmstadt Germany, 2018
3. DIN EN ISO 5801 – Industrial fans – Performance testing using standardized airways (ISO 5801:2007, including Cor 1:2008); German version EN ISO 5801, 2008
4. DIN EN ISO 5167-2 –Measurement of fluid flow by means of pressure differential devices inserted in circular-cross section conduits running full – Part 2 Orifice plates (ISO 5167-2: 2003; German version EN ISO 5167-2, 2008
5. R.P. Mueller, G. Kreuzfeld – Designing new compressors from scratch and compressor redesign in: CompressorTechTwo Magazin, Aug./Sept. 2011, p. 78-82
6. T. Carolus – Ventilatoren, Aerodynamischer Entwurf, Schallvorhersage, Konstruktion 3. Auflage Springer Vieweg, 2012