ORIGINAL PAPER



Experimental investigation on a lightweight, efficient, counter-rotating fan with and without boundary layer ingestion

T. Lengyel-Kampmann¹ · J. Karboujian¹ · K. Koc¹ · R. Schnell¹ · P. Winkelmann²

Received: 9 March 2023 / Revised: 12 January 2024 / Accepted: 15 January 2024 / Published online: 14 March 2024 © The Author(s) 2024

Abstract

The experimental investigations on the counter-rotating CRISPmulti fan with boundary layer ingestion are presented in this paper. The Counter Rotating Integrated Shrouded Propfan (CRISP) was designed, aerodynamically optimized and manufactured in lightweight material by the German Aerospace Center (DLR). The main task was to maximize the aerodynamic efficiency under restricted static, dynamic and acoustic properties of the blade designed with lightweight carbon-fiber material and to apply it as a demonstrator for the development of a new innovative blade manufacturing technology. The rotors were manufactured at the DLR Institute of Structures and Design and tested on the Multi-stage 2-shaft Compressor Test Facility (M2VP) at the DLR Institute of Propulsion Technology. The instrumentation enabled the measurement of the aerodynamic, static, dynamic and acoustic behavior of the fan. In general, the targets of the experimental investigation were to validate the optimization results, the numerical calculational methods and the acoustic tools. Additionally, inlet distortion measurements were carried out on the fan rig to investigate the effect of the boundary layer ingestion on a real counter-rotating propfan. A radially traversable disturbance body, designed to simulate the boundary layer of the aircraft fuselage on an aircraft architecture with embedded engines, enables a comprehensive measurement program to be carried out for both undisturbed and disturbed inflow.

Keywords Ducted fan · Counter-rotating · CFRP · BLI · Experiment · Inlet distortion

Abbreviations

ADP	Aerodynamic design point
AGATA ^{3s}	Impact of boundary layer ingestion on engine
	fans: aerodynamics, aeroelasticity, structural
	mechanics and acoustics
BL	Boundary layer
BLI	Boundary layer ingestion
BLR	Boundary layer rake
CRISP	Counter rotating integrated shrouded propfan
CFRP	Carbon fiber reinforced plastic
DLR	German aerospace center
HCS	Honeycomb straightener
ILD	Inlet distortion
IPCT	Image pattern correlation technique

T. Lengyel-Kampmann timea.lengyel@dlr.de

¹ Department of Fan and Compressor, Institute of Propulsion Technology, German Aerospace Center, Linder Höhe, 51147 Cologne, Germany

² Institute of Structures and Design, German Aerospace Center, Pfaffenwaldring 38-40, 70569 Stuttgart, Germany

M2VP	Multi-stage 2-shaft compressor test facility
SM	Stall margin
p _{tot}	Total pressure
p _s	Static pressure
R	Gas constant
γ	Specific heat ratio

1 Introduction

One of the most important goals of engine research is to reduce fuel consumption during the flight mission to achieve more climate-friendly air traffic [1]. For this reason, the concept of a counter-rotating fan or propeller has attracted the interest of the industry as early as the 1990's [2]. Preliminary studies to increase the isentropic fan efficiency were very promising [3–7]. However, the disadvantages of the concept, such as the complexity of the shaft system and the weight of the second rotor lead to an increased design effort when used in aero engines [5].

The advanced calculation methods for carbon fiber composites (CFRP), the automated optimization and design procedures and the increasing interest in an industrially usable manufacturing process for CFRP blades led to the decision in 2010 at the German Aerospace Center (DLR) to design a new blading for the CRISP-1 m rig (see Fig. 1) and to apply this rig for the new experimental investigations. Originally the test rig was developed by MTU Aero Engines in the 1980s [8]. The rig allows blade pitch control at a standstill and can thus also realize the test of rotor blades in thrust reversal position.

The design of the new blades was carried out in collaboration with the Institute of Structures and Design (DLR-BT) as part of the CRISP2 project [9]. For the design, a numerical process chain was developed, which allows the coupling of the flow simulation and the structural FEM analysis of the blades. This process chain was embedded in an automated optimization procedure (AutoOpti [10]) based on evolutionary algorithms and supported by surrogate models (kriging and neural networks). This allows the generation of aerodynamically optimized blade geometries and simultaneously fulfills all mechanical criteria. The FEM analysis focused on considering the anisotropic properties of the thermoplastic fiber composite material (CF/PEEK) [11]. The final blade geometries were approved for manufacturing through a review process in the follow-up project CRISPmulti and subsequently manufactured in Stuttgart at the DLR-BT.

Numerous studies are currently looking at engine concepts where the engines are embedded in the aircraft fuselage to further reduce the aircraft's fuel consumption [12, 13]. The associated aircraft fuselage boundary layer ingestion (BLI, see Fig. 2) leads to a multitude of interactions. Investigating these interactions of boundary layer ingestion on a real engine fan are the main objective of DLR's internal project AGATA³⁸ (Impact of boundary layer ingestion on engine fans: aerodynamics, aeroelasticity, structural mechanics and acoustics). The CRISPmulti rig was selected as the test vehicle for the inlet distortion tests on the multistage 2-shaft compressor test rig (M2VP). The advantage of the BLI concept is the reduced frontal area and thus the smaller overall and frictional drag of the engine. Furthermore, the propulsion efficiency of the engine increases due to the fact



Fig. 2 Boundary layer ingestion—Embedded engine (left) and the pressure distribution in the engine inlet with inlet distortion due to the sucked-in fuselage boundary layer (right) [14]

that the ingested fuselage boundary layer is re-energized. Nevertheless, the resulting inhomogeneous inflow into the engine is a major drawback of such engine integration. The associated consequences are being investigated in detail and on a multidisciplinary basis in the AGATA^{3s} project.

2 Fan design

The result of the design is presented below. The final blade geometry of the CRISPmulti fan in the x-r coordinate system is shown in Fig. 3. A detailed description of the design process was published in [9]. Because the blades were designed for an existing rig, some conditions from the rig were used for the design. The blade tip clearance for the design was taken as a common value (approx. 0.5% of the diameter) from real fans. The flow turning through the fan should remain as low as possible (close to 0) and the axial Mach number should not be less than 0.69 (similar to modern aircraft engines). The boundary conditions are summarized in Table 1.

Maximizing the efficiency at the design point (OP0) was defined as the optimization goal (Table 2) [9]. To



Fig. 1 CRISP-1 m shaft system



Fig. 3 Final CRISPmulti blade geometry

Table 1 Boundary conditions for the blade design

Boundary conditions			
Rig constrain			
Blade number Rotor1	10		
Blade number Rotor2	12		
Hub-to-tip ratio	0,27		
Outer diameter (m)	1		
Assumption			
Tip clearance R1 @ ADP (mm)	0,65		
Tip clearance R2 @ ADP (mm)	0,5		
Flow turning @ ADP (°)	<3		
Axial Mach number	0,69		

Table 2 Design targets

Optimization target	CRISP-1 m	CRISPmulti
Maximize is. efficiency (%)	87	93
Stall margin restriction (%)	>10	>10

Table 3 Optimization results

Results of the optimization	CRISP1m	CRISPmulti
Speed ratio R2 to R1	0,86	0,79
100% speed of R1 (RPM)	4980	5044
Fan pressure ratio ADP	1,25	1,30
Mass flow ADP (kg/s)	166	159

avoid the reduction of stall margin, a second operating point was calculated on the same speed line at near stall condition (OP1) and a parameter SM defined in Eq. 1 was restricted.

$$SM = \frac{\pi_{OP1} - \pi_{OP0}}{\pi_{OP0} - 1} \tag{1}$$

The rotational speeds and the fan pressure ratio are considered as free parameters in the optimization, i.e. the optimal values are set by the optimization.

The resulted values are compared in Table 3 with the initial values from the CRISP-1 m rig. It is noteworthy that the fan pressure ratio has been increased by the optimizer to achieve higher efficiencies. The optimal fan pressure ratio for the counter-rotating fan resulted in 1.3. Figure 4 shows the calculated fan map of the optimized CRISPmulti geometry compared with the map of the initial geometry.

The final geometry is analyzed in detail statically and dynamically using FEM simulations.



Fig. 4 Fan map of the optimized CRISPmulti geometry compared with the initial CRISP-1 m geometry



Fig. 5 Manufacturing process of the CFRP blades

3 Manufacturing and experimental analysis of the blades

3.1 Manufacturing

The designed blade geometry serves as a demonstrator for a new, automatable manufacturing technology for thermoplastic CFRP fan blades, which was developed as part of the Organo-sheet CRISP project at DLR. In contrast to the conventional "onion skin structure" with individual cuts, the possibility that the thermoplastic base material is used here to generate the aerodynamic blade geometry by forming a so-called "organic sheet" on the skeleton surface of the blades and subsequent milling (Fig. 5). The design and manufacturing process are described in detail in different publications [15, 16].

3.2 Spin test

To validate the calculation results, centrifugal tests were carried out on blades for rotor 1 and 2 to determine the static strength. Two Rotor 1 blades were spun in their original installation position, with the blade axis of rotation radially aligned. Since the test was carried out under vacuum, the bending moments from the pressure load, which compensate the bending moments from the centrifugal load on the blade root, were missing. The test was limited to 100% speed (5045 rpm), but due to the lack of compensation, the load in the critical foot area was 130% of the calculated load. The blades survived the test undamaged.

One rotor 2 blade was tested to bursting. To consider the influence of the pressure load for a more realistic test, the blade was tilted out of the plane of rotation in such a way that the bending moment at the blade root generated by the centrifugal force corresponded to the calculated combined load from pressure and centrifugal force. The corresponding tilt angle was determined by simulating the test setup. In the test, the blade reached a speed of 6027 rpm (150%) before bursting.

3.3 Shakertest

The dynamic strength was verified by shaker tests. For this purpose, 5 blades of rotor 1 and 1 blade of rotor 2 were tested on the institute's own shaker (Fig. 6). The tip amplitude as a control variable of the test was set for a first test series in such a way that the computationally permissible dynamic strain amplitude at the blade root was reached. With these amplitudes, the 6 blades were tested with 107 load cycles.

At the end of this first test series, the 1st natural frequency of rotor 1 and rotor 2 had each fallen by approx. 3.5%, the 2nd natural frequency by 10% (rotor 1) and 4% (rotor 2). The natural frequencies had stabilized at the end of the test block. The stronger drop of the 2nd natural frequency of rotor 1 can also be found in the Campbell diagram after the test (see Chapter 7.1).

In a subsequent test series (1 blade each rotor 1 and rotor 2) the amplitude was increased to 1.5 times the permissible amplitude. With these amplitudes, the blades were tested up to 70 and 80 million load cycles respectively. When these load cycles were reached, the natural frequencies had

T. Lengyel-Kampmann et al.

dropped by a further 1-2%, with stable behavior during the last 10-20 million load cycles.

4 Instrumentation

4.1 Target of the measurements

The aim of the experiment is firstly to experimentally verify the optimization result and secondly to investigate the effects that are caused on the fan due to boundary layer ingestion. For this purpose, a high-quality experimental database is to be generated, which is applied to validate the methods used. Therefore, modern measurement techniques are installed to obtain the necessary accuracy and detail of the experimental data, which are necessary for the validation of the numerical results. The following measurement techniques are used:

- p_{total}, T_{total} rakes, boundary layer rakes (BLR)
- 5-hole probe measurement
- Unsteady Pressure Sensors
- Transient hot wire anemometry (HW)
- Particle Image Velocimetry (PIV)
- Image Pattern Correlation Technique (IPCT)
- Acoustic measurement with microphones

Extensive instrumentation is required to fulfill all measurement tasks. The experimental investigations are carried out in the M2VP test bed in Cologne, at the Institute of Propulsion Technology. Figure 7 shows a schematic representation of all measurement planes with and without inlet distortion.

4.2 Inlet distortion

The boundary layer of the aircraft fuselage ingested into the engine leads to a reduction in the total pressure in front of the fan. This is generated on the test bench by a metal grid



Fig. 6 Experimental setup of the shaker experiments



Fig. 7 Instrumentation scheme

sheet (see Fig. 8). The sheet thickness is 5 mm. The design of the distorsion grid was carried out as part of the AGATA project and is described in detail in the paper [17]. Two variants were made:

- (a) Type4: triple-graded grid with an overall opening ratio of 66% (Fig. 8).
- (b) Type6: constant opening ratio of 76% (Fig. 9).

To be able to measure the various circumferential positions in the case of the non-circumferentially-symmetrical inlet distortion with all planned measurement techniques, the inlet distorsion (in the plane E0) itself can be rotated in the circumferential direction through a rotating channel. For this reason, it is sufficient to install the various measuring probes in just one circumferential position.

The distortion device was designed so that it can be traversed not only in the circumferential direction but also in the radial direction. Figure 9 shows the radial traversing device with the distortion grid. This enables the continuous and slow introduction of the distortion grid into the inflow during ongoing operation, controlled from the control room. A motor provides a maximum movement of 150 mm. The advantage of this method is the possibility of acceleration and deceleration with clean inflow and the variability of the distortion depth. If the blade vibrations are too strong, the distortion can be driven out of the channel immediately.

A honeycomb straightener is applied downstream of the inlet distortion to remove the unrealistic radial velocity components from the flow and to homogenize the flow [17]. The honeycomb thickness is 77 mm, the clear width of the honeycomb opening is 6 mm, with a wall thickness of 0.1 mm (see Fig. 8). The distortion grid with the traversing device and the honeycomb straightener were first mounted in



Fig. 8 Honeycomb and distortion grid Typ4



Fig. 9 Radial and circumferential traversable inlet distortion device with the Type 6 distortion grid

a ring channel and then used together in the circumferential traversing.

4.3 Pressure and temperature measurement

Without the application of the inlet distortion, four boundary layer rakes (BLR) were used in the measurement plane E1 to determine the mass flow (Fig. 10). The BLR consists of 41 Pitot tubes to determine the total pressure distribution in the wall boundary layer. The distance between the pitot tubes is: 1.5 mm, 3 mm, 5 mm and 9 mm depending on the distance from the wall. Hotwire probes are also used in the same plane between the boundary layer rakes to determine the flow turbulence parameters at the inlet. The same boundary layer rakes can also be placed in the plane E2b in the



Fig. 10 Boundary layer rakes (BLR) and the installation positions in plane E1 (in flow direction)



Fig. 11 Measuring cross with Pitot tubes and 3-hole probes (on the left as a CAD model and on the right mounted in a part of the inflow channel)



Fig. 12 Circumferential position of the p_{total} and T_{total} exit rakes at the E6 plane (right) and the p_{total} rake at the 220° position (left)

circumferential positions 45° ; 135° ; 225° and 315° to be able to determine the mass flow when using the honeycomb straightener.

To characterize the total pressure distribution in front of the fan due to the inlet distortion, a measuring cross with pitot tubes is used in the measurement plane E2a. The measuring cross consists of four arms (circumferential positions: 0° , 90° , 180° , 270°) with each 26 Pitot tubes and a central body that connects the 4 arms with each other (Fig. 11).

To determine the total pressure and total temperature distribution downstream of the fan, exit rakes are used in the plane E6 (Fig. 12). Two total temperature rakes, each with 12 radial measurement tubes, are used in the circumferential positions of 140° and 340° . The tubes are equipped with pt100 resistance sensors. Two total pressure rakes, each with 12 Pitot tubes and 2 three-hole probes, are used in the circumferential positions of 60° and 220° to determine the total pressure at the outlet and thus the fan pressure ratio. 1 mm holes on the casing wall at each measurement plane and along the flow channel at several axial positions are provided to measure the static pressure. All in all over 500 pressure measurement points were recorded in the entire rig via a PSI system. All measured values are displayed and saved via a LabView program during the measurement.

4.4 Special measurement technology

The measurement plane O1 shown in the inflow channel in Fig. 7 includes three large windows (266 mm x 185 mm) to ensure optical access for the Image Pattern Correlation Technique (IPCT) measurements. This method enables the measurement of blade deformation under rotating conditions without a disruptive installing process [18].

The measurement techniques, which are intended to characterize the aerodynamic properties, are installed in the rotor casing, in the planes E3 – upstream of the fan, E4 – interstage and E5 – downstream of the fan (see Fig. 7). Radially traversable probes, like a 5-hole probe or a hot-wire probe can be applied in the rotor casing at all these planes (see the example of the arrangement in Fig. 13). In all 3 planes it is possible to measure the flow velocities with Particle Image Velocimetry (PIV) through special windows. One array of unsteady static pressure sensors is installed above both rotor rows to obtain information about the shock system in the gap (Fig. 14).

The investigation of the acoustic properties of the fan both with and without inlet distortion plays a very important role in the validation. Therefore, about 200 microphones were used for the investigations.

To be able to use the different measurement techniques without mutual interference, several configurations are required (see Table 4).

4.5 Measurement technology for online vibration monitoring

In addition to the measurement techniques presented so far, some sensors for operational monitoring will be implemented on the rig. These all require a high sampling rate to be able to realize the transient phenomena on the rotating components immediately. A high-quality data acquisition system from Dewetron DEWEOrion 1624–200 with 160 channels and a maximum sampling rate of 200 kHz per channel is used for this.

4.5.1 Strain gauges

To monitor the blade vibration, strain gauges are implemented on the pressure side of the blades. The strain gauges are operating based on the change in resistance caused by the change in wire length due to stretching or compression (see Fig. 15).

The big challenge when using the strain gauges is the positioning and implementation of the blades. The four positions shown in Fig. 16 are determined using the FEM





Fig. 14 Kulite array from the outside (top left) and from the inside (right) of the channel

simulation. One blade is implemented with 2 strain gauges. Each strain gauge position is doubled, resulting in 8 strain gauges per rotor, which are distributed on 4 blades (see Fig. 17). Each strain gage used for operational monitoring is connected to the bridge circuit by 2-wire technology [19], since only the dynamic part (AC) of the measurement signal is considered for monitoring. The DC component (<10 Hz) of the signal is filtered by the telemetry. Therefore, the strain gage measurement can only be used to measure the vibration and not the static deformation. Both rotors have an 8-channel telemetry system for signal transmission from the rotating to the stationary system. On Rotor1 there is the possibility to transmit the bearing temperatures via 8 additional channels on the telemetry.

	Configura	ation						
Instrumentation	0	1	2	3	4	5	6	7
Inlet distortion	-	-	-	-	х	х	х	-
Honeycomb	-	-	-	-	х	х	х	x
E1: BLR	х	х	х	-	-	-	-	_
E1: Hot wire	-	_	х	-	-	-	-	_
E2a: p _{total} cross	-	-	-	-	-	х	-	x
O1: IPCT	-	х	-	-	-	-	-	-
E2b: BLR	-	-	-	-	-	-	-	x
E3: HW+5HP/PIV	-	_	-	-	-	-	х	_
E4: HW + 5HP / PIV	-	-	-	-	-	-	х	-
E5: HW + 5HP / PIV	-	-	-	-	-	-	х	_
A-Arrays: Microphones	-	-	-	х	х	-	х	-
B-Arrays: Kulites	-	-	-	-	-	-	х	x
FFA: Microphones	-	-	-	х	х	-	х	-
E6: pT-, TT-rakes	х	х	х	х	х	х	х	х

Table 4Configurations for themeasurements



Fig. 15 Strain gauges on the CFRP blades (left); Cable routing on the blades and on the titanium feet with the solder point



Fig. 16 Strain gauge positions for Rotor2



Fig. 17 Implemented blades of Rotor1

4.5.2 Tip clearance, tip timing

Further instrumentation for operational monitoring is implemented on the rotor casing. DLR's own tip clearance measurement system uses 4 capacitive distance sensors per rotor to monitor the tip clearance online. The installation positions in the circumferential direction for both rotors are at 45°, 135°, 225° and 315°. The axial sensor position is determined using numerical simulations by determining the position of



Fig. 18 Instrumentation of the casing for the online vibration and tip clearance monitoring $% \left({{{\mathbf{F}}_{\mathbf{F}}}_{\mathbf{F}}} \right)$

the minimum clearance height on the entire fan map. These are usually located on the trailing edge of the blades (see Fig. 18).

The entire measurement campaign is supported by the BSSM (non-contact blade vibration measurement system) from MTU Aero Engines as a measurement assignment [20]. This measurement method (also known as tip timing) allows not only the monitoring of the tip clearance height but also the blade vibrations. A total of 12 capacitive distance sensors are used for this (see Fig. 18), which are distributed on the circumference of the rotor casing in 2 axial positions (2×3 pieces) for Rotor1 and in one axial position (6 pieces) for Rotor2.

4.5.3 Accelerometer

The spectral analysis of the bearing vibrations is the most important method to monitor the rotor dynamics in operation. Therefore, as many bearing points as possible in the rig are supplied with acceleration sensors. A total of 20 vibration sensors are installed in the rig. The signal is recorded as an acceleration signal, from which the software calculates the speed signal by an integral function, which was limited during the operation. Important phenomena can be recognized from the FFT diagrams of the bearing vibrations, for example imbalance, misalignment of the couplings, misalignment of the shafts or bearing damage.

5 Commissioning, rig test without inlet distortion

After the very demanding rig assembly, the rig was installed in the test bed (see Figs. 19 and 20). The rotor casing was then mounted and aligned to ensure the even gap heights over the individual blades. The inlet channel closes the section between the nozzle and the rotor casing and carries the instrumentation upstream of the fan. In Fig. 19, the 2 electric motors (each 5 MW) are shown,



Fig. 19 M2VP test facility with the engines and oil system



 $\ensuremath{\mbox{Fig. 20}}$ Cross section with the gearbox, throttle, test object (fan) and the inlet section

which drive the two shafts (Rotor1 core shaft and Rotor2 hollow shaft) via a two-stage gearbox system. The gearbox has a total gear ratio of 4.3, which allows speeds of up to 8000 RPM to be achieved. The oil system and the compressed air pipes can be seen in the background. The oil system ensures the oil supply to the bearings in the rig and in the gearbox. The sealing air chambers of the rig are supplied with compressed air via the compressed air pipes.

During commissioning, the instrumentation and the operational monitoring was checked. The balance condition of the rotors was checked by operational balancing, whereby it was found that operational balancing was not necessary. The rig was then run up to 100% speed and the first Campbell diagrams were created using the measurement results from the strain gauges.

One task in this first phase was the measurement of the fan map with clean inflow. Boundary layer rakes are used to finally resolve the boundary layer of the flow and to determine the total inlet pressure profile. The inlet flow velocity profile was determined using the total pressure measurement of the boundary layer rakes in the E1 plane and the static pressure measurement in the casing with the following equation for the port number i:

$$Ma_{i} = \sqrt{\left(\left(\frac{p_{tot,i}}{p_{s}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right) \cdot \frac{2}{\gamma - 1}}$$
(2)

$$T_{s,i} = \frac{T_t}{1 + \frac{\gamma - 1}{2} \cdot Ma_i^2} \tag{3}$$

$$\rho_i = \frac{P_s}{R \cdot T_{s,i}} \tag{4}$$

$$a_i = \sqrt{\gamma \cdot R \cdot T_{s,i}} \tag{5}$$

$$v_i = a_i \cdot M a_i \tag{6}$$

where *R* is the gas constant and γ the specific heat ratio. The mass flow in the entire channel can be determined by the sum of the mass flow for each individual channel ring surface. The port with the smallest radius was taken for the entire inner tube of the channel.

$$\dot{m} = \sum_{i=1}^{41} v_i \cdot A_i \cdot \rho_i \tag{7}$$

The static pressure was measured about ½D downstream of the E1 plane to avoid the effect of the wakes generated by the rakes. Since the measurement plane E1 is more than 2 m upstream from the fan, the pressure losses of the inlet channel are also considered in the measured pressure ratios, so the measured pressure ratio is lower than the pressure ratio calculated with the designed fan geometry.

The exact recalculation of the measured fan map with the inlet and outlet channel, the measured tip gap heights and the inlet boundary conditions were carried out after the measurement campaign and published in [21]. The measured and calculated fan map is compared in Fig. 21.



Fig. 21 Measured and calculated fan map [21]

6 Rig tests with inlet distortion

After the successfully completed commissioning, an extensive test program was carried out (see Table 5). The first 4 measuring tasks were still planned without inlet distortion (ILD) and without honeycomb straightener (HCS). As the first measurement task, the blade deformation at 40% speed is measured using IPCT to validate the structural analysis with the CFRP material at a non-critical operating point. These validation results serve as a criterion to accelerate to the higher speeds, since the limit values for the operational monitoring of the blades are based on the simulation. Furthermore, without ILD and HCS, turbulence measurements in the inlet plane (E1) using hot-wire probes and acoustic reference measurements with a few microphones in the A1 array were also carried out [22, 23]. For further measurements with the special measurement technology, a new inlet pipe section with the inlet distortion device and with integrated honeycomb straightener was mounted in a rotating channel section (see Fig. 10).

In the following, the distortion grid is used as an inlet distortion. First, Type 6 is applied from the two manufactured distortion grid variants (Fig. 10). Since the pressure losses generated by the grid Type 6 with the opening ratio of 0.76 proved to be too low, the grid Type 4 with the overall opening ratio of 0.66 was used for the further measurement campaign. During the first use of the distortion grid, the effect on the flow is evaluated using acoustic and aero measurements at 3-speed lines (65%, 85% and 95%).

6.1 Operating point determination without boundary layer rakes

The boundary layer rakes were removed from plane E1 to install the inlet distortion. Due to the inhomogeneous

inflow the reduced mass flow could no longer be determined directly by the rakes. Therefore, the data already collected from the map measurements without inlet distortion was used to approximate the reduced mass flows without built-in boundary layer rakes. For this purpose, the static pressures measured in the channel wall of the inlet nozzle were used (see Fig. 22). First, the static pressures were scaled with atmospheric total pressure to avoid the effect of ambient conditions. Based on the measured reduced mass flows at different operating points of the measured clean fan map, 6th degree polynomial functions were generated as a function of the scaled static pressures for each 42 static pressure measurement point in the inlet nozzle (Figs. 22 and 23). The polynomial functions were created in the form

$$y_i = ax_i^6 + bx_i^5 + cx_i^4 + dx_i^3 + ex_i^2 + fx_i + g$$
(8)

where the reduced mass flow represents the criterion variable *y* and the scaled static pressure represents the predictor variable *x*.

Since we applied the method for 42 static pressure taps, it resulted 42 functions. In an ideal case, the functions



Fig. 22 Instrumentation applied for mass flow correlation

Conf	Measurement task		
Without ILD / without	HCS		
1	1. Measurement of the blade deformation at 40% speed using IPCT		
0	2. Fan map measurement		
2	3. Turbulence measurement with HW at the inlet		
3	4. Acoustics-reference measurement		
With ILD / with HCS			
4	5. Acoustic measurement to evaluate the inlet distortion		
5	6. Fan map measurement		
5	7. IPCT		
6	8. Acoustics + HW measurement + 5-hole probe + Kulites at the same time		
6	9. PIV-Measurement		
7	10. Surge limit tests, map		
5	11. Resonance tests (with inlet distortion at resonance OPs)		

 Table 5
 Test program of the

 CRISPmulti and AGATA
 measurement campaigns with

 and without inlet distortion
 the



Fig. 23 Dependency of the red. mass flow from the scaled static pressure based on measurements without ILD with BLR

should result the same mass flow values based on each measured static pressure in the nozzle, but in reality, there were some differences due to the measurement uncertainties. In order that the approximation of the reduced mass flow is representative, static pressures at different axial and radial positions were chosen. The condition for selecting the suitable static pressures was the strength of their negative correlation (anti-correlation) with the reduced mass flow:

$$r_{xy} = \frac{\sum_{i=1}^{n} (x_i - \bar{x}) (y_i - \bar{y})}{\sqrt{\sum_{i=1}^{n} (x_i - \bar{x})^2 \sum_{i=1}^{n} (y_i - \bar{y})^2}} < -96,5\%$$
(9)

21 out of 42 static pressures were selected to form the polynomial functions. To compensate for outliers from the indirect mass flow determination, the arithmetic mean of the polynomial-determined reduced mass flows was formed in the last step. In this way, the reduced mass flow could be determined indirectly using the static pressure measurement in the nozzle with the polynomial functions without installed boundary layer rakes.

Using this method, three characteristic curves are measured again as a reference without the distortion. Five operating points are determined on the basis of the characteristic curves, in which the detailed measurements with the special measurement technology (IPCT, acoustics, hot wire and PIV) will be applied (see Table 6, WL means a point on the working line and SM means a point near the expected stall margin).

When determining these operating points, it is considered that the acoustically relevant operating points (approach, cutback and end of field) should be approached. At the same time, the blocked speeds, at which the natural frequencies cross the engine orders on the Campbell diagram, should be avoided. This results in the operating points that are summarized in Table 6 and shown in Fig. 24.

 Table 6 Operating points for the detail measurements with special measurement technology

Operating point	Red. mass flow	Pressure ratio		
	kg/s	_		
65% WL (Approach)	113	1,093		
65% SM	99	1,124		
85% WL (Cutback)	140	1,19		
85% SM	134	1,21		
95% WL (End of Field)	150	1,27		



Operating points for the detailed measurements

Fig. 24 Operating points for the detailed measurements with special measurement methods



Fig. 25 Relative total pressure distribution (BLI-2) at the inlet of an embedded engine (left) [17] and measured in the inlet flow of the experiment (right)

6.2 Definition of the immersion depth

The targeted total pressure distribution at the inlet plane of the rotor was defined in [17] (see Fig. 25) by the CFD simulation of an embedded engine. This should be achieved during the test; however, the distortion grid can only approach these boundary conditions.

The inlet distortion should be strong enough at the selected operating points to be able to investigate aerodynamic and acoustic effects with acceptable blade vibrations. The procedure for finding the optimal immersion depth for the strong and weak distortion was as follows: the distortion grid shown in Fig. 9 was slowly driven into the flow with the traversing device and meanwhile the blade vibrations were monitored online. The limit values of the blade vibrations were only approached above an immersion depth of 120 mm, therefore the maximum immersion depth was set at 120 mm and declared in the measuring program as a "strong distortion". The "small distortion" was selected at half of the maximum traverse path, i.e. at an immersion depth of 70 mm. Figure 26 shows the pressure distribution measured in plane E2a with the measuring cross (Fig. 11) over the entire cross-section of the duct for the immersion depths of 70 mm and 120 mm at the operating point 85% at the working line. The measured two-dimensional pressure distributions show very good agreement with the pressure





Fig. 26 Measured inflow distortion at 70 mm and 120 mm immersion depth of the distortion grid at 85% speed WL

distributions at the engine inlet (BLI-1 and BLI-2) defined from the embedded aircraft simulation and specified for the tests (see Fig. 25). At the blade tip area, up to 7% pressure reduction is achieved with the small disturbance and up to 15% with the large disturbance. Accordingly, this distortion grid is well suited with the defined immersion depths as an inlet distortion for the fan to simulate the ingested boundary layer in the engine.

7 Evaluation of the measurement data without distortion

7.1 Validation of the Campbell diagram of rotor 1 using the BSSM measurements

To validate the calculated Campbell diagram, a synchronous evaluation of the BSSM data was carried out. With one exception, only measurements from the last three days of the test could be evaluated, as the data from the measurements without HCS in the inflow (see Chapter 6) were disturbed or no controlled slow shutdowns were carried out. The BSSM results thus document the condition at the end of the measurement campaign and do not allow a comparison of simulation and new conditions. Figure 27 shows the calculated and experimental Campbell diagram for rotor 1 with the individual measured crossings for modes 1-3 (1st and 2nd bending mode as well as torsion mode, see Fig. 28). The fitted curve for mode 2 is based on the measured values of the third and second last test day, with partly identical values on both days. On the last day of the test (lower symbols for mode 2), operating points were also reached that generated loads above those defined as permissible at the beginning. The significant drop in natural frequencies from



Fig. 27 Campbell diagram for Rotor 1



Fig. 28 Mode shapes for 1st (left) and 2nd (middle) bending mode and torsion mode (right)

the penultimate to the last test day can probably be attributed to this higher load.

Before the rotor assembly, the natural frequencies were determined for all individual blades by means of a ping test. After the end of the test campaign, this was repeated on one blade of rotor 1. The change in natural frequencies over the course of the respective test are shown in Table 7 for the rig and shaker tests as well as for the theoretical and experimental Campbell diagrams. Both the ping and shaker test and the comparison of the theoretical Campbell diagram with the one after the end of the test show a similar qualitative behavior: moderately lower natural frequencies in modes 1 and 3, more significant deviations in mode 2. It can therefore be assumed that the deviations between the experimental and theoretical Campbell diagrams are caused by fatigue of the rotor blades. Disassembly of a rotor blade after the end of the experiment showed that the adhesive layer between the CFRP rotor blade and the titanium root had partially failed, which led to a reduction in the stiffness of the blade root.

The shift in the natural frequencies must be considered in the still pending validation of the aeroelastic methodology developed in the project [20], especially for the 65% speed as

 Table 7
 Decrease in natural frequencies over the course of the experiment

Mode	Shaker (10 ⁷ load cycles)	Ping test before / after rig test	Campbell, n = 65% (theor. / exp., end of rig test)
1	-3.8%	-2.1%	-6%
2	-10%	-9.9%	-7.5 / -15%
3	-	-2.1%	-5%

this measuring point in the experiment is significantly closer to a crossing than predicted.

7.2 Analysis of the natural frequencies of the blades of Rotor 2 using the strain gauges

Based on the measurements with strain gauges on the Rotor 2, the natural frequencies of the blades could also be determined. The similarity of the natural frequencies of the blades with the calculated natural frequencies also confirm the good agreement of the blade geometries with the nominal geometry and thus a very good manufacturing accuracy.

The redundant measuring points delivered the same frequencies and amplitudes. Figure 29 shows an example of the measured Campbell diagrams. With the strain gauge position "P1" shown here, mode 5 could not be detected, but the other modes show very good agreement with the calculated frequencies, which are marked with black rectangles.

7.3 Influence of the honeycomb straightener on the flow

Without the honeycomb straightener, the BL rakes are used in plane E1. Here the p_{tot} is outside the boundary layer for all speeds at approx. 101000 Pa, i.e. the ambient pressure. In Fig. 30 the radial distributions of the p_{tot} at the inlet in the plane E1 for the diff. speeds are shown. The measured boundary layer can be compared at different speeds.

When the HCS is applied, the p_{tot} -losses through the honeycomb can be measured by the boundary layer rakes installed in plane E2b (see Fig. 31). The losses through the honeycomb are Mach number dependent. The measured loss coefficients in percent as a function of the Mach number



Fig. 29 Campbell diagram of blade No. 2 of the Rotor 2 measured using strain gages at "P1" position (on the hub) at the beginning of the measurement campaign



Fig. 30 Measured total pressure distribution at the inlet without honeycomb straightener



Fig. 31 Measured total pressure distribution at the inlet with honeycomb straightener



Fig. 32 Pressure loss through the honeycomb straightener

are plotted in Fig. 32 based on the measurement with the boundary layer rakes.

The reduction of the total pressure in front of the fan also leads to a reduction of the physical mass flow to keep the reduced mass flow constant. The honeycomb works in a similar way to an inlet throttle. Furthermore, this also reduces the static pressure in the duct accordingly, which is clearly



Fig. 33 Measured operating points at 100% speed (1–6, top left), the radial distribution of the total pressure measurement with two outlet rakes (top right) and the measured radial distributions at 100% speed without HCS (bottom)

noticeable by setting the throttle. With regard to turbulence and rotor dynamic, the honeycomb has a very positive effect. The turbulence in the inflow drops sharply and the orbit of the shaft decreases. As a result, the blades become steadier and the blade vibrations measured are significantly lower.

7.4 Evaluation of the pressure measurement

The total pressure ratio is one of the most important key parameters of the fan. To determine this value, the total pressure downstream to the fan (level E6) is measured as a radial distribution by two ptot rakes and the mean value is determined from this as an area average. The fan pressure ratio is defined by the relation of the mean p_{tot} at the outlet and at the inlet. At the outlet, two pressure rakes are used (Fig. 12), whose measured values are arithmetically averaged for the online fan map measurement. This method has proven itself well without inflow distortion, which is also confirmed by the good agreement between the values measured by the two rakes (see Fig. 33, top on the right). The radial distributions measured with the outlet rakes in the operating points 1-6 are plotted in Fig. 33 below. Point Nr.2 corresponds to the ADP and point Nr.6 is the last measured point before the expected stall. The reduction in the p_{tot} near the blade tip indicates that the profiles are no longer working in the optimum working range here. On the other hand, in the midblade area, even more pressure increase is produced by the increase in throttling, therefore the overall pressure ratio remains nearly constant with the increased back pressure.

The static pressure distribution along the axial position on the duct wall is shown in Fig. 34. The nozzle is located between the positions 6–8.5 m, in which the flow is accelerated by the reduction in cross-sectional area and the static pressure drops according to Bernoulli's equation to keep the total pressure constant. In the inlet channel (8.5–11 m) the flow speed is constant due to the constant cross-section area, the slight drop in p_{stat} is realized due to the pipe losses. The fan is located in the axial position between 11 and 12 m. Here the enthalpy of the flow is increased by the supplied energy and the static pressure increases as a result. Downstream to the fan is a diffuser, the speed decreases respectively and p_{stat} increases.

7.5 Evaluation of the temperature measurement

As already mentioned in Chapter 4.3 the fan exit temperature measurement was carried out in plane E6 (see on Fig. 12) to enable the determination of the fan efficiency. During the measurement the recovery factor of the temperature rakes were unknown, so the calibration of the rakes was necessary after the measurement to correct the measured values [21]. Then the isentropic efficiency can be calculated with Eq. 10:

$$\eta_{is} = \frac{\pi^{\frac{\gamma-1}{\gamma}} - 1}{\mathcal{T} - 1}$$
(10)

With $\pi = \frac{p_{t,outlet}}{p_{t,inlet}}, T = \frac{T_{t,outlet}}{T_{t,inlet}}$

Due to the low fan pressure ratio the temperature rise through the fan is very low (20-30 K). Therefore, the



Fig. 34 Static pressure distribution in the flow direction at the channel wall

uncertainty of the temperature measurement through the recovery factor has a significant influence on the efficiency measurement. Figures 35 and 36 show the corrected measured radial temperature distribution for the design point at 100% speed (Fig. 35) and for an operating point at 65% speed (Fig. 36) with the error bars. We can conclude, that this method is not applicable for the isentropic efficiency measurement under pressure ratios of 1.2.

The further difficulty in the temperature measurement was the sensitivity of the measurement on the flow angle downstream of the fan. The expected outflow angle at ADP is about 0° , but near the choke margin the outflow angle deviation is significant (see Fig. 37). The temperature rake was axial positioned at each operating point. This resulted in a very limited range for the reliable efficiency determination due to temperature measurement (see Fig. 38). The



Fig. 35 Uncertainties of the temperature measurement for the Design Point at 100% speed



Fig. 36 Uncertainties of the temperature measurement at 65% speed



Fig. 37 Calculated outflow angle downstream of the fan in different operating points (Choke, stall and ADP)



Fig. 38 Measured isentropic efficiency at high speed compared to the expected, calculated values

most reliable value resulted at the ADP with about 92%. This is comparable with the corresponding calculated value of 91% [21].

8 Effect of the inlet distortion on the fan

8.1 Radial traversing of the distortion grid

This chapter examines the events occurred when the distortion grid is slowly driven into the inflow. Each variable can be analyzed as the function of the immersion depth. The graphs in Fig. 39 present the results for the corrected mass flow, the total pressure ratio and the calculated power of both rotors. This analysis considers the experimental data for one operating point at 85% speed. Figure 39 shows an almost





Fig. 39 Evolution of the relative mass flow, total pressure ratio and power of both rotors during the traversing

constant mass flow during the traversing. Therefore, the inlet distortion seems to have a negligible effect on the mass flow. In spite of this, the total pressure ratio, plotted as relative to the maximum value, is reduced by about 1,5%. The distortion has a significant effect from the 40% immersion depth.

This behavior is also visible for the calculated motor power, also plotted as relative value to the maximum (1,1 MW for the rotor 1 and 0,99 MW for the rotor 2). This power is calculated by multiplying the rotational speed with the torque of the motor. In this analysis, both motor powers decreased by about 2,5% during the traversing. This highlights one of the positive effects of the BLI on the engine performance. In this case it means the decreasing of the fuel consumption due to the reduced required fan power.

Further, it was observed that the rotor axis shifts due to the change in the tip gap heights, when the disturbance is driven in. The gap heights are lower in the direction of the distortion body and larger in the opposite site. To explain this phenomenon, the static pressure on the circumference is plotted in planes E3, E4 and E5 in Fig. 40. Behind the inlet distortion, at an angle of 180°, the static pressure is strongly reduced. The relative reduction becomes lower in the interstage plane E4. Downstream of the fan the effect of the distortion at the casing wall is in the opposite direction: the static pressure becomes higher behind the distortion.

Figure 41 shows that the static pressure upstream of the fan is almost the same for both configurations. But in front of the first rotor, the static pressure without distortion is 1% higher than behind the distortion. This can be explained by a higher acceleration of the flow due to the fan behind the distortion which further decreases the static pressure in comparison with a clean inflow. The opposite behavior happens behind the second rotor. The static pressure becomes again same for both cases downstream the fan. This behavior is documented by Longley et al. [24, 25] as well.

8.2 Circumferential traversing of the distortion grid

In the second part of the evaluation study, a two-dimensional analysis of the plane E6 downstream of the fan with and without inlet distortion is carried out, which shows the important effects of the BLI. This evaluation was allowed by the circumferential traversing of the distortion grid. Since the outlet rakes can only measure the total pressure at 2



Fig. 40 Static pressure distribution on the casing wall in different planes with inlet distortion



Fig. 41 Static pressure distribution on the casing wall along the channel with and without inlet distortion

circumferential positions (60° and 220°), the distortion is traversed 360° in the circumferential direction in 18° steps to be able to record the entire flow field with both rakes. This means that both rakes go through the entire 360° area. In Figs. 42 and 43, the measured values are shown transformed in the relative system as if the distortion were fixed always at 180° (at 6 o'clock, down). This method is suitable for displaying the pressure and temperature distribution with inlet distortion downstream of the fan. Figures 42 and 43 present the measured total pressure and temperature distribution at the outlet with and without the distortion. The quantities are plotted in relative value by dividing the maximum for both cases.

This analysis was done for five different experimental operating points at three different speeds (95%, 85%, 65% of the nominal speed) marked in black in Fig. 24. Three of these points are on the working line and two others near the stall area of the speedlines 65% and 85%. In this paper, the results for the immersion depth of 120 mm are presented.

The 2D plot of the total pressure distribution shows that the distorted area downstream of the fan has almost disappeared, the blades smooth the distortion to the level that the pressure fluctuation on the circumference was reduced to 1-2%. It can be recognized that the area with slightly lower pressure is remained at 180° . The reason for this phenomenon is the counter rotation of both rotors: the front rotor turned the distortion to about 150° position, the rear rotor turns the distortion back to its original position, but the profile is more blurred as in the inlet flow. The averaged



Fig. 42 Total pressure distribution relative to the maximal pressure in the E6 plane without distortion (top) and with the "strong distortion" (bottom) at 85% speed, working line operating point

total pressure in plane E6 is slightly reduced, this leads to a decreasing of the total pressure ratio of about 0,5%.

In the next step, the effect of the distortion on the total temperature will be investigated (see Fig. 43).

The distorted area in the Fig. 43 could be recognized by the red, higher temperature area. The fluctuation of the total temperature is about 1-2% on the circumference, which has a significant effect on the efficiency. This phenomenon can be explained by the reduction of the axial velocity component behind the distortion. This leads to a positive incidence of the flow, which results increased losses of the profiles and this leads to the temperatures rise.

Based on the 2D pressure and temperature distribution a mean value can be determined by area-weighted averaging. This serves as the total pressure and temperature at the outlet when determining the pressure ratio of the fan with inlet distortion. Similarly, the total pressure at the inlet is calculated from the 2D distribution measured by the p_{tot} cross (shown



Fig. 43 Total temperature distribution relative to the maximal temperature in the E6 plane without distortion (top) and with the "strong distortion" (bottom) at 85% speed, working line operating point

in Fig. 25). This method allows the determination of the fan performance with inhomogeneous inflow.

Finally, Table 8 summarizes the deviation for the five measured points in the percentage of decreasing (in blue) and increasing (in red) compared to the clean case. Based on these results we can state, that the total pressure ratio decreases with the distortion. This reduction is lower for the lower rotational speeds. The averaged total temperature increases with the distortion. Otherwise, the conclusion of the temperature increasing is quantitatively difficult due to the measurement uncertainties. The engine power is globally

 Table 8
 Influence of the distortion on the total pressure, temperature and motor power

On constinue Desint	Total pressure	Tatal townshine	Power		
Operating Point	ratio	Total temperature	Motor 1	Motor 2	
95% RPM	0,07% 🛸	0,14% 🦊	0,93% 🦊	0,66% 🦊	
85% RPM Working-line	0,47% 🔦	0,28% 🦊	2,93% 🛬	0,34% 🐋	
85% RPM Stall Area	0,27% 🔦	0,34% 🦊	1,76% 🐋	0,46% 🐋	
65% RPM Working-line	0,04% 💊	0,41% 🦊	1,02% 🔦	0,54% 🦊	
65% RPM Stall Area	0,09% 🐋	0,36% 🦊	0,61% 🖕	0,23% 🖕	



Fig. 44 Effect of the distortion on the fan performance

decreased, especially the motor 1 which drives the second rotor. The results for the operating point with a speed of 95% are not comparable, since the immersion depth of the inlet distortion is lower than for other points.

The Fig. 44 concludes the distortion effects on the fan performance. The differences between with and without distortion are not significant, because the distortion is relatively slight. For all points, the same trend can be noticed. The decreasing in the total pressure ratio and an increasing in the corrected mass flow was realized. The effect of the distortion is lower for the speed of 65% than for the 85% speed.

9 Summary

The experimental investigations on a counter-rotating fan with an inlet distortion served as a boundary layer ingestion were presented here. The aim of this work is to show the experimental setup, the instrumentation and the measurement process. This confirmed that the project goals of the CRISPmulti and AGATA projects could be achieved. An extensive experimental database was generated for the validation of the numerical methods, which can be evaluated in future works. The impact of the inlet distortion on the fan can be analyzed in detail using the data. Both the aerodynamic phenomena and the mechanical and aeroelastic properties of the blades can be evaluated using the measurement data.

The main results of this study can be summarized as follows:

- the development of the method how to measure and evaluate the fan performance with non-symmetric inflow
- the validation of the natural frequencies of the CFRP blades
- to investigate the aerodynamical effects of the distortion on the p_{tot}—and T_{tot}—distribution in the outlet plane

We can conclude, that the applied inlet distortion has no significant effect on the fan performance (mass flow and total pressure ratio), but it has a strong influence on the total temperature and with this on the isentropic efficiency of the fan.

Funding Open Access funding enabled and organized by Projekt DEAL. The authors did not receive support from any organization for the submitted work. The experiments were carried out in the frame of the DLR intern projects CRISPmulti and AGATA^{3S.}

Declarations

Conflict of interest The authors declare no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

Open Access This article is licensed under a Creative Commons Attribution 4.0 International License, which permits use, sharing, adaptation, distribution and reproduction in any medium or format, as long as you give appropriate credit to the original author(s) and the source, provide a link to the Creative Commons licence, and indicate if changes were made. The images or other third party material in this article are included in the article's Creative Commons licence, unless indicated otherwise in a credit line to the material. If material is not included in the article's Creative Commons licence and your intended use is not permitted by statutory regulation or exceeds the permitted use, you will need to obtain permission directly from the copyright holder. To view a copy of this licence, visit http://creativecommons.org/licenses/by/4.0/.

References

- European Commission.: Directorate general for research and innovation and European Commission. Directorate general for mobility and transport. Flightpath 2050: Europe vision for aviation: maintaining global leadership and serving society needs. Publications Office, LU (2011)
- Schimming, P.: Counter rotating fans an aircraft propulsion for the future? J. Therm. Sci. 12(2), 97–103 (2003). https://doi.org/ 10.1007/s11630-003-0049-1
- Talbotec, J., and Vernet, M.: SNECMA counter rotating fan aerodynamic design logic & tests results., ICAS 2010, pp. 10, Nice, France (2010)
- Lengyel, T., Nicke, E., Rüd, K.-P., Schaber, R.: Optimization and examination of a counter-rotating fan stage: the possible improvement of the efficiency compared with a single rotating fan. In: ISABE: pp. 11. Gothenburg, Sweden (2011)
- Lengyel-Kampmann, T., Otten, T., Schmidt, T., Nicke, E.: Optimization of an engine with a gear driven counter rotating fan Part I: Fan performance and design. In: Proceedings of the 22nd International Symposium on Air Breathing Engines, pp. 25–30. Phoenix, AZ, USA (2015)
- Druzhinin, I., Rossikhin, A., Mileshin, V.: Computational investigation of aerodynamic and acoustic characteristics of counterrotating fan with ultra high bypass ratio. In: European Conference on Turbomachinery Fluid Dynamics and Thermodynamics. Stockholm, Sweden (2017)
- Meillard, L., Stanica, C.M., Nasr, N.B., Riéra, W.: Design of a counter rotating fan using a multidisciplinary and multifidelity

optimisation under high level of restrictions. In: ISABE 2017. Manchester, UK (2017)

- Sieber, J.: Aerodynamic Design and experimental verification of an advanced counter-rotating fan for UHB engines. Third European Propulsion Forum, Paris (1991)
- Goerke, D., Le Denmat A-L., Schmidt, T., Kocian, F. and Nicke, E.: Aerodynamic and mechanical optimization of CF/PEEK blades of a counter rotating fan. ASME Turbo Expo 2012 Copenhagen, Denmark GT2012–68797 (2012)
- Aulich, M., and Siller, U.: High-Dimensional constrained multiobjective optimization of a fan stage. Vancouver, British Columbia, Canada (2011)
- Aulich, A.-L., Goerke, D., Blocher, M., Nicke, E., and Kocian, F.: Multidisciplinary automated optimization strategy on a counter rotating fan. San Antonio, Texas, USA (2013)
- Castillo Pardo, Alejandro and Hall, Cesare A. "Aerodynamics of boundary layer ingesting fuselage fans." J Turbomachin 143, ISSN: 0889–504X, https://doi.org/10.1115/1.4049918
- Mennicken, M., Schoenweitz, D., Schnoes, M. and Schnell, R. "Fan design assessment for BLI propulsion systems."CEAS Aeronaut J 13, ISSN: 1869–5582, 1869–5590, https://doi.org/10.1007/ s13272-021-00532-8
- Vinz, A., Raichle, A., "Investigation of the effects of BLI engine integration on aircraft thrust requirement", DLRK Konferenz at Dresden (2022)
- Schmid, T., Lengyel-Kampmann, T., Schmidt, T., Nicke, E.: Optimization of a carbon-fiber composite blade of a counter-rotating fan for aircraft engines. In: European Conference on Turbomachinery Fluid Dynamics and Thermodynamics, Lausanne, Switzerland (2019)
- Forsthofer, N., and Reiber, C.: Structural mechanic and aeroelastic approach for design and simulation of CFRP fan blades. DLRK2016. https://elib.dlr.de/109299/1/DLRK2016_CRISP_ Final.pdf
- Kajasa, B. und Lengyel-Kampmann, T. und Meyer, R.: Numerical and experimental design of a radial displaceable inlet distortion device. In: 25th ISABE. International Society of Air Breathing Engines, 25th Conference, 25.-30. Sep. 2022, Ottawa, Kanada. https://elib.dlr.de/191751/

- Willert, C., Klinner, J., Schroll, M. and Lengyel-Kampmann, T.: Measurement of aerodynamically induced blade distortion on a shrouded counter-rotating prop-fan. 20th International Symposium on Application of Laser and Imaging Techniques to Fluid Mechanics, 11–14. Jul. 2022, Lisbon, Portugal. ISBN 978-989-53637-0-4.
- 19. Hoffmann, K.: An Introduction to stress analysis using strain gauges, HBK Germany
- Zielinski, M., Ziller, G.: Noncontact vibration measurements on compressor rotor blades (2000). https://doi.org/10.1088/0957-0233/11/7/301
- Lengyel-Kampmann, T., Karboujian, J., Charroin, G., Winkelmann, P.: Experimental investigation of an efficient and lightweight designed counter-rotating shrouded fan Stage, ETC-2023 Budapest, Hungary (2023)
- 22. Diouf, M., Lengyel-Kampmann, T., and Schnell, R. Interaction of an aircraft fuselage boundary layer with a contra-rotating turbofan." (2018) https://doi.org/10.25967/480125.
- Eichner, F., Belz, J., Winkelmann, P., Schnell, R. and Lengyel-Kampmann, T. "Prediction of aerodynamically induced fan blade vibration due to boundary layer ingestion." 13th European Conference on Turbomachinery and Fluid Dynamics, ETC2019-370 (2019)
- 24. Longley, J.P., Greitzer, E.M.: Inlet distortion effects in aircraft propulsion system integration. Cambridge University, England (1992)
- Gunn, Ewan J., Tooze, Sarah E., Hall, Cesare A. and Colin, Yann. "An experimental study of loss sources in a fan operating with continuous inlet stagnation pressure distortion." J Turbomachin 135(5), ISSN: 0889–504X, 1528–8900, https://doi.org/10.1115/1. 4007835

Publisher's Note Springer Nature remains neutral with regard to jurisdictional claims in published maps and institutional affiliations.