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UHBR Open-Test-Case Fan ECL5/CATANA

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Abstract: Application of composite fans enables disruptive design possibilities but increases sensitivity to multi-physical resonance between aerodynamic, structure dynamic and acoustic phenomena. As a result, aeroelastic problems increasingly set the stability limit. Test cases of representative geometries without industrial restrictions are a key element of an open scientific culture but currently non-existent in the turbomachinery community. In order to provide a multi-physical validation benchmark representative of near future UHBR fan concepts, the open test case fan stage ECL5 has been developed at Ecole Centrale de Lyon. Design intention was to develop a geometry with high efficiency and a wide stability range that can be realized using carbon fibre composites. This publication aims to introduce the final test case which is currently fabricated and will be experimentally tested. The fan blades are composed of a laminate made of unidirectional carbon fibres and epoxy composite plies. Their structural properties and the ply orientations are presented. To characterise the test case, details are given on aerodynamic design of the whole stage, structure dynamics of the fan and aeroelastic stability of the fan. These are obtained with a state-of-art industrial design process: static and modal FEM, RANS and LRANS simulations. Aerodynamic analysis focuses on performance and shows critical flow structures like tip leakage flow, radial flow migration and flow separations. Mechanical modes of the fan are described and discussed in the context of aeroelastic interactions. Their frequency distribution is validated in terms of resonance risk with respect to synchronous vibration. Aeroelastic stability of the fan is evaluated at representative operating points with a systematic approach. Potential instabilities are observed far from the operating line, not compromising experimental campaigns.

Keywords: Fan; Composite; Aeroelasticity; Flutter; Aeroacoustics

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1. Introduction

With application of modern lightweight blade geometries, aeroelastic coupling phenomena tend to reduce the operating range of compressors and fans in aircraft propulsion engines. As synchronous excitation mechanisms are well understood today, the most challenging aeroelastic phenomena for future turbomachinery applications are of non-synchronous¹ nature. A list of these phenomena is presented hereafter: **Flutter** is defined as self-excited blade vibration and usually involves blade-to-blade coupling. It is considered as initially small amplitude blade oscillation in a specific eigenmode that exponentially amplifies through a positive feedback loop with the aerodynamic field (negative aerodynamic damping). Linear modelling approaches are capable to predict the onset of this mechanism and to determine critical modes and to develop countermeasures. The disturbance is only dependent on the blade vibration and disappears as soon as the vibration stops. For large vibration amplitudes limit-cycle-oscillation may occur due to aerodynamic or structure dynamic non-linearity. For

¹ not a multiple of the shaft rotation speed

36 turbo-engine fans critical modes are typically dependent on swirling acoustic modes
37 that develop between the inlet and the fan stage [1].

38 **Buffeting** describes the interaction between an aerodynamic instability (typically
39 vortex shedding) that comprises a characteristic frequency and blade vibration in a spe-
40 cific eigenmode. Typically, no circumferential blade-to-blade coupling is necessary but
41 often planar acoustic duct modes establish and synchronize the phase of the aerodynamic
42 instabilities leading to a zero-nodal-diameter vibration [2].

43 **Rotating stall** is a purely aerodynamic phenomenon occurring due to overloading
44 of a blade row and subsequent flow separation leading to the establishment of circumfer-
45 entially propagating stall-cells. It typically excites structural eigenmodes due to unsteady
46 loading but for small amplitudes the propagation is not coupled with the blade vibration
47 [3].

48 **Non-synchronous forced response:** in multistage compressors, a non-synchronous
49 coupling mechanism between a trapped acoustic mode, coincident with a structural
50 vibration pattern has been observed [4]. The phenomenon is less relevant for fan
51 applications.

52 **Convective Non-Synchronous-Vibration (NSV):** Typically occurring close to the
53 stability limit but before the onset of rotating stall, a complex lock-in mechanism between
54 propagating aerodynamic vorticity disturbances and blade vibration is described under
55 the term NSV [5,6]. Blade vibration leads to the change of free vorticity or even the
56 formation of radial vortices in the passage flow which are convected from blade to blade
57 if sufficient blockage is present in the passage. Interaction with trailing blades leads
58 to modal forcing and the shedding of subsequent vorticity. Propagating aerodynamic
59 disturbances can lock-in with structural vibration patterns and lead to a coherent fluid-
60 structural interaction. The phenomenon needs to be differentiated from flutter as the
61 aerodynamic disturbance appears without blade vibration and has a characteristic
62 convective propagation speed (typically described under the term rotating-instability).

63 For all of the described phenomena, the flow structure in the tip region and particu-
64 larly the influence of the tip leakage flow and the passage shock is of great importance.
65 Significant blockage enables the circumferential transport of disturbances or provokes
66 separation of the boundary layer which may be susceptible to acoustic or structure-
67 dynamic feedback.

68 For high-speed fans rotating stall, buffeting and flutter are the most common
69 instability mechanisms and are well understood today. The establishment of UHBR-
70 configurations with low-speed fans however leads to a substantial change of relevant
71 characteristics.

- 72 • Low-speed fans will predominantly operate on the flat part of the compression
73 characteristic, making them more susceptible for stall-driven instability [7].
- 74 • The flutter frequencies (in the stationary frame) will be lower compared to high-
75 speed designs. Acoustic liners in the intake, which are designed to attenuate higher
76 frequency community noise will not affect the modes relevant for aero-elastic
77 instability.
- 78 • The intake length will be shorter for low-speed fans, leading to stronger inflow
79 asymmetry and altered acoustic interaction [8]. This gives rise to stronger broad-
80 band excitation and shifted resonance frequencies.
- 81 • The relative Mach number and shock strength are lower, the tip-clearance relative
82 to the blade chord and solidity² are smaller than for conventional direct-drive fans
83 and more sensitive to geometric variability [9].
- 84 • A strongly nonlinear fluid-structure-interaction at low-frequencies has been ob-
85 served for fans with low solidity related to the pressure untwist of the blades. At
86 transonic conditions, slight deviations of the local stagger angle at the blade tip
87 can cause a fundamentally different shock structure between adjacent blades that

² solidity = blade chord length / pitch

88 affects the stability of distinct rotor sections [10]. This circumstance affects the ap-
 89 plicability of promising methods like intentional blade mistuning [11] to suppress
 90 the development of circumferentially propagating modes.

91 A selection of the state of the art High-Bypass-Ratio engines is presented in Figure
 92 1³. The propulsion industry has focused on different strategies but all showing common
 93 goals: the reduction of tip speed, the reduction of number of blades and the reduction of
 94 individual blade weight. General Electric started in the 1990s with the establishment of
 95 layered carbon fibre composite fan blades in the GE90 engine for wide body aircrafts
 96 and is the only manufacturer with in-service composite fans. The more recent GENx
 97 shows a reduced blade number of 18. Following this development, the CFM LEAP
 98 engine has been established for the A320neo class, with 18 blades constructed from
 99 3d-woven carbon fibres. Pratt&Whitney introduced the first Geared Turbofan with
 100 20 (hybrid)-metallic blades, also for the A320neo class. Still following a conventional
 101 approach in fan design recent Rolls-Royce engines still comprise hollow titanium blades
 102 but technology demonstrators based on variants of the Trent1000 engine incorporate
 103 composite fibre fan blades. Exemplarily shown in Figure 1 is the fan of the Trent XWB
 104 engine for the Airbus A350 high bypass engine with 22 blades (Trent 7000 has 20 blades).

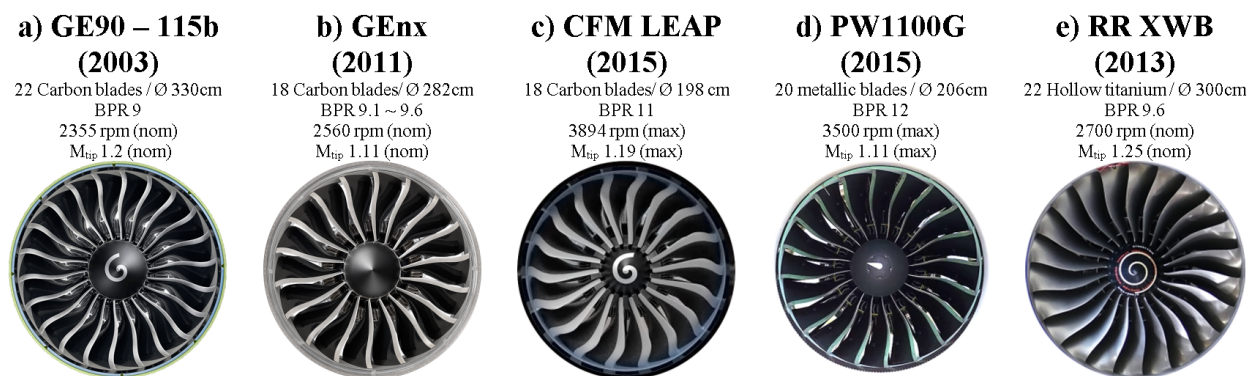


Figure 1. Selection of established modern (Ultra)-High-Bypass-Ratio Engines

105 All of the shown fan blades comprise significant 3D-features, particularly forward
 106 sweep which is known to be beneficial for aerodynamic performance but emphasize
 107 aeroelastic sensitivity for torsional or chordwise bending modes and particularly NSV.
 108 Recent research [12,13] showed that the use of anisotropic properties of composite in
 109 fans could help to improve both mechanics and aerodynamics. Composites also present
 110 potential to control flutter by modification of eigenmodes [14].

111 In order to enable further technological advancements into this direction, extensive
 112 research is necessary to identify and characterize the relevant instability mechanisms for
 113 the novel type of low-speed fans. Particularly the complex flow structure at part load
 114 and part speed is challenging for state-of-the-art numerical approaches and requires
 115 experimental benchmark data on representative geometries.

116 To address these research objectives, an extensive research program in collaboration
 117 between Ecole Centrale de Lyon and Von Karman Institute for Fluid Dynamics has
 118 been initiated. A fan module, ECL5, fabricated from composite material has been
 119 developed as an open test case. It will be investigated with a focus on non-synchronous
 120 coupling mechanisms between aerodynamics, acoustics and structure dynamics within
 121 the European CleanSky 2 project CATANA (Composite Aeroelastics and Aeroacoustics,
 122 catana.ec-lyon.fr). The fan stage has been designed at Ecole Centrale de Lyon with the

³ image sources a) <https://www.geaviation.com/commercial/engines/ge90-engine>, b) <https://www.geaviation.com/commercial/engines/genx-engine> <https://c-fan.com/products/>, c) <https://www.cfmaeroengines.com/engines/leap/>, d) <https://www.flugrevue.de/flugzeugbau/getriebefan-triebwerke-pratt-whitney-will-probleme-beseitigen/>, e) <https://www.airlinerratings.com/news/a350s-trent-xwb-engine-hits-twin-milestones/>

123 intention to be representative of near-future composite low-speed fans in the following
 124 terms:

- 125 1. General aerodynamic design parameters (Mach number, blade loading, solidity,
 126 aspect ratio, hub-to-tip ratio, mass flow density, etc.)
- 127 2. Aerodynamic flow structure due to its influence on instability mechanisms (shock
 128 patterns, radial flow migration, secondary flow, separations, etc.)

129 From the current point of view, the most promising research configuration is a
 130 UHBR low-speed fan stage with high subsonic design speed, designated to be installed
 131 in midrange jets (like Airbus A320neo). Hence, the open test case ECL5 that will be
 132 introduced in the following has been developed from an extrapolation of industrial
 133 concepts for current low-speed designs. In Figure 2⁴, the final design is shown in
 134 comparison to the recently certified GE9x and the Rolls Royce UltraFan technology
 135 demonstrator.



Figure 2. Modern industrial low-speed composite fans and Open-Test-Case ECL5/CATANA (CAD)

136 In the following, the final design and the composite structure of the fan are pre-
 137 sented. Details are given on the modelling strategies used to simulate the different
 138 physics : stationary flow simulations (RANS), mechanical simulations (FEM) and time-
 139 linearized simulations (LRANS). Finally, aerodynamic behaviour and structural dynam-
 140 ics of the fan are deduced and its aeroelastic stability is assessed.

141 2. ECL5 Description

142 2.1. Design approach

143 Based on an initial design [15], the final design was obtained with the use of a
 144 modularly coupled design chain, including a parametric blade geometry generator,
 145 automatic meshing, a steady RANS solver, and structural FEM simulations. The fan
 146 geometry fulfils the following design objectives:

- 147 • Blade number of 16 (lowest blade count in technology demonstrators as shown
 148 above).
- 149 • Fan diameter 508 mm to enable integration into test facility PHARE-2, blade root
 150 compatible to existing disk from Project ENOVAL [16].
- 151 • (Fan-only) Aerodynamic design point at peak efficiency with a pressure ratio of
 152 1.36, mass flow density maximum of 200 kg/s/m² and a rotation speed of 11000
 153 rpm.
- 154 • Near sonic relative tip Mach number at design point.
- 155 • (Fan-only) Isentropic efficiency exceeding 94% at design point.
- 156 • At peak pressure of design speed no flow separation at the trailing edge.
- 157 • Peak efficiency at 105% speed not lower than 92% (transonic speedline).

⁴ image sources: a) <https://www.additivemanufacturing.media/mission-critical/>, b) <https://www.air-cosmos.com/article/rolls-royce-tests-ultrafan-low-pressure-system-592> ; image flipped

- 158 • Nominal tip clearance ~ 1 mm (1.1% tip chord) to ensure stall inception and surge
159 experiments without casing contact, future experiments are planned with further
160 reduced tip clearance.
161 • The fan will be fabricated without integration of a metallic leading edge.

162 For the OGV, no specific design criteria were set except for a minimization of
163 (numerical) corner separation at highly loaded conditions and minimum losses at design
164 point. The blade number was fixed to 32 in the optimization process and reduced to 31
165 for the final design to avoid the establishment of planar acoustic interactions.

166 Stage inlet and outlet geometry is similar to previous installations [16]. Same applies
167 to the casing geometry in the fan section. The OGV is located far downstream of the
168 fan to minimize interactions as the focus of the research project lies on the fan. As the
169 facility comprises only a single flow channel, engine representative OGV aerodynamics
170 are not intended, merely axial stage outflow and homogeneous radial conditions are
171 required to ensure detailed performance analysis and to enable future research projects
172 with fundamental background on noise generation.

173 The development process is described in detail in Pagès [17]. The hubline from
174 nosecone to OGV outlet was open for optimization. A section of the design is presented in
175 Figure 3. Figure 4 shows the blade profiles at different span heights, clearly emphasizing
176 the transonic blade design at the fan tip and the wide chord geometry around midspan.
177 At the hub, the relative tangential velocity and the rotational speed have the same
178 direction at the trailing edge, which is necessary to meet the design criteria laid out
179 above but creates a challenging flow field with high swirl velocities upstream the OGV.

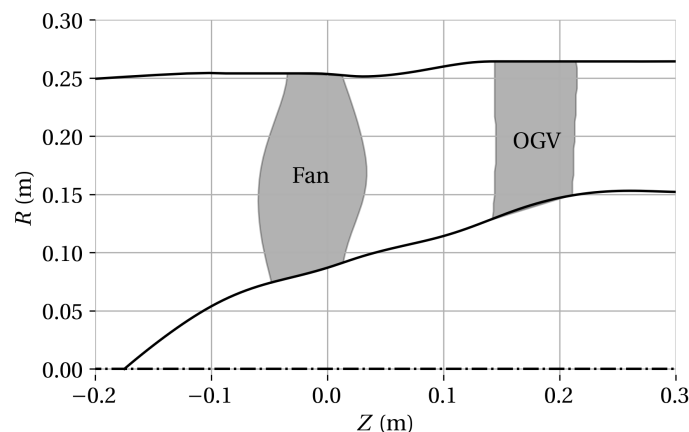


Figure 3. Section of ECL5 fan stage

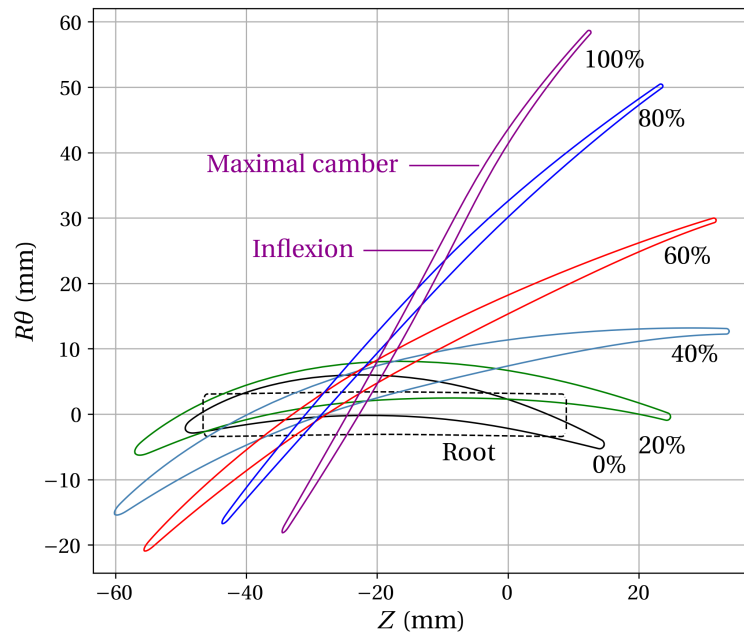


Figure 4. Fan profiles at different span heights

180 The tip clearance in dependence of the rotation speed is presented in Table 1:

Table 1. Tip clearance as predicted by FEM-Model (tip chord = 90 mm)

Rotation speed (100% = 11000 rpm)	Tip Clearance (mm)		Tip Clearance (% tip chord)	
	Leading edge	Trailing edge	Leading edge	Trailing edge
0%	1.00	1.50	1.11	1.67
50%	0.98	1.30	1.09	1.45
80%	0.87	1.12	0.97	1.25
100%	0.78	0.99	0.87	1.10
105%	0.77	0.92	0.86	1.02

181 2.2. Fan composite structure

182 The blade is exclusively made of unidirectional carbon fibres and epoxy composite
 183 plies stacked into a laminate with intentional ply orientations . A ply is considered to be
 184 made of orthotropic material. The three material orthotropy directions are defined locally
 185 as follows : direction 1 is the orientation of the carbon fibres, direction 2 is orthogonal
 186 to direction 1 within the ply and direction 3 is orthogonal to the ply. The constitutive
 187 law of the composite material is described classically by nine constants listed in Table 2
 188 (indicated by composite manufacturer).

Table 2. Material properties

ρ	E_1	$E_2 = E_3$	G_{23}	$G_{12} = G_{13}$	ν_{23}	$\nu_{12} = \nu_{13}$
1560 kg/m ³	123.0 GPa	9.3 GPa	5.0 GPa	4.5 GPa	0.3	0.26

189 The maximum thickness of the blade (at the root) is 13.3 mm. The thickness of a ply
 190 is 0.15 mm so the maximum number of plies is 88. The internal structure is symmetric
 191 with respect to the laminate midplane to minimize the coupling of in-plane and out-
 192 of-plane movements. Each point of the midplane is associated with a number of plies
 193 defined by the blade local thickness. Plies are ranked by their size from largest to smallest
 194 (see Figure 5).

195 For the design process, plies have been divided in two families. The 18 largest plies
 196 are called blade plies and the others - the smallest - are called core plies. To reduce the

197 number of degrees of freedom for the design, orientation of core plies is set considering
 198 half a blade to the sequence $((0^\circ/90^\circ)_6(0^\circ/90^\circ/0^\circ)_4(0^\circ/0^\circ))_s$, with 0° defined as the
 199 radial direction. Their influence on global elastic properties is minor and they mainly
 200 contribute in thickness adjustment. Thus the total number of fibre orientations to define
 201 is set to 18. These orientations are listed in Table 3. The orientation of a ply is defined as
 202 illustrated in Figure 5b. The orientation is imposed on a chosen point at the root of the
 203 blade. Local orientation along the blade is determined by the draping of the surface [18].

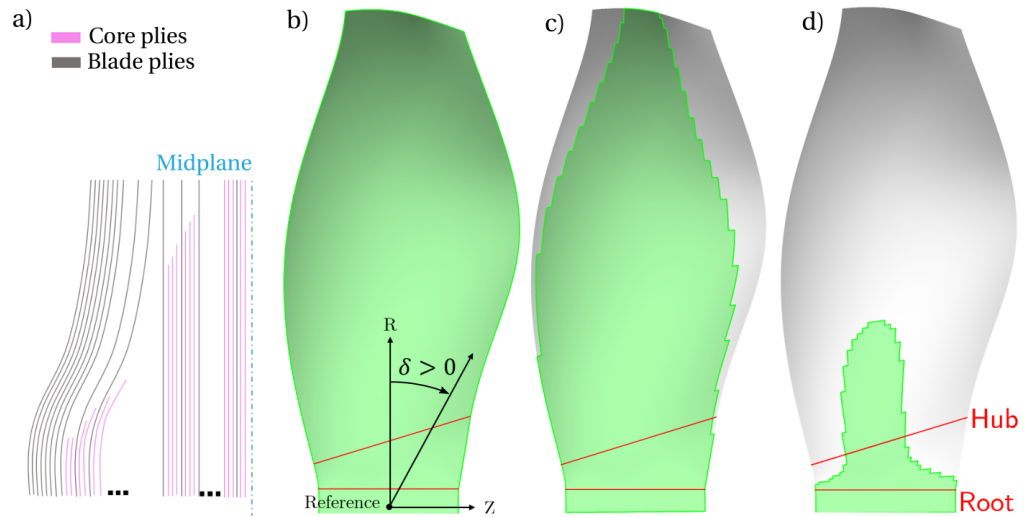


Figure 5. Plies of ECL5 Fan (colored in green) (a) Plies alternation (b) Ply 1 - blade: wrap (c) Ply 6 - blade (d) Ply 19 - core

204 The manufacturing process consists of three steps. First, each pre-impregnated ply
 205 is cut to the pre-defined shape. Secondly, the plies are stacked into two half-molds. Then
 206 both molds are pressed together before heating to start the polymerisation process of
 207 epoxy.

208 The plies are stacked in a particular way to improve the manufacturing process and
 209 final resistance to delamination. It is made of an alternation of blade plies and core plies
 210 as illustrated in Figure 5a. One particular feature is that Ply 1 (the largest, Figure 5b)
 211 is put at the external surface of the blade. Thus, the external surface is composed of only
 212 one ply and has a very low surface roughness without any sharp step.

Ply	1	2	3	4	5	6	7	8	9
Orientation	45°	-45°	-15°	-30°	15°	-15°	70°	0°	0°
Ply	10	11	12	13	14	15	16	17	18
Orientation	-15°	0°	0°	60°	-60°	0°	0°	60°	-60°

Table 3. Blade plies orientation

213 3. Modelling strategies

214 3.1. Aerodynamics

215 For aerodynamic simulations, a structured mesh has been created using AutoGrid⁵.
 216 A single passage of the fan domain contains 3.5 Million cells, with 141 layers in radial
 217 direction. 41 layers are used in the tip gap. The OGV domain contains 1.7 Million cells,
 218 with 101 layers in radial direction and no tip gap. Fan meshing is illustrated in Figure 6.

219 Stationary flow simulations have been conducted with the RANS solver FineTurbo
 220 ⁶. The wall resolution of this mesh is below $y^+ = 1$ for design conditions. A $k - \omega$

⁵ <https://www.numeca.com>

⁶ <https://www.numeca.com>

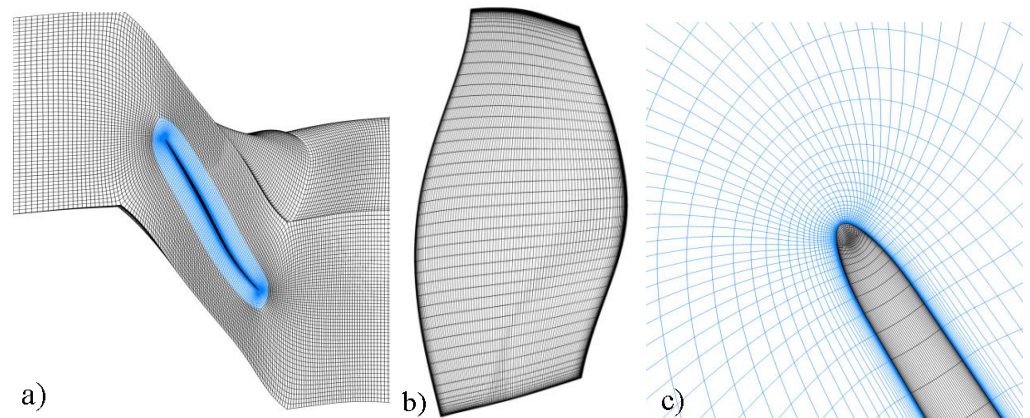


Figure 6. Views of structured mesh for aerodynamic simulations (a) Blade to blade (b) Meridional (c) O-mesh around the leading edge and the matching tip-clearance mesh

221 SST turbulence model is used for the simulations presented in the following. This
 222 methodology was validated in a study on a comparable low-speed fan for the operating
 223 range between choke and peak-pressure [16]. A study on mesh-convergence is presented
 224 in [17].

225 3.2. Mechanics

226 FEM solve Ansys has been used for mechanical simulations⁷. The simulation do-
 227 main and boundary conditions are presented in Figure 7a. A cyclic symmetry condition
 228 is imposed on the faces of the disk sector (in purple) since fan is supposed to be tuned
 229 and all blades identical. The shaft of the test facility is not modeled. To represent the
 230 contact between the disk and the shaft, the disk is clamped in the blue area and the axial
 231 displacement is set to zero in the green area. The aerodynamic pressure field is imposed
 232 on the red surface. Static simulations include large deflection.

233 The blade composite laminate is modelled by a hexaedric 3D mesh (Ansys ACP⁸)
 234 and dedicated procedure to model the ply alternation. To reduce the mesh size, a stack
 235 of plies is modelled with a equivalent element. Each half-thickness is modelled by an
 236 equivalent element (see Figure 7b) with orthotropic properties which are homogenized
 237 from classical laminate theory [19].

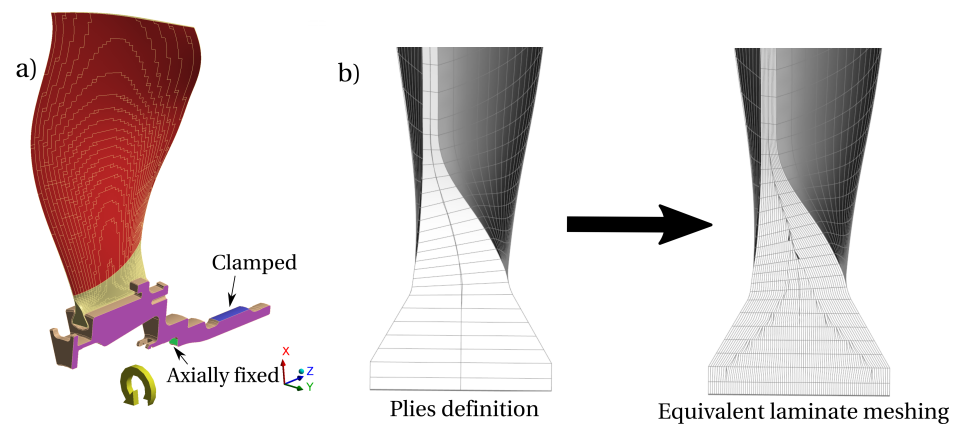


Figure 7. Mechanical model (a) Boundary conditions (b) Blade thickness meshing

⁷ <https://www.ansys.com/products/structures/ansys-mechanical>

⁸ <https://www.ansys.com/products/structures/composite-materials>

238 3.3. Aeroelasticity

239 3.3.1. Energy method

240 Aeroelastic stability is investigated with the energy method [20]. This approach is
 241 based on the assumption that flow fluctuations do not affect structural dynamics and
 242 are linear with the vibration amplitude. It consists in two steps. First, eigenmodes and
 243 natural frequencies are determined. Secondly, an unsteady flow simulation is performed
 244 to compute the harmonic variations of the flow caused by the vibration. Aeroelastic
 245 stability is determined by the energy transfer from the structure to the flow. It is assessed
 246 by pressure force work W per vibration cycle. For a positive work ($W > 0$), the blade
 247 releases energy to the fluid so that vibrations are damped. For a negative work ($W < 0$),
 248 energy is transferred from the fluid into the structure, thus exciting the blade vibration
 249 and leading to aeroelastic instability. In agreement with industrial convention, the
 250 aerodynamic damping value is defined in the following as :

$$\zeta = \frac{W}{4\pi E_k} = \frac{W}{2\pi M_0 \omega_0^2} \quad (1)$$

251 with the modal kinetic energy E_k of the travelling wave, the modal mass M_0 and the
 252 vibration angular frequency ω_0 .

253 3.3.2. Time-linearized simulations

254 The isolated fan is considered for the stability study according to common practice
 255 for fan flutter studies based on self-excitation [1,21,22]. The OGV is one chord distant
 256 downstream of the fan so its potential influence is significantly reduced.

257 To assess the stability of the fan, a time-linearized method (LRANS) is used. Based
 258 on the assumption of small harmonic perturbations, the coordinates of the grid vertices
 259 as well as the flow solution can be decomposed in a steady part and a time dependent
 260 harmonic perturbation. These two parts are simulated separately.

261 The steady flow necessary for LRANS simulations is obtained with the compressible
 262 RANS solver elsA⁹. The solver relies on cell-centered finite volume method on multi-
 263 block structured grids. Convective fluxes are obtained with Jameson scheme with
 264 artificial dissipation. The turbulence is modeled by $k - \omega$ Kok model with vorticity-
 265 based computation of turbulent kinetic energy production. A single passage of the fan
 266 domain is reduced to 2.3 Million cells, with 101 layers in radial direction. 21 layers are
 267 used in the tip gap. The wall resolution remains below $y^+ = 1$. A comparison of the
 268 steady state results to those obtained with FineTurbo was carried out and showed few
 269 disparities, resulting in differences lower than 0.3% in total pressure ratio and 0.3% in
 270 isentropic efficiency for the studied operating points.

271 The unsteady flow is obtained with the solver Turb'Lin [23]. The solution is obtained
 272 in the frequency domain. This vertex-centered LRANS solver has been validated for
 273 transonic separated flows [23,24]. The turbulence model is linearized (see [25] for
 274 comparison of results with frozen turbulence assumption). Upstream and downstream
 275 of the fan, the mesh was extended by numerous fan diameters with strongly reduced
 276 axial resolution to attenuate acoustic reflections.

277 4. Results

278 4.1. Aerodynamic characteristics

279 Stage characteristics of steady RANS calculations, throttled via static pressure at the
 280 stage outlet are presented in Figure 8. It can be seen that overall peak efficiency is well
 281 located at the 100% speedline at a point where the total pressure characteristic is almost
 282 flat. For high speedlines above 100% of design speed, choke occurs in the fan, otherwise
 283 it occurs in the OGV. Towards low mass-flow the speedlines are rolled over and limited
 284 by numerical convergence. This is comparable to the case presented by Rodrigues *et al.*

⁹ <http://elsa.onera.fr/>

285 [16], which has shown that through application of a choked nozzle far downstream the
 286 numerical stability limit can be further extended but still not reaching the stability limit
 287 of the experiment. At highly throttled conditions the performance in experiments are
 288 expected to significantly deviate from numerical results. In this paper, five operating
 289 points representative of the different flow topologies and limits are presented. OP-A is
 290 the design point, OP-B is the maximum pressure ratio and OP-C presents a choked flow
 291 at 100% speed. OP-D and OP-E are operating points near the maximum pressure ratio at
 292 80% speed.

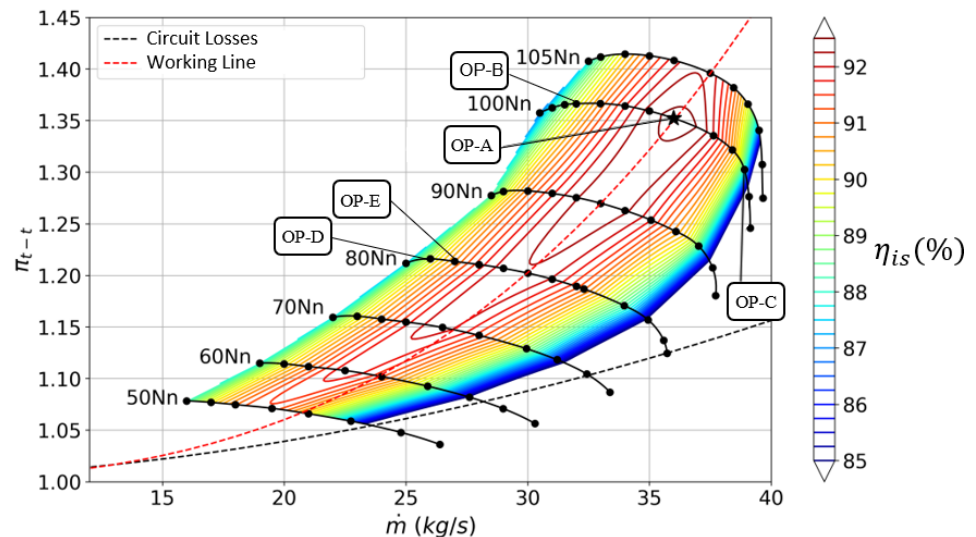


Figure 8. Stage compressor map with operating points of interest

293 A detail of the 100% speedline is given in Figure 9, showing pressure and efficiency
 294 characteristics as well as radial profiles at rotor exit¹⁰. In a wide range between OP-B
 295 and OP-C (representing 19% of design mass flow, depicted cyan in Figure 9), the fan
 296 produces a flat efficiency characteristic above 88%. The evolution between the lowly
 297 throttled operating point OP-C and the higher throttled conditions shows that the fan
 298 is barely affected between hub and midspan. The maximum pressure ratio is observed
 299 near midspan and throttling increases loading of the blade tip. At design point OP-A, the
 300 efficiency profile is almost constant between 20% and 80% span. Influence of tip-leakage
 301 flow is observed between 85% and 100% span for all shown conditions.

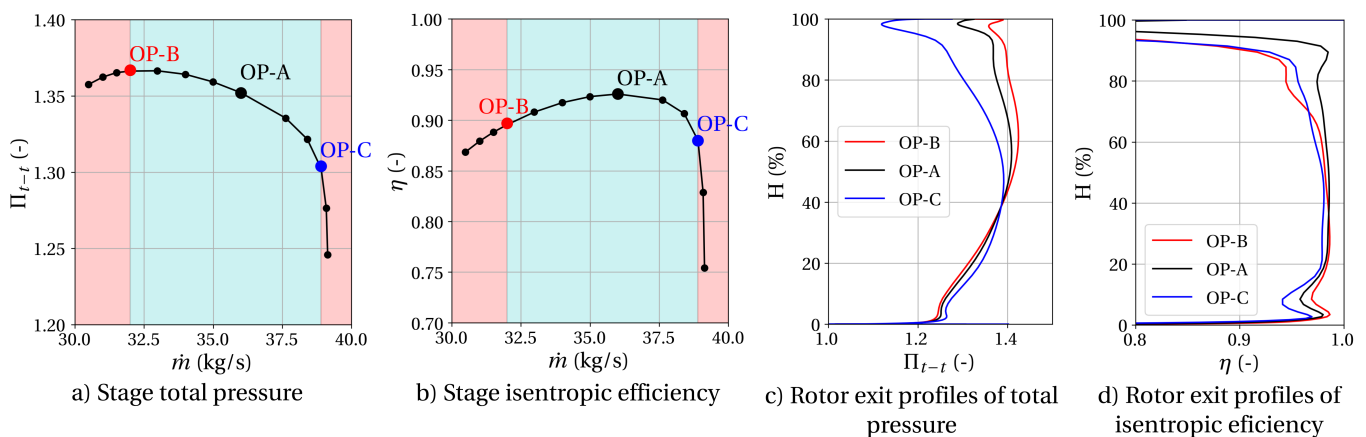


Figure 9. Detail of flow conditions at different operating points at 100% speed

¹⁰ Aerodynamic simulations for 1.5mm constant tip clearance, fabricated fan has tip clearance according to Table 1

302 In Figure 10 the flow structure is presented in more detail for design point OP-A.
303 Isentropic Mach number distribution on the suction side shows the shock structure
304 from midspan and emphasizes the reduced loading at blade tip. Surface streamlines are
305 homogeneous without indications for flow separations but moderate radial migration
306 towards the trailing edge. The blade-to-blade view of the relative Mach number and
307 radial velocity at 95% span indicates low blockage in the tip region and a clearly depicted
308 tip leakage vortex that mildly interacts with the shock.

309 This situation changes at maximum pressure condition OP-B, where the pre-shock
310 Mach number strongly increases, and the surface streamlines show signs of boundary
311 layer separation between 70% span and the casing. Also, radial migration is accen-
312 tuated leading to low relative Mach numbers below 0.2 at the trailing edge at lower
313 channel heights. The blade-to-blade view depicts a strong blockage zone enclosed by the
314 emphasized tip-clearance flow. Here, a stronger interaction with the shock is indicated.

315 The same illustration is given for the maximum pressure point at the 80% speedline
316 OP-D. Here, the flow is mostly subsonic. The supersonic zone (without shock) in
317 isentropic Mach number distribution is due to the circumvention of the leading edge.
318 Surface streamlines indicate significant local backflow at the leading edge for higher
319 channel heights. The blockage region at the fan tip is more severe than at design speed
320 with a widely inclined zone of negative radial velocity. Radial flow migration is highly
321 accentuated at this condition, particularly close to the casing. However, due to low
322 relative velocity, the amplitude of radial velocity close to the casing is lower than for the
323 higher speedline.

324 4.2. Mechanics

325 4.2.1. Static mechanics

326 Hot shapes of the fan blade were determined for each speedline along the work-
327 ing line. Due to the use of anisotropic properties of composite, the maximal blade
328 displacement is 1.15 mm at maximal speed corresponding to 1.0% of the chord. It is
329 comparable to the optimization presented by Schmid *et al.* [13]. Maximum static stress in
330 fiber direction does not exceed 17% of elastic limit. Static stress is then low compared to
331 elastic limit which allows sufficiently high level of dynamic stress during experimental
332 campaign.

333 4.2.2. Mode Shapes

334 Concerning synchronous excitation phenomena, intake distortions are the predomi-
335 nant sources. The potential effects of the OGV are negligible because of the axial distance
336 from the fan, of more than one chord. Experience on similar configurations shows that
337 above the 6th engine order, forced response amplitudes are negligible [16,26]. At 105%
338 of nominal rotation speed, the 6th engine order frequency is 1155 Hz. Mode 3 and Mode
339 4 frequencies are respectively around 900 Hz and 1500 Hz at maximal rotation speed so
340 only the first three modes are considered in the following.

341 Modal data are presented only for positive nodal diameters because they are identi-
342 cal for both co-rotative and contra-rotative modes. Modal frequencies are given in Figure
343 11a for each nodal diameter (ND). Frequency variations are less than 2% except for low
344 nodal diameters between modes of the same family. Mode 1ND0 frequency is lowered
345 by 9% compared to the other NDs. Mode 2ND1 and Mode 3ND1 frequencies are also
346 reduced by respectively 12% and 14%. To assess for mode shape variations between
347 nodal diameters, the Mode Assurance Criterion (MAC) is used [27]. For each mode, the
348 mode shape is compared to its ND8 counterpart. MAC is equal to 1 if the mode shapes
349 are identical, 0 if they are inconsistent. MAC is represented in Figure 11b. Except for
350 ND1, the three modes are superimposed at 1. Modes 2ND1 and 3ND1 are very different
351 from other NDs due to high disc contribution. Mode 1ND0 mode shape is comparable
352 even if its frequency is slightly different. These significant variations between nodal
353 diameters have to be taken into account for aeroelastic studies. Furthermore the accuracy

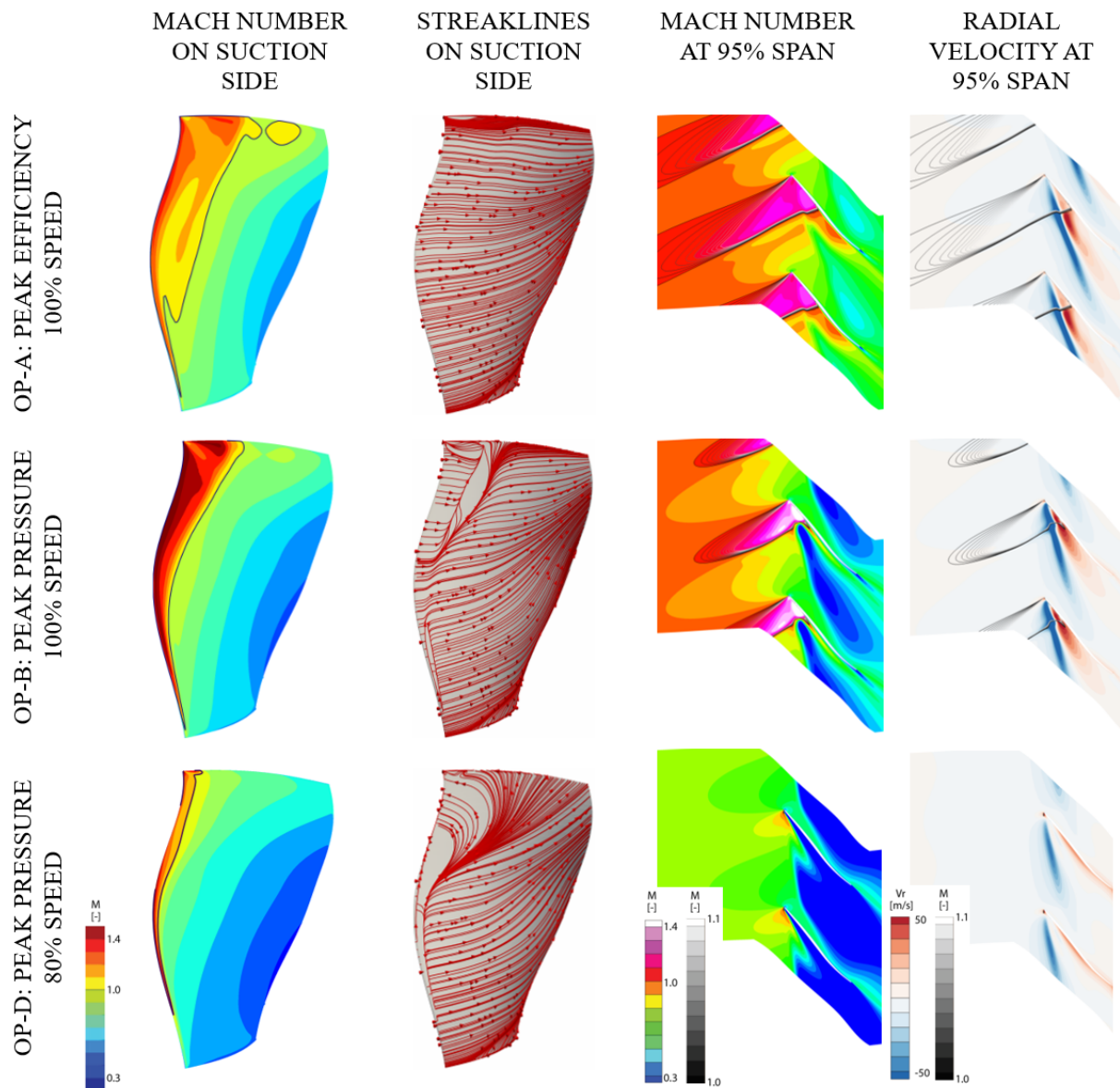


Figure 10. Detail of flow conditions at different operating points

354 of the prediction can be affected by support stiffness and gyroscopic effects. Thus the
 355 choice to model the disk without the shaft could have an important impact on this nodal
 356 diameter (ND1) and must be addressed in further studies.

357 Mode shapes at ND2 are shown in Figure 12. ND2 is representative of all nodal
 358 diameters except ND1. Mode 1 is a bending mode with very low torsional component.
 359 The displacement is almost only circumferential without axial component. Mode 2
 360 shows a pronounced torsional component at the blade tip which is known to be critical
 361 for fluid-structure interactions [6,28]. Mode 3 represents a torsional mode with torsional
 362 center positioned at half-chord at midspan and at the leading edge at the tip.

363 Mode shapes at ND1 are shown in Figure 13. As identified in Figure 11b, Mode
 364 2ND1 and Mode 3ND1 are very different from their other equivalent NDs. In particular,
 365 the mode shape of 3ND1 is closer to Mode 2 than Mode 3.

366 4.2.3. Campbell diagram

367 The Campbell diagram obtained for the fan is presented in Figure 14. The crossings
 368 of the different modes with EO lines are highlighted. Modal frequencies were adjusted

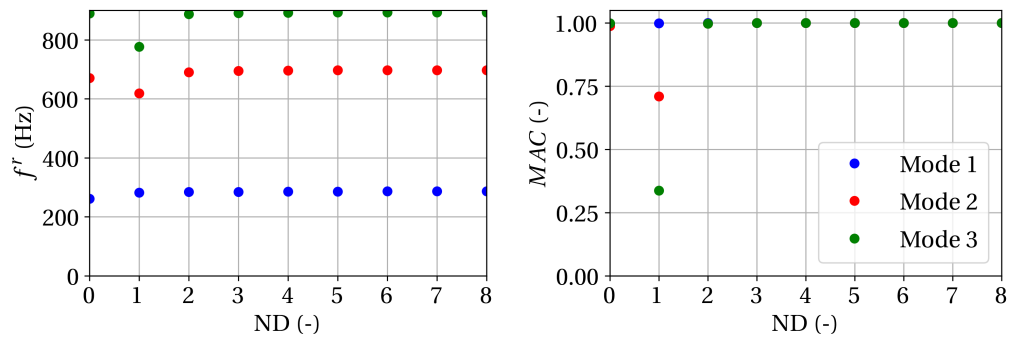


Figure 11. Nodal diameter influence on modes at 100% speed (a) Frequency (b) MAC in comparison with ND8

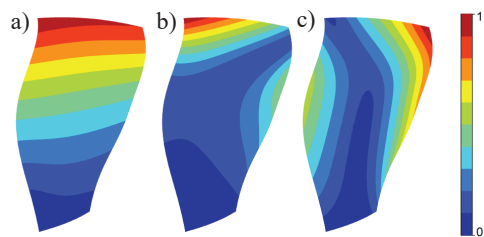


Figure 12. Normalized modal displacement amplitude at ND2 (a) Mode 1 (b) Mode 2 (c) Mode 3

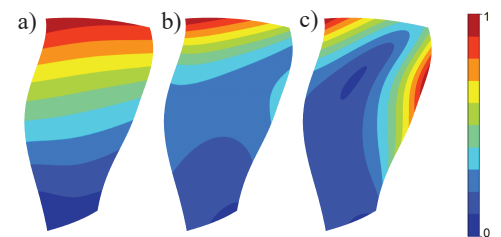


Figure 13. Normalized modal displacement amplitude at ND1 (a) Mode 1 (b) Mode 2 (c) Mode 3

369 during the design process - thanks to carbon fiber orientations - to prevent crossings
 370 above 100% speed. These are sufficiently distributed in frequency to enable safe tests - i.e.
 371 without resonance caused fatigue - at low part-speed ($\sim 50\%$), high part-speed ($\sim 80\%$)
 372 and full speed (100% and above). Experience shows that the predicted frequencies
 373 are slightly higher than those of manufactured blades. Fans produced by the same
 374 manufacturer show that the frequency predictions are nearly 5% higher for torsional
 375 modes [16,26]. Moreover, in previous experimental campaigns, crossing with engine
 376 orders higher than 3 never prevented speedline exploration. Hence, the 80% speedline
 377 should be safely investigated despite the crossing of Mode 3.

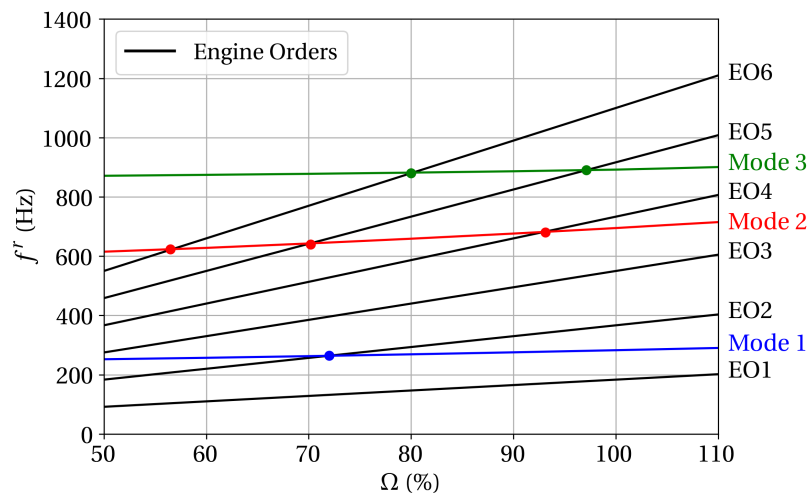


Figure 14. Campbell diagram at ND2

378 4.3. Overall stability analysis

379 4.3.1. Strategy

380 To be representative of industrial configurations and to enable comprehensive
 381 experimental campaigns, it is necessary that the fan presents stable operating conditions
 382 at every planned experimental rotation speed. An exhaustive review would be to test
 383 every nodal diameter of every mode at each rotation speed. Even with LRANS method,
 384 this exceeds the scope of a design study. The systematic stability analysis is restrained to
 385 4 operating points. The design point OP-A serves as reference and must be stable in any
 386 case. OP-B, OP-C, and OP-D are representative of the different flow limits (see Figure 8).

387 Flutter events detected on modern fans mostly involve first bending mode at low
 388 nodal diameter [1,21,22]. However it has been shown by Stapelfeldt and Brandstetter [6]
 389 that torsional mode shapes could lead to aeroelastic instability for a convective mecha-
 390 nism denoted as Non-Synchronous Vibration. Thus all three modes were investigated.
 391 Table 4 gives tip reduced frequencies (see Equation 2) based on chord and upstream
 392 relative velocity at both considered speeds.

$$k_r = \frac{f^r c}{U^{tip}} \quad (2)$$

393 To reduce the number of aeroelastic configurations to simulate, only even numbers
 394 of nodal diameters were simulated in a first step. The nodal diameters ± 1 were added
 395 because their aeroelastic behaviour cannot be deduced from other nodal diameters due
 396 to their mode shape (see Figures 12 and 13). Odd nodal diameters were simulated if
 397 discontinuities in aerodynamic damping curves were observed.

398 4.3.2. Aerodynamic damping

399 Figures 15 and 16 show the aerodynamic damping ratio ζ as function of the nodal
 400 diameter at operating points OP-A to OP-D. The minimal values of each curve are
 401 gathered in Table 5. At OP-A, the design point, aerodamping is always positive and the
 402 minimal value reached is 1.38%. Mode 1 aerodamping variations as function of nodal
 403 diameter are smooth for every operating point. Smallest value occurs between ND0
 404 and ND2 and it exceeds 5% for high NDs. For every operating point studied, Mode 1
 405 is globally more stable than Mode 2 or Mode 3, especially at high NDs. Aerodamping
 406 curves for Mode 2 and Mode 3 are more erratic and minimal values are reached for
 407 various NDs. Only one point was found to be unstable : Mode 2ND5 at OP-D. Among
 408 all other simulated configurations, the aerodamping is always higher than 0.69%.

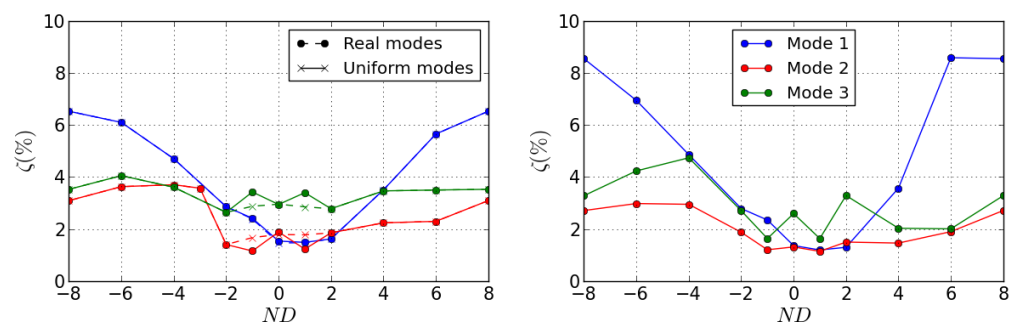


Figure 15. Aerodynamic damping as function of nodal diameter (a) OP-A (b) OP-B

Table 4. Reduced frequency on working line (ND8)

Speedline	80%	100%
Mode 1	0.09	0.08
Mode 2	0.22	0.19
Mode 3	0.30	0.24

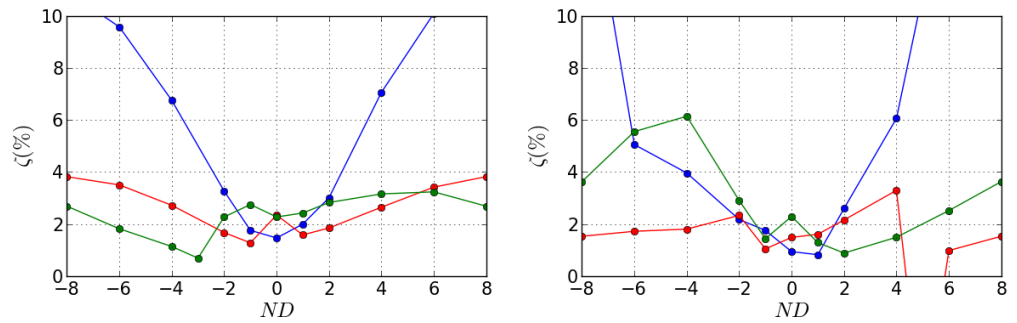


Figure 16. Aerodynamic damping as function of nodal diameter (a) OP-C (b) OP-D

Table 5. Minimum of aerodynamic damping

Operating point	Mode 1	Mode 2	Mode 3
OP-A (100%, 36.0 kg/s)	1.50%	1.38%	2.64%
OP-B (100%, 32.0 kg/s)	1.20%	1.15%	1.64%
OP-C (100%, 38.9 kg/s)	1.47%	1.28%	0.69%
OP-D (80%, 26.0 kg/s)	0.83%	-5.60%	0.89%

4.3.3. Influence of mode frequency and mode shapes on damping

Abrupt changes are observed in damping curves. Among possible reasons, one could be the change of mode shape and frequency for low nodal diameters (see Figure 11). This dependency was quantified at OP-A in Figure 15a. Aerodynamic damping is computed with the same frequency and mode shape from ND8 (Uniform modes) which is similar to ND2 shown Figure 12. It is compared to aerodynamic damping obtained with modes from FEM (Real modes). Resulting curves with uniform modes are smoother. Aerodamping is mostly impacted at mode $2ND \pm 1$ and mode $3ND \pm 1$ whose modal shapes and frequencies were very different of other NDs. Mode $1ND_0$ has a shift in frequency of 9%, however the impact on the stability behavior is marginal. Then erratic changes observed between nodal diameters -2 and 2 are mostly due to mode shapes variations.

4.4. Instability analysis

Upstream and downstream acoustic conditions can produce discontinuities in stability behaviour of the blades [29]. On the operating points studied, numerous switches of cut-on/cut-off conditions can be observed but only three discontinuities related to acoustic behaviour that could lead to instability were identified :

- For Mode $2ND-3$ at OP-A, the stability curve is flat at each side of the discontinuity and the fan remains stable (see Figure 15a).
- For Mode $3ND-3$ at OP-C, the stability reaches a minimum but the fan remains stable (see Figure 16a).
- For Mode $2ND5$ at OP-D, the fan is predicted unstable (see Figure 16b). The stability behaviour for ND4 to ND6 is investigated in the following .

4.4.1. Aeroelastic instability of Mode 2 at OP-D

For Mode $2ND5$ at OP-D, the fan is predicted unstable and the aerodamping reaches a negative minimum at $\zeta = -5.6\%$. The local work at the surface of the blade, normalized like the global work by $U = 4\pi E_k$ (see Equation 1), is plotted Figure 17 for ND4 to ND6. The local work is mostly located at the tip of the blade because of the mode shape (see Figure 12b). Corresponding flow fluctuations at ND5 are more than one order of magnitude higher than for adjacent nodal diameters. For example, if the displacement is arbitrarily set to 1.0% of the chord at the tip, the maximal pressure fluctuations will be 65% of inlet static pressure. At ND5, fluctuations cannot be analyzed quantitatively

441 because it is out of the hypothesis of small perturbations on which the linearized method
442 is based.

443 At ND4 and ND6, the hypothesis of small perturbations remains valid and the
444 work distribution can be analysed. Figure 17 shows that both suction side and pressure
445 side are equivalently involved in overall stability of the blade. The local work patterns
446 between ND4 and ND6 on the pressure side are similar but of opposite sign. On the
447 suction side, patterns are different. At ND4, the work is negative then positive near
448 the leading edge with high values. Whereas the work is mildly positive at ND6 on a
449 large area. This important change of behaviour between these nodal diameters indicates
450 resonance, which is reached near ND5. Damping value obtained at ND5 is questionable
451 due to resonance. However it still shows that instability could be expected at these
452 operating conditions which must be investigated with caution during experiments.

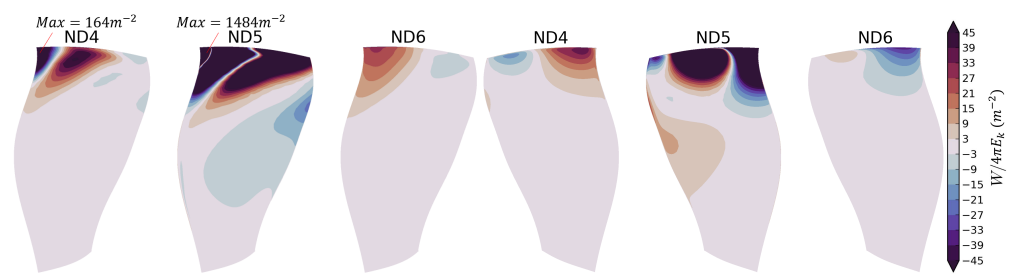


Figure 17. Local work at OP-D for Modes 2ND4 to 2ND6 (left) Suction side (right) Pressure side

453 4.4.2. Acoustic propagation conditions

The propagation conditions [30] of Mode 2 at OP-D are presented in Figure 18. This figure gives an understanding of acoustics in the duct. An acoustic pressure wave generated by a vibrating fan can propagate axially if the frequency in the steady frame of reference f^s is higher than a frequency f^{cut} called cut-off frequency. f^s is determined by the Equation 3.

$$f^s = |f^r + \Omega ND| \quad (3)$$

with f^r the frequency in the rotating frame of reference (i.e. the modal frequency). On the assumption of uniform flow in radial direction, rigid body flow and hard-walled duct, the acoustic cut-off frequency for a subsonic axial flow can be estimated according to Hellmich and Seume [30] as :

$$f^{cut} = \frac{k_r a}{2\pi} \left(\sqrt{1 - M_x^2} \pm M_\theta \right) \quad (4)$$

454 with M_x and M_θ the axial and circumferential Mach number, a the sound speed and
455 k_r the radial modal wave number calculated from Bessel functions. M_x and M_θ are
456 determined by the mean value far upstream and downstream of the blade. Propagation
457 conditions are shown in Figure 18 and were verified in linearized simulations results.
458 Upstream, acoustic waves propagate for ND0 to ND6. Downstream, acoustic waves
459 propagate at ND4 but neither ND5 nor ND6. At the opposite neither ND5 nor ND6
460 propagates. Vibration patterns which fulfill this condition of being cut-on only in one
461 axial direction are known to be critical for aeroelastic stability [4,31]. Discontinuity
462 observed in the aerodamping curve (Figure 19) for OP-D seems to coincide with change
463 of the acoustic propagation conditions shown in Figure 18.

464 4.4.3. Influence of operating point

465 Aeroelastic stability is compared between OP-D and OP-E for Modes 2ND4 to 2ND6.
466 At OP-E, the massflow is 3.8% higher than at OP-D and the leading edge separation
467 size is reduced to 25% of the chord. The acoustic cut-off frequencies at OP-E are not

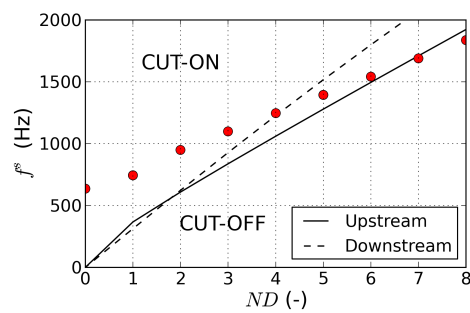


Figure 18. Mode 2 (red dots) acoustic conditions at OP-D et OP-E

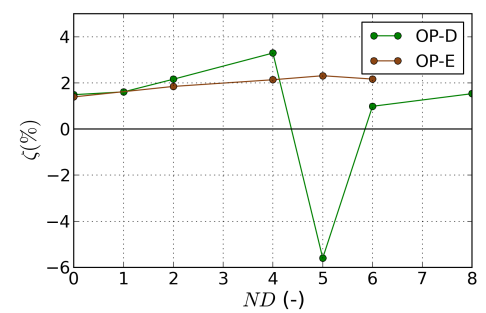


Figure 19. Mode 2 aerodynamic damping at OP-D and OP-E

468 shown in Figure 18 because the differences with OP-D (of less than symbol size) are
 469 not significant compared to the model accuracy. It is verified in simulations that the
 470 acoustic propagation conditions upstream and downstream for ND4 to ND6 are the
 471 same for both operating points. The normalized local work distributions are presented
 472 in Figure 20. Contrary to OP-D (see Figure 17), results show very little influence of the
 473 nodal diameter at this operating point and only stabilizing aerodynamic damping higher
 474 than 2.0% (see Figure 19). On the suction side, the work repartition looks like ND4 at
 475 OP-D but with lower amplitude. Unlike results at OP-D, the work on the pressure side
 476 is one order lower than on suction side. Here, the stability of the blade is driven by the
 477 suction side. The shift of 3.8% in massflow leads to a stable system and suppresses the
 478 strong influence of nodal diameter on fan stability. Thus, even if the acoustic conditions
 479 are similar, the fan remains stable closer to the operating line. Based on the presented
 480 study it is not clear if the acoustic conditions are the dominant driver for instability
 481 or if a convective mechanism as described by Stapelfeldt and Brandstetter [6] leads to
 482 aeroelastic lock-in (Non-Synchronous Vibrations). The latter case is not unlikely as the
 483 ND5 pattern can be in resonance with a backward travelling aerodynamic disturbance
 484 of wave-number 11 (aliased on a 16 blades rotor). This disturbance would propagate
 485 with a speed of $\approx 60\%$ of the fan speed in the absolute frame of reference which is well
 486 in the range of reported convective disturbances [16] but it has not been investigated
 487 before using time-linearized methods. Nevertheless, the detected instability at OP-D
 488 does not prevent experimental exploitation of the speedline as it occurs only close to the
 489 stall limit and in case of convective lock-in limit cycle oscillations are expected.

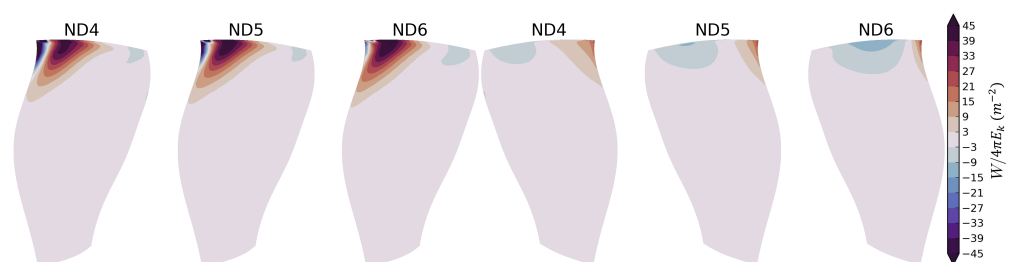


Figure 20. Local work at OP-E for Modes 2ND4 to 2ND6 (left) Suction side (right) Pressure side

490 5. Conclusions

491 A fan representative of modern UHBR fans was designed and presented as an
 492 open test case. The fan blades are made of unidirectional carbon fibres and epoxy
 493 composite plies. To enable an analysis of the dynamic behaviour, structural properties,
 494 manufacturing process and ply orientations were presented.

495 The second mode of the fan has a significant torsional component at the blade tip
 496 which makes it likely to be critical for fluid-structure interactions. Modes of a same

497 family vary significantly at low nodal diameters. The impact of this disparity was
498 quantified in aeroelastic studies. Fan stability was systematically investigated over
499 representative operating points with a time-linearized method. At nominal speed, no
500 aeroelastic instabilities were predicted. Thus, it will be possible to investigate highly
501 loaded operating conditions. At part-speed, a potential instability was identified at
502 throttled conditions and will be investigated in more details in future work. It was
503 shown that the instability area remains localized far from the operating line and it will
504 be in the focus of the planned experimental campaign which will start mid 2022.

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507 Pierre Duquesne: conceptualization, methodology, investigation, writing-review, supervision
508 Stéphane Aubert: conceptualization, methodology, investigation, writing-review, supervision,
509 funding acquisition : CIRT
510 Laurent Blanc: conceptualization, methodology, investigation, writing-review, supervision
511 Pascal Ferrand: conceptualization, methodology, investigation, writing-review, supervision
512 Xavier Ottavy: conceptualization, methodology, investigation, writing-review, supervision, project
513 administration/funding acquisition: CleanSky N°864719
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530 Abbreviations

531 The following abbreviations are used in this manuscript:

532	EO	Engine Order
	FEM	Finite Element Method
	LRANS	Linearized Reynolds-Averaged Navier–Stokes
	ND	Nodal Diameter
533	NSV	Non Synchronous Vibration
	OGV	Outlet Guide Vanes (Stator)
	OP	Operating Point
	RANS	Reynolds-Averaged Navier–Stokes
	UHBR	Ultra High Bypass Ratio

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