

Experimental and Numerical Investigation of Aerodynamic Damping

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Abstract

Despite the progress made in numerical prediction of aerodynamic damping of turbomachinery, large uncertainties in comparison to experiments are still no exception. Furthermore, there is only a limited amount of available test data on aerodynamic damping in open literature to further improve numerical models and to extend the physical understanding of blade vibration. On this basis, the present thesis introduces a new aeromechanical test facility for axial compressor rotors called **ACTIVE** (**A**xial **C**ompressor **T**est rig for the **I**nvestigation of blisk **V**ibrations with an active **E**xcitation system), which is located at the Institute of Thermal Turbomachinery and Machinery Laboratory (ITSM) of the University of Stuttgart. Thanks to the closed loop design, the test facility can be operated at variable pressure levels as well as at vacuum without changing the test setup. An electromagnetic excitation system is developed, which allows for the excitation of normal modes independent of the rotational speed and operating point and without disturbing the flow field due to its location below the hub line. The test facility is equipped with a blade tip-timing system (BTT) to measure blade vibration contactless. A unique validation experiment is presented, where the BTT system is validated by means of a scanning laser Doppler vibrometer in combination with an optical derotator to enable vibration measurements on the rotating blisk.

The present thesis shows the capability of the test facility to experimentally determine the aerodynamic damping of an axial compressor rotor with high accuracy. The rotor is manufactured as a blisk (blade integrated disk) and was designed as part of the present work. Furthermore, the experimental results are compared to accompanying numerical simulations. In a first step, the natural modes of the blisk are determined by means of experimental modal analyses to validate the finite element (FE) simulations. Campbell diagram measurements reveal a strong influence of the Coriolis force for some modes causing a significant split of the backward and forward traveling wave modes. Subsequently, detailed flow field measurements are applied to validate the steady-state computational fluid dynamics (CFD) simulations. Hundreds of excitation experiments are conducted to measure aerodynamic damping of

multiple modes of the first bending mode family (B1) at various operating points and different rotational speeds. Aerodynamic damping is determined by subtracting the structural and material damping, which is measured with the blisk operating in vacuum, from the overall damping. The results indicate a clear dependency of aerodynamic damping on the fluid density. Furthermore, aerodynamic damping correlates with the mass flow rate for all investigated modes. The aerodynamic damping CFD simulations agree well with the experimental results. However, limitations of the simplified CFD model are observed for one mode, where an acoustic resonance is detected in the experiment. An analysis of the casing pressure reveals a linear correlation of the pressure amplitudes with the blade displacement amplitudes and it is shown, that blisk damping can be determined solely with time-resolved pressure information on the rotor casing. The comparison of measurements and simulations for modes of the first torsional mode family (T1) with high reduced frequencies confirm the good agreement in terms of aerodynamic damping.

Kurzfassung

Trotz des Fortschritts bei der numerischen Vorhersage der aerodynamischen Dämpfung von Turbomaschinen sind große Unsicherheiten im Abgleich mit Messungen keine Seltenheit. Zusätzlich ist die Verfügbarkeit von experimentellen Daten in der Literatur zur Validierung und Weiterentwicklung der numerischen Modelle sehr begrenzt. Um diesen Zustand zu verbessern präsentiert die vorliegende Arbeit einen neuen aeromechanischen Prüfstand für Rotoren von Axialverdichtern, genannt ACTIVE (Axial Compressor Test rig for the Investigation of blisk Vibrations with an active Excitation system), welcher am Institut für Thermische Strömungsmaschinen und Maschinenlaboratorium (ITSM) der Universität Stuttgart aufgebaut wurde. Dank des Aufbaus als geschlossener Kreislauf (closed loop) kann das Druckniveau im Prüfstand frei gewählt werden. Es besteht auch die Möglichkeit den Prüfstand im Betrieb zu evakuieren ohne den Aufbau zu ändern. Um Eigenmoden des Verdichterrotors unabhängig von der Drehzahl und ohne eine negative Beeinflussung der Strömung anzuregen, wurde ein elektromagnetisches Anregungssystem entwickelt, welches in der Nabe stromauf des Rotors positioniert wird. Der Prüfstand ist mit einem optischen Messsystem zur berührungslosen Schaufelschwingungsmessung ausgestattet (blade tip-tipping system (BTT)), welches im Rahmen der Arbeiten validiert wird. Hierzu wird ein Laservibrometer in Kombination mit einem optischen Derotator verwendet, um Schaufelschwingungen simultan zum BTT System im rotierenden Bezugssystem zu messen.

In der vorliegenden Arbeit werden die Möglichkeiten des Prüfstandes gezeigt, die aerodynamische Dämpfung eines Axialverdichterrotors mit einer hohen Genauigkeit zu messen. Der Rotor ist als blisk (blade integrated disk) gefertigt und wurde im Rahmen der vorliegenden Arbeit strömungstechnisch und strukturdynamisch ausgelegt. Zusätzlich werden die experimentellen Ergebnisse mit begleitenden numerischen Simulationen verglichen. Im ersten Schritt werden die Eigenmoden der blisk durch experimentelle Modalanalysen bestimmt und mit den Ergebnissen der Finite Elemente (FE) Simulationen verglichen. Ein durch Messungen erzeugtes Campbell Diagramm zeigt

einen deutlichen Einfluss der Coriolis Kraft für einige Moden, wodurch ein signifikantes Aufsplitten der Eigenfrequenzen für vorwärts und rückwärts laufende Moden verursacht wird.

Detaillierte Strömungsfeldmessungen werden für die Validierung der stationären numerischen Strömungssimulationen verwendet. Für die experimentelle Messung der aerodynamischen Dämpfung werden anschließend eine Vielzahl an Anregungsexperimenten bei verschiedenen Drehzahlen und Betriebspunkten durchgeführt. Der Fokus liegt hierbei auf den Moden der ersten Biegemode-Famile (B1). Die aerodynamische Dämpfung wird hierbei bestimmt indem die Material- und Strukturdämpfung, welche bei Vakuum gemessen wird, von der Gesamtdämpfung abgezogen wird. Die Messergebnisse zeigen eine deutliche Abhängigkeit der aerodynamischen Dämpfung von der Fluidichte. Es kann zusätzlich eine Korrelation der aerodynamischen Dämpfung mit dem Massenstrom des Verdichters festgestellt werden. Ein Vergleich mit den Ergebnissen der numerischen Bestimmung der aerodynamischen Dämpfung zeigt eine gute Übereinstimmung mit den experimentellen Ergebnissen. Ein signifikant größerer Fehler der numerischen Simulationen tritt nur für eine Eigenmode auf, bei welcher eine akustische Resonanz im Experiment vorliegt. Die Auswertung von Wanddrücken ergibt einen linearen Zusammenhang zwischen den Druck- und Schaufelamplituden und es wird gezeigt, dass eine Bestimmung der Dämpfung auch auf Basis der zeitaufgelösten Druckdaten im Bereich der blisk möglich ist. Auch ein Vergleich der aerodynamischen Dämpfung für Moden der ersten Torsionsmode-Familie (T1) bestätigt eine sehr gute Übereinstimmung von Messung und Simulation.

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Contents

Abstract	iii
Kurzfassung	v
Acknowledgments	vii
Nomenclature	xv
1 Introduction and Motivation	1
2 Basics of Vibrations and Aeroelasticity in Turbomachinery	5
2.1 Undamped, Free Vibration	7
2.2 Damped, Forced Vibration	13
2.2.1 Viscous Damping	15
2.2.2 Friction Damping	18
2.2.3 Hysteresis Damping and Energy Method	18
2.3 Aeroelasticity	20
2.3.1 Flutter	24
2.3.2 Forced Response	26
2.3.3 Other Aeroelastic Excitation Phenomena	29
2.4 Aerodynamic Damping	30
2.4.1 Influence of Geometry and Material	31
2.4.2 Influence of Operating Conditions	33
2.4.3 Influence of Installation Conditions, Acoustic Resonance and Acoustic Reflection	34

2.4.4	Determination of Aerodynamic Damping with CFD	37
2.4.5	Determination with Experiments	41
2.5	Influence of Mistuning and Coriolis Effect	44
3	Review of Experimental Research on Vibration and Aerodynamic Damping	47
3.1	Test Rigs with Rotor-Stator Interaction Excitation	47
3.2	Test Rigs with an Additional Excitation System	49
3.2.1	Piezoelectric Excitation	50
3.2.2	Flow Jet Excitation	51
3.2.3	Acoustic Excitation	52
3.2.4	Magnetic Excitation	52
3.2.5	Conclusions	55
4	Objectives and Approach	57
5	Experimental Setup and Test Object	59
5.1	Test Facility	59
5.2	Test Object	64
5.3	Measurement Technique	68
5.3.1	Vibration Monitoring and Rotational Speed	68
5.3.2	Flow Meter and Inlet Conditions	69
5.3.3	Casing Pressure and Pneumatic 4-hole Probes	71
5.3.4	Blade Vibration	74
5.4	Excitation System and Blisk Monitoring	75
5.5	Operation and Data Post Processing	79
5.5.1	Compressor Map, Steady Measurements and Vacuum . . .	79

5.5.2	Unsteady Pressure Data	81
5.5.3	Excitation	82
5.5.4	Damping	89
6	Numerical Models	93
6.1	FE Simulations	93
6.2	CFD Simulations	95
6.2.1	Influence of Different CFD Domains	97
6.2.2	Influence of CFD Mesh Resolution	101
6.2.3	Influence of Displacement Amplitudes and Number of Harmonics	103
6.2.4	Conclusions and Final CFD Setup	105
7	Structural Dynamics of the Blisk and FE Validation	107
7.1	Experimental Modal Analysis: Free-Free	107
7.2	Experimental Modal Analysis: Blisk Mounted	113
7.3	Campbell Diagram	117
7.4	Conclusions	119
8	Validation of the Blade Tip Timing System	121
8.1	Conclusions	130
9	Steady Flow Field Measurements and CFD Validation	133
9.1	Conclusions	140
10	Preliminary Studies for Damping Measurements	143
10.1	Sweep Rate	143
10.2	Excitation Level and Reproducibility	144
10.3	Free Decay versus Excitation Sweep	148

10.4 Material Damping of Mode Family B1	149
10.5 Conclusions and Discussion of B1ND0	153
11 Results of the First Bending Mode Family	157
11.1 Aerodynamic Damping at Different Pressure Levels	158
11.2 Aerodynamic Damping for the Complete Compressor Map	161
11.3 S-Curves	173
11.4 Unsteady Casing Pressure at Constant Excitation	174
11.5 Unsteady Casing Pressure at Sweep Excitation	181
11.6 Capability and Limitations of the Excitation System	183
11.7 Conclusions	185
12 Aerodynamic Damping of Torsional Modes	189
13 Summary and Outlook	195
Bibliography	201
A Appendix to Chapter 2	223
A.1 Kinetic Energy of Double Modes	223
B Appendix to Chapter 5	227
B.1 Blade Tip Timing Measurement Conversion	227
B.2 BTT Sensor Positions	228
C Appendix to Chapter 6	231
C.1 Superposition of a Double Mode in CFD	231
D Appendix to Chapter 8	233

E	Appendix to Chapter 9	235
E.1	Reynolds Number	235
F	Appendix to Chapter 10 and 11	237

Nomenclature

Latin Symbols

A	m^2	area
C	N s m^{-1}	damping
$\underline{\underline{C}}$	N s m^{-1}	damping matrix
c	m s^{-1}	fluid velocity
c_{ax}	m	axial chord length
c_s	m s^{-1}	speed of sound
d	m	displacement
$\underline{e_h}$	–	unit vector of blade displacement
E	GPa	Young's modulus
f	Hz	frequency
$\dot{f}$	Hz s^{-1}	frequency sweep rate
f_0	Hz	natural frequency
f_d	Hz	damped frequency
F	N	force magnitude
$\underline{F}$	N	force vector
g	m s^{-2}	gravitational acceleration
$\underline{h}$	m	complex blade displacement vector
i	–	imaginary unit, $i = \sqrt{-1}$
I	A	electric current
$\underline{\underline{I}}$	–	identity matrix
j	–	integer value
k	–	reduced frequency
k	N m^{-1}	stiffness
k_m	m^{-1}	wave number
$\underline{\underline{K}}$	N m^{-1}	stiffness matrix
l	m	length

L	H	inductance
$\underline{m}$	kg	modal mass matrix
$\underline{\underline{m}}$	kg	mass
$\dot{m}$	kg s^{-1}	mass flow rate
$\underline{\underline{M}}$	kg	mass matrix
Ma	–	Mach number
N	–	number of blades
N_w	–	number of windings
n_{nd}	–	number of nodal diameter
n	rpm	rotational speed
$\underline{n}$	–	surface normal vector
p	mbar or Pa	(static) pressure
Q	–	quality factor
r	–	normal mode index
r	m	radial coordinate
R	Ω	electric resistance
Re	–	Reynolds number
t	s	time
T	$^{\circ}\text{C}$ or K	temperature
U	V	voltage
$\underline{v}$	ms^{-1}	blade vibration velocity vector
W	J	work or energy
w	ms^{-1}	relative velocity
x	m	displacement
$\dot{x}$	ms^{-1}	first time derivative of x (velocity)
$\ddot{x}$	ms^{-2}	second time derivative of x (acceleration)
z	m	axial coordinate

Greek Symbols

α	$^{\circ}$	absolute flow angle
α	$^{\circ}$	mode vector angle

α	—	damping coefficient for Rayleigh damping
β	°	blade angle
β	—	damping coefficient for Rayleigh damping
γ	—	heat capacity ratio
δ	—	logarithmic decrement
ζ	—	critical damping ratio
θ	rad	circumferential coordinate
κ	—	reduced sweep velocity
ν	—	Poisson's ratio
ν	$\text{m}^2 \text{s}^{-1}$	kinematic viscosity
π	—	pressure ratio
ρ	kg m^{-3}	density
σ	MS m^{-1}	electric conductivity
σ	rad	interblade phase angle
σ_{vM}	N mm^{-2}	von Mises stress
$\underline{\phi}$	$\text{m kg}^{-0.5}$	mass-normalized eigenvector
$\underline{\underline{\Phi}}$	$\text{m kg}^{-0.5}$	mass-normalized eigenvector matrix
φ	rad	phase
ψ	—	specific damping capacity
$\underline{\psi}$	m	eigenvector
$\underline{\underline{\Psi}}$	m	eigenvector matrix
ω	rad s^{-1}	frequency
ω_0	rad s^{-1}	natural frequency
ω_{d}	rad s^{-1}	damped frequency
Ω	rad s^{-1}	angular velocity

Subscripts

—	vector
$\underline{\quad}$	matrix
$\underline{\underline{\quad}}$	
a	alternating
ave	average

aero	aerodynamic
aero.damp.	aerodynamic damping
aero.excit.	aerodynamic excitation
calib	calibrated
corr	corrected
crit	critical
est	estimated
ext	external
in	inlet
m	mean
mag	magnetic
mat	material
max	maximum value
min	minimum value
norm	normalized
out	outlet
ref	reference
struc	structural
s	static
t	total

Abbreviations

1D-SLDV	1-dimensional scanning laser Doppler vibrometer
3D-SLDV	3-dimensional scanning laser Doppler vibrometer
AC	alternating current
ACARE	Advisory Council for Aviation Research and Innovation in Europe
AGARD	Advisory Group for Aerospace Research and Development
B1	first bending mode family
blisk	blade integrated disk
BPF	blade passing frequency
BTT	blade tip timing

BTW	backward traveling wave
CFD	computational fluid dynamics
CI	confidence interval
DAQ	data acquisition
DC	direct current
DFT	discrete Fourier transformation
DP	design point
EMA	experimental modal analysis
EO	excitation order
EXP	experimental / experiment
FE	finite element
FFT	fast Fourier transformation
FRF	frequency response function
FTW	forward traveling wave
HCF	high cycle fatigue
IC	influence coefficients
ITSM	Institute of Thermal Turbomachinery and Machinery Laboratory, University of Stuttgart, Germany
IQR	interquartile range
KTH	Royal Institute of Technology, Stockholm, Sweden
LCF	low cycle fatigue
LE	leading edge
MAC	modal assurance criterion
MDOF	multi-degree-of-freedom (system)
MIT	Massachusetts Institute of Technology, Cambridge, USA
NC	nodal circle
NC	near choke
ND	nodal diameter
NS	near stall
OP	operating point
PLC	programmable logic controller
RANS	Reynolds-averaged Navier-Stokes (equations)
RSD	relative standard deviation

SD	standard deviation
SDOF	single-degree-of-freedom (system)
T1	first torsional mode family
TE	trailing edge
TWM	traveling wave mode
TUD	Technical University of Darmstadt, Germany

Pressure Units

Millibar (mbar) as well as Pascal (Pa) are used in the present work as pressure units:

$$1 \text{ mbar} = 0.1 \text{ kPa} = 1 \text{ hPa} = 100 \text{ Pa}$$

$$1 \text{ bar} = 1000 \text{ mbar} = 1 \times 10^5 \text{ Pa}$$

1 Introduction and Motivation

Following the first successful operations of gas turbines in the late 1930's and early 1940's for electric power production in Switzerland [51] and as jet engines in Germany and England [51, 131], gas turbines quickly became the main propulsion system for military and large commercial aircraft. With their high specific power and the continuous improvement in efficiency and safety since the first applications, gas turbines paved the way for today's commercial aviation as the safest means of transportation with more than 4.5 billion passengers in 2019 and a total revenue of 835 billion USD [56, 109]. This market is still growing significantly and forecasts predict a demand of more than 40000 large passenger and freight aircraft over the next 20 years with an annual growth in passenger traffic of 4 % [23].¹

With the market for gas turbines growing constantly, the requirements for new gas turbine generations in terms of efficiency and weight reduction are increasing as well. This is not only driven by airlines' demand to minimize operating costs but more importantly by fiercer environmental regulations and ambitious goals of the European Union to decrease the CO₂ and NO_x emissions of commercial aircraft in 2050 by 75 % and 90 %, respectively [55]. In order to meet these targets and to reduce the specific fuel consumption, significant efficiency improvements as well as weight reductions are key. With focus on the compressor of the gas turbine, design engineers attempt to further improve efficiency and decrease weight by reducing the number of stages, which necessitates a rise in pressure ratio of each compressor stage if the overall pressure ratio and thus engine power shall be maintained. Furthermore, instead of assembling compressor rotors with individual blades on a rotor hub disk, rotors are more and more designed as one integral part, so called 'blisks' (blade integrated disk), which offer a significant reduction in weight and complexity. However, the mechanical damping in the blade-to-

¹The commercial aviation market forecast of Boeing [23] was published in October 2020 during the COVID-19 pandemic. Boeing assumes no long term impact of the COVID-19 pandemic: "We remain confident in the resilience of commercial aviation" and "The fundamentals that have driven air travel the past five decades and doubled air traffic over the past 20 years remain intact."

disk connection, which is beneficial in terms of vibrations, is removed with a blisk design. As a consequence of the decrease in damping and the increase in pressure ratio and hence in loading, the risk of high blade vibration amplitudes intensifies, which can compromise the structural integrity due to high cycle fatigue (HCF). The trend towards thinner blades with more complex shapes to increase aerodynamic efficiency along with a reduction in the spacing between stages to reduce weight further amplify HCF risks. Two main vibration phenomena are commonly observable for different compressor stages: The flutter phenomenon as a self-excited blade vibration in the fan and the first stage of the compressor and forced response blade vibrations due to the rotor-stator interaction in all stages. In order to develop a compressor design, which ensures safe operation even in off-design conditions, engineers have to consider both, the steady and unsteady aerodynamics as well as structural dynamics and the non-linear coupling between the different disciplines. This combined field of research is known as aeroelasticity. Often, blade vibration and HCF problems become visible in late stages of turbomachinery development or even when production already started, hence costly redesigns or short overhaul periods are inevitable. The US Air Force, for example, estimated their costs just due to HCF at over 400 million USD per year in 1994 [80].

In order to optimize compressor design in terms of blade vibrations, extensive test campaigns had to be conducted in the past as the only option available. With the increase in computational power and the improvement in numerical methods in the last decades, unsteady computational fluid dynamics (CFD) in combination with finite element (FE) simulations are applied nowadays in early stages of the development to reduce the number of required experiments. While aeroelastic simulations offer the valuable option to systematically investigate different designs, the reliability and accuracy of the numerical results are often insufficient and differ significantly with various flow solvers and different numerical approaches. In particular the numerical computation of the aerodynamic damping, which describes the amplifying or damping influence of the unsteady flow field of a vibrating blade, is challenging. Although progress has been made in recent years and the numerical models become more sophisticated, significant uncertainties in the order of 100-200 % are no exception [67]. Nevertheless, there are strong efforts to further increase the use of numerical methods and ultimately replace expen-

sive measurements entirely. In 2017, the ACARE set the target in their strategic research agenda that by 2050, advanced multi-physics simulation models should “eliminat[e] ground test rigs completely” [1]. Crucial for reaching these very ambitious targets is the development and improvement of numerical methods, which necessitates high quality test data in the first place.

With regards to research on aerodynamic damping of compressor bladings, numerous publications are available applying unsteady CFD and CFD-FE coupled models but very few offer comparisons to experimental data for the validation of numerical models. Moreover, if experimental results of aerodynamic damping measurements are shown in open literature, they are often limited to one or a few operating points and mode shapes.

The intention of the present work is to reduce this discrepancy by contributing additional validation data to the research community for an axial compressor blisk. Therefore, a test rig with an innovative excitation system is built and the aerodynamic damping is measured contactless and with high accuracy for a broad range of operating points and a multiple normal modes. Furthermore, the results are compared to CFD simulations and conclusions are drawn on how the aerodynamic damping varies for different flow conditions.

Srinivasan [194] wrote in 1997 regarding future research in the the field of aeroelasticity: “An excellent beginning can be made at the laboratory level to obtain good data. We need accurate measurements of aerodynamic and structural characteristics through nonintervening measurement tools” - this statement is still very true today and it is exactly the motivation for the work presented in the following.

2 Basics of Vibrations and Aeroelasticity in Turbomachinery

Vibrations in turbomachinery are of great concern for design engineers and they are subject to turbomachinery research since the beginning of the 20th century. With the increasing industrial production of steam turbines in the 1910's and 1920's and the tendency to enhance power output and efficiency by introducing larger and slimmer rotor designs, failures due to vibrations increased significantly. Baumann [10] shows this in his review paper on steam turbine development, where he summarizes steam turbine accidents in the United States. Most of the failures can be attributed to excessive vibrations causing fractures and severe rotor disk bursts. Baumann names shaft imbalances but also partial steam admission and small blade-to-blade variations in the stators as vibration excitation sources with excitation frequencies as harmonics of the rotational speed. Besides, he describes the failure mechanism as a local exceedance of the elastic stress limit due to a superposition of static centrifugal stress and additional vibratory stress. A few years later in 1924, Campbell [30] provides deep insight in steam turbine failures due to vibrations with an impressive collection of photographs of broken turbine disks. Furthermore, Campbell experimentally investigates rotor disk vibrations in detail and introduces the famous frequency-speed diagram, known as the *Campbell diagram*. The importance of this paper cannot be underestimated as Campbell laid the foundation for a century of research on turbomachinery vibration and important parts of the paper will be further discussed in this chapter.

Similar to earlier observations in steam turbines, vibration problems occurred in gas turbines as they entered the market in the 1940's. Shannon [191] describes in 1945 multiple failures in gas turbines. He distinguishes between vibrations due to rotor-stator interaction and stall flutter, which is "due to the aerodynamic damping changing from positive to negative". Shannon also discusses the determination of aerodynamic damping of compressor blades analytically. It is impressive to see that by 1945, researchers already developed a deep understanding of basic aeroelastic principles in turbomachin-

ery, which are still subject of today's research.

One year after Shannon's review paper, Collar [38] publishes his famous and often cited paper, where he connects inertia, elastic and aerodynamic forces by introducing the "triangle of forces" (also known as *Collar triangle*, see fig. 2.7), which specifies the definition of aeroelasticity. While Collar summarized observations and investigations in the development of aircraft in the 1930's and 1940's and manifested the term aeroelasticity for research on aircraft wing vibrations, the turbomachinery community quickly adopted the definition as similar physical phenomena are observed for turbomachinery blading. Later, the Collar triangle was extended to aerothermal effects and active vibration control [72, 168].

Within the following decades, aeroelasticity became an important field of research as new vibration problems appeared due to turbomachinery with significantly higher loads, for example by introducing transonic compressors in the 1950's [179], and more aggressive blade designs with higher aspect ratios (thinner and longer blades). A comprehensive collection of aeroelastic research up to the late 1980's is provided in the AGARD manual [179, 180]. The manual shows the progress made on the prediction of aeroelastic effects in turbomachinery using empirical and analytical theories along with numerical calculations, which became affordable due to the developments in computer systems. However, the numerical simulations were still predominantly two-dimensional in their application, see also [212], and "the importance of three-dimensional flow effects [was] still insufficiently understood" [179] at that time. This changed in the last three decades with the emergence of numerous 3D-CFD RANS solvers and the further increase in available computational power, which allowed for detailed studies of turbomachinery flow fields, firstly in a steady-state manner but more and more often also with transient, time-resolving simulations. In recent years, most of the commonly used CFD solvers in academia and industry implemented methods for aeroelastic simulations enabling the numerical determination of aeroelastic forcing and damping in the design process of turbomachinery (for example [39, 153, 167, 206]).

The rise of aeroelastic CFD simulations was accompanied by the application of sophisticated measurement technique. Strain gauges were often used in the past to measure blade vibration in operation allowing for the determination of aerodynamic damping [41, 120, 171], and they still represent a reli-

able technique in today's experimental research. However, the development of non-intrusive blade tip timing systems in the last decades, see for example [95, 126, 227], enable faster and more detailed measurements of blade vibration and hence support and will ultimately replace expensive strain gauge measurements. More information on blade tip timing is provided in section 5.3.4 since it is also used in the present work as the main measurement technique for blade vibration. In addition, very fast pressure transducers are often applied in recent years to gain insight into the transient flow field and to provide validation data for aeroelastic simulations, for example [22, 68, 215].

In the following sections, aeroelasticity in turbomachinery will be discussed in detail along with a review of important research in this field. A very broad overview of aeroelasticity in general can be found in Dowell [48].

2.1 Undamped, Free Vibration

A turbomachinery rotor can be considered as a system with multiple degrees of freedom (MDOF) in terms of its structural dynamics. For a thorough overview of structural dynamics in general and with respect to turbomachinery, the work of Ewins [57–60] is recommended, which is used as one of the main references for the following section.

In general, the undamped free vibration of a MDOF system can be calculated using the following equations of motion:

$$\underline{\underline{M}}\ddot{\underline{x}} + \underline{\underline{K}}\underline{x} = \underline{0} \quad (2.1)$$

where $\underline{\underline{M}}$ is the mass matrix, $\underline{\underline{K}}$ the stiffness matrix and $\underline{x}$ represents the displacement vector. The process of solving eq. (2.1) for the normal mode shapes and natural frequencies of the system is often referred to as *modal analysis*. This is commonly done by assuming a solely harmonic motion of the system for a specific frequency ω

$$\underline{x}(t) = \underline{X}e^{i\omega t} \quad (2.2)$$

with $\underline{X}$ as a vector of complex displacement amplitudes. Using this assumption, eq. (2.2) becomes

$$(\underline{\underline{K}} - \omega^2 \underline{\underline{M}}) \underline{X} e^{i\omega t} = \underline{0} \quad (2.3)$$

Solutions exist for specific natural frequency values ω_0 (or eigenvalues ω_0^2), which can be calculated by solving

$$\det |\underline{\underline{K}} - \omega^2 \underline{\underline{M}}| = 0 \quad (2.4)$$

The corresponding normal mode shape $\underline{\psi}_{0,r}$ for a specific natural frequency $\omega_{0,r}$ can be determined subsequently with

$$(\underline{\underline{K}} - \omega_{0,r}^2 \underline{\underline{M}}) \underline{\psi}_{0,r} = \underline{0} \quad (2.5)$$

It has to be noted that while the natural frequencies represent unique values, the normal mode shapes $\underline{\psi}_0$ consist of relative amplitudes and they can be scaled arbitrarily [57].

The solution for a N degrees of freedom system can be summarized with two $N \times N$ eigenmatrices, a diagonal matrix $\underline{\omega}_0^2$ containing the eigenvalues and $\underline{\Psi}$ consisting of the eigenvectors $\underline{\psi}_0$.

A very convenient way to scale the mode shapes is the *mass-normalization*:

$$\underline{\underline{\Phi}}^T \underline{\underline{M}} \underline{\underline{\Phi}} = \underline{\underline{I}} \quad (2.6)$$

where $\underline{\underline{I}}$ represents the identity matrix and $\underline{\underline{\Phi}}$ the mass-normalized eigenvector matrix, which can be derived with the modal mass matrix $\underline{m}_r$, where m_r is the modal mass of mode r :

$$\underline{\underline{\Phi}} = \underline{\underline{\Psi}} \sqrt{\underline{m}_r} \quad (2.7)$$

$$\underline{\underline{\Psi}}^T \underline{\underline{M}} \underline{\underline{\Psi}} = \underline{m}_r \quad (2.8)$$

For turbomachinery rotors, the mode shapes can be understood as a combination of isolated mode shapes of the blades and the isolated hub disk with varying contribution of the two parts to the combined mode shape of a bladed disk.

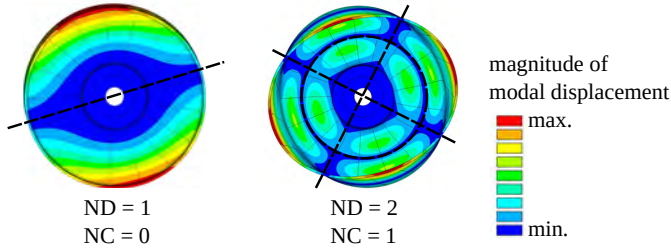


Figure 2.1: Exemplary mode shapes from a FE simulation of the hub disk (blades removed) of a compressor rotor blisk investigated in the present work (ITSM-ASB1) with distinctive nodal diameter and nodal circle patterns.

Solely considering a disk for a moment, the mode shapes show distinctive patterns of regions with no displacement, which are referred to as nodal diameter (ND) and nodal circles (NC), see for example fig. 2.1. For a perfectly symmetric disk, the maximum number of nodal diameter and circles is infinite. The natural frequencies increase with an increase in nodal circles for a constant number of nodal diameter [117]. Furthermore, nodal circles are used to group disk mode shapes, where all modes with different numbers of nodal diameter but the same number nodal circles correspond to the same mode family. In general, the natural frequencies increase within a mode family with an increasing number of nodal diameter.

By the introduction of blades to the disk to form a bladed disk¹, the structure becomes a cyclic symmetric system and the maximum number of nodal diameter is limited by the number of blades N :

$$n_{nd,max} = (N)/2 \quad (\text{for an even number of blades}) \quad (2.9)$$

$$n_{nd,max} = (N - 1)/2 \quad (\text{for an odd number of blades}) \quad (2.10)$$

The normal modes always exist in pairs of two real modes (*double modes*), except for modes with no nodal diameters and, for blisks with an even number of blades, modes with the maximum number of nodal diameter. For these

¹There are multiple options to mount blades to a hub disk, for example by using fir-tree or dovetail attachments or by welding the blades to the hub. Furthermore, the blades can be manufactured as an integral part of the disk (blisk = blade integrated disk).

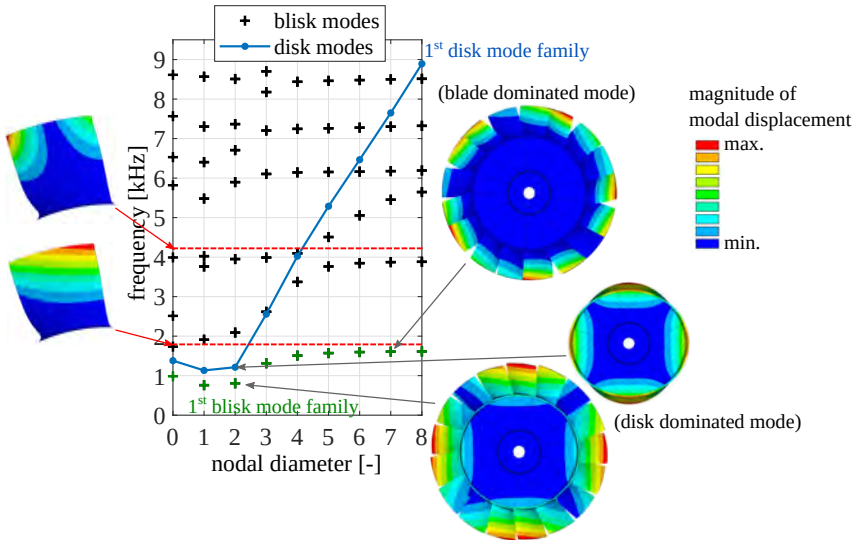


Figure 2.2: Natural frequency vs. number of nodal diameter for a blisk (ITSM-ASB1 at 16200 rpm, FE simulations) and the corresponding isolated hub disk along with natural frequencies of an isolated blade.

cases, always just one real mode exists. Both real modes of a pair share the same value for the natural frequency and the same mode shape but with a different, linear independent orientation to each other. For an ideal blisk, the orientation of the nodal diameter patterns is arbitrary with respect to the geometry and linear combinations of both eigenvectors are also solutions of eq. (2.1). For real systems, even small imperfections in the manufacturing of blisks (*mistuning*) break the cyclic symmetry and the orientation of the modes becomes fixed to the geometry². [118, 125]

A very common depiction of the natural frequencies and number of nodal diameters of a blisk is shown in fig. 2.2. Furthermore, the first mode family

²This is important when results of an experimental modal analysis of a real blisk are compared to simulation results of an ideal geometry, where the orientation is arbitrary and should be rotated to match the mode orientation of the experiment, see also chapter 7.

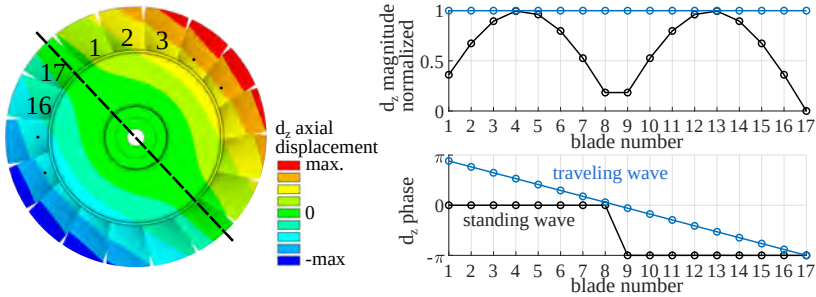


Figure 2.3: Left: Instantaneous axial displacement of a first bending mode with one nodal diameter line; Right: Difference in magnitude of the axial displacement and phase for a standing and a traveling wave mode, evaluated at the leading edge of the blade tip.

of the hub disk with removed blades is depicted along with the first two natural frequencies of an isolated blade³. The first mode family of the blisk in green shows an asymptotic trend of its natural frequencies towards an upper frequency limit with increasing nodal diameters, which is the isolated blade natural frequency. For blisk modes with a low number of nodal diameters, the mode shape of the hub disk is dominating the blisk mode and the blades mostly act as rigid bodies following the disk mode. In this region, the natural frequencies of the disk mode and the corresponding blisk mode are close. Towards higher numbers of nodal diameter, the disk becomes 'stiffer' and its natural frequencies increase significantly. The first mode family of the blisk does not show the same trend as the disk due to the attached blades, which dominate the blisk mode in this region with their first natural frequency. The natural frequency of the blisk converges to the blade frequency and most of the mode displacement is concentrated in the blades. The stiffness of the hub disk and the blades as well as the mass ratio of both parts significantly influence the resulting normal modes of blisks, further discussion on this topic can be found in Corral [39] or Keller [117].

³fixed at the blade root

The double modes can be either excited with spatially stationary or rotating forces, which will result in standing or traveling wave modes, respectively. Figure 2.3 illustrates this for a mode with one nodal diameter. The natural frequencies are not affected by different excitation methods⁴. In the case of a fixed excitation, as for example for an experimental modal analysis, the modes respond with a standing wave form, where all blades between two nodal diameter lines share the same phase. At the nodal diameter lines, a phase change of π occurs. For the mode with one nodal diameter in fig. 2.3, half of the rotor blades are in phase while the phase of other part differs by $-\pi$. As the orientation of the nodal diameter line is fixed for a standing wave, a reduced axial displacement magnitude for blades close to the nodal diameter line is observed. This is in contrast to the traveling wave mode, where the nodal diameter line rotates with respect to the geometry and all blades share the same displacement magnitude. The traveling mode wave follows the excitation and its direction is hence determined by the direction of rotation of the excitation force. In case of a traveling wave mode, the phase difference σ (interblade phase angle) between the blades depends on the number of nodal diameter lines as well as the number of blades and the direction of rotation:

$$\sigma = a \cdot \frac{2\pi \cdot n_{nd}}{N}, \quad a = \begin{cases} -1 & \text{(forward traveling wave (FTW))} \\ 1 & \text{(backward traveling wave (BTW))} \end{cases} \quad (2.11)$$

It is common to correlate the direction of mode rotation to the direction of the blisk rotation instead of the force direction. This is due to the fact that in the operation of turbomachinery, the direction of the mode exciting force pattern can change its direction, see section 2.3.2, but the direction of rotation of the shaft is usually constant. Thus, the nodal diameter lines of a forward traveling wave mode rotate in the same direction as the blisk and vice versa for a backward traveling wave mode. As a consequence, a forward traveling wave mode will result in a negative interblade phase angle according to eq. (2.11). This means that by observing two neighboring blades, the blade further in the direction of rotation will have a lower phase value. Referring to

⁴At this point, a system without damping is considered and hence the response frequency is equal to the natural frequency. Furthermore, Coriolis forces are neglected here.

the example in fig. 2.3, the phase distribution of the traveling wave shows a negative gradient ($\sigma < 0$) and therefore represents a forward traveling wave⁵.

In order to calculate the instantaneous displacement of a traveling wave mode, the double mode with two real modes is considered as a complex mode with one mode of the pair as the real and the second mode as the imaginary part:

$$\tilde{\phi}_{12} = \phi_1 + i\phi_2 \quad (2.12)$$

This complex mode is then rotated with the respective natural frequency ω in time and the real part yields the instantaneous mode displacement:

$$\underline{d}_{12}(t) = \Re(\tilde{\phi}_{12} \cdot e^{i\omega t}) \quad (2.13)$$

The direction of rotation can be changed by changing the sign of ω or by swapping ϕ_1 and ϕ_2 . If $\tilde{\phi}_{12}$ is real ($\phi_2 = 0$), eq. (2.13) will provide the time-dependent displacement of a standing wave.

2.2 Damped, Forced Vibration

A modal analysis of an undamped vibrating system provides information to describe the free vibration of real systems to a certain extend. However, this is still a theoretical consideration and damping has to be introduced to model real systems. Moreover, turbomachinery rotors are exposed to various sources of excitation and damping, which necessitate the consideration of a damped, forced vibration system to describe the vibration sufficiently. Hence, eq. (2.1) is extended with the product of a damping matrix $\underline{\underline{C}}$ and the time derivative of the displacement vector $\dot{\underline{x}}$ (assuming viscous damping, see section 2.2.1) along with the time-dependent force vector $\underline{F}(t)$ on the right-hand side of the equation:

$$\underline{\underline{M}}\ddot{\underline{x}} + \underline{\underline{C}}\dot{\underline{x}} + \underline{\underline{K}}\underline{x} = \underline{F}(t) \quad (2.14)$$

⁵There exist different definitions for the sign of the interblade phase angle in literature. Some authors use the sign to illustrate the direction of rotation (negative for backward traveling), which reverses the definition in eq. (2.11).

Assuming a purely harmonic excitation

$$\underline{F}(t) = \underline{F} e^{i\omega t} \quad (2.15)$$

and a harmonic response (c.f. eq. (2.2)), eq. (2.14) becomes

$$(\underline{K} + i\omega \underline{C} - \omega^2 \underline{M}) \underline{X} = \underline{F} \quad (2.16)$$

The forcing on the right-hand side of eq. (2.14) originates largely from various aerodynamic phenomena present in turbomachinery and will be discussed in section 2.3. The damping matrix can consist of contributions from various different damping mechanisms. In general, they can be categorized in material, structural (or mechanical) and aerodynamic damping [120].

Material damping is present in every vibrating structure. For metals, one of the main mechanisms of material damping is thermoelasticity on micro and macro scales, where kinetic energy is dissipated by internal friction causing an increase in material temperature [14]. The amount of damping is a material property and can vary significantly for different materials, composites and alloys [14, 93].

Structural or mechanical damping is mainly associated with friction between different parts. For bladed disk assemblies with individual blades mounted to the rotor disk, friction due to relative motion at the boundaries provides damping in case of vibration. Since friction can be a very effective damping mechanism, there are numerous designs to introduce additional friction to turbomachinery rotors, see for example [49, 86, 110, 188, 198, 199]. While friction dampers are beneficial in terms of vibration, their use causes costs of additional parts, complexity and weight along with aerodynamic efficiency losses, if the damping structures are placed within the flow path [26, 101].

Estimations from open literature for damping values of material, structural and aerodynamic damping are provided in table 2.1. It can be assumed in general that $\zeta_{\text{struc}} > \zeta_{\text{aero}} > \zeta_{\text{material}}$, but there are exceptions for example in case of flutter, where $\zeta_{\text{aero}} < 0$, see section 2.3. For blisks, as investigated in the present work, structural damping can be neglected due to the integral design of the rotor and only material and aerodynamic damping are relevant.

Different damping models are introduced in the following to provide an overview of how the various physical damping mechanisms can be modeled.

Table 2.1: Overview of estimations for aerodynamic, material and structural damping of turbomachinery rotors in general and turbines. Damping values are provided as critical damping ratios ζ , c.f. eq. (2.21) and eq. (2.25). Structural damping implies friction and friction dampers.

ref., type	ζ aerodynamic	ζ material	ζ structural
[194], general	$1 \times 10^{-3} \dots 3.1 \times 10^{-3}$	$0.5 \times 10^{-4} \dots 1.7 \times 10^{-4}$	$2 \times 10^{-4} \dots 3.3 \times 10^{-2}$
[54], general	$1 \times 10^{-3} \dots 1 \times 10^{-2}$	$\sim 2 \times 10^{-4}$	$5 \times 10^{-3} \dots 3 \times 10^{-2}$
[198], turbines	$0 \dots 1 \times 10^{-3}$	$< 5 \times 10^{-4}$	$2 \times 10^{-4} \dots 2 \times 10^{-2}$

2.2.1 Viscous Damping

The viscous damping model is widely used to describe the dampening influence of fluid on a vibrating structure by assuming that the damping force is proportional to the vibration velocity.

The simplest way to describe the influence of damping on a vibrating system is to consider the free decay of SDOF mass-spring-damper model with the following equation of motion:

$$m\ddot{x} + c\dot{x} + kx = 0 \quad (2.17)$$

The damper is modeled in this example as a viscous damper, with a damping linearly proportional to the vibration velocity: $c\dot{x}$. Assuming a harmonic system with a complex natural frequency, the time-dependent complex displacement of the SDOF model can be described according to [59] as

$$x(t) = X e^{-\omega_0 \zeta t} \cdot e^{i\omega_d t}, \quad (2.18)$$

with the complex displacement amplitude X , the natural frequency of the undamped system

$$\omega_0 = \sqrt{\frac{k}{m}}, \quad (2.19)$$

the oscillation frequency of the damped system

$$\omega_d = \omega_0 \sqrt{1 - \zeta^2}, \quad (2.20)$$

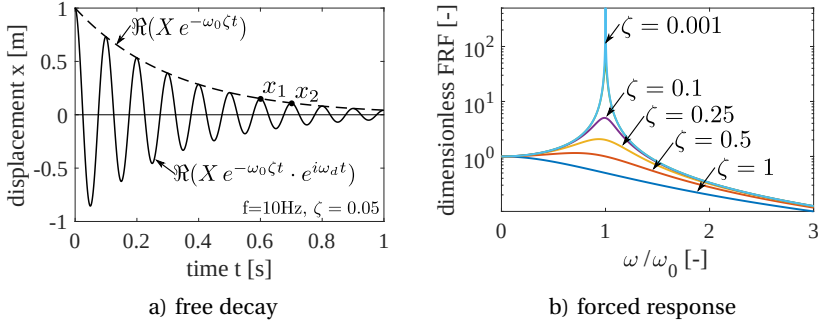


Figure 2.4: Examples of spring-mass-damper SDOF systems

and the critical damping ratio

$$\zeta = \frac{c}{c_{\text{crit}}} = \left(\frac{c}{2\sqrt{km}} \right). \quad (2.21)$$

Figure 2.4a depicts the free decay of the described SDOF system according to eq. (2.18). It becomes apparent how the first part of the right-hand side of eq. (2.18) is responsible for the decay in displacement amplitude while the second part represents the harmonic oscillator. The summation of both parts leads to the characteristic curve of a spring-mass-damper system with a viscous damper and an initial displacement at $t = 0$. The decay rate is defined by $-\omega_0 \zeta$. The critical damping ratio can also be determined by evaluating two consecutive displacement peaks, for example x_1 and x_2 in fig. 2.4 [84]: First, the logarithmic decrement δ is calculated

$$\delta = \ln \frac{\Re(x(t))}{\Re(x(t + \frac{2\pi}{\omega_0}))} = \ln \frac{x_1}{x_2} \quad (2.22)$$

which can then be related to the critical damping ratio

$$\delta = \frac{2\pi\zeta}{\sqrt{1-\zeta^2}}, \quad 0 \leq \zeta < 1. \quad (2.23)$$

For small values⁶ of ζ , eq. (2.23) simplifies to [84]:

$$\delta = 2\pi\zeta, \quad \zeta \ll 1 \quad (2.24)$$

There are other definitions of damping used in literature, such as the loss factor η , the quality factor Q or the specific damping capacity ψ . For small damping values and at resonance conditions, the various damping definitions can be interrelated [84, 217]:

$$\zeta = \frac{\delta}{2\pi} = \frac{1}{2Q} = \frac{\eta}{2} = \frac{\psi}{4\pi} \quad (2.25)$$

In order to model the forced vibration, a harmonic force is added to the right-hand side of eq. (2.17). The frequency response function for the SDOF model in case of a forced harmonic excitation can be described in a non-dimensional manner as [59, 84]:

$$|H(\omega)| = \frac{1}{\sqrt{\left[1 - \left(\frac{\omega}{\omega_0}\right)^2\right]^2 + \left(2\zeta\left(\frac{\omega}{\omega_0}\right)\right)^2}} \quad (2.26)$$

In fig. 2.4b, the evaluation of eq. (2.26) is shown for different values of ζ . The curves can be interpreted as amplitude amplification factors for the SDOF system with a constant excitation amplitude and a varying relative excitation frequency. It can be noted that for small values of ζ the resonance frequency ω_d approximately corresponds to the natural frequency ω_0 , c.f. eq. (2.20).

Rayleigh Damping

For a MDOF system, the stiffness matrix $\underline{K}$ and the mass matrix $\underline{M}$ can be constructed using material properties (Young's modulus and density), but there is no similar concept for the damping matrix $\underline{C}$. To simplify the description of the damping matrix, an assumption can be made that the distribution of damping is proportional to the stiffness and mass matrix and hence, two

⁶small vales of ζ are defined in the present work as $\zeta < 0.1$, which is a valid assumption for turbomachinery applications (c.f. table 2.1) and in particular for compressor blisks, where the summation of the different damping mechanisms still results in $\zeta_{\text{overall}} < 0.1$

scalar values are sufficient to construct a damping matrix for an MDOF system with viscous damping [145]:

$$\underline{\underline{C}} = \alpha \underline{\underline{M}} + \beta \underline{\underline{K}} \quad (2.27)$$

Subsequently, the critical damping ratio can be related to α and β with

$$\zeta = \frac{\alpha}{2\omega_0} + \frac{\beta\omega_0}{2} \quad (2.28)$$

For turbomachinery applications, often the participation of the mass matrix is neglected with $\alpha = 0$ [129, 153, 219]. Furthermore, Rayleigh damping does not change the mode shapes, which simplifies the solution of forced response problems significantly [145]. A Rayleigh damping model is implemented in most FE solvers and it is often used to model the aerodynamic damping or the combined aerodynamic, structural and material damping in forced response simulations.

2.2.2 Friction Damping

While a viscous damper is suitable to model for example the effects of aerodynamic damping, the damping force due to dry friction shows a different behavior and is not proportional to the vibration velocity but rather constant during a vibration cycle. The damping model for dry friction is also known as *Coulomb damping* and the free decay of a SDOF system with Coulomb damping will result in a linear decrease in amplitude as opposed to the exponential decay of a viscous-damper SDOF system. [37, 61]

For further information on the application of Coulomb damping in turbomachinery, also considering nonlinear systems, see for example Laxalde & Throuverez [136] or Krack & Throuverez [130].

2.2.3 Hysteresis Damping and Energy Method

Hysteresis damping models are often attributed with material damping as hysteresis loops are formed if the excitation force and the responding displacement are presented together in one diagram. However, such diagrams

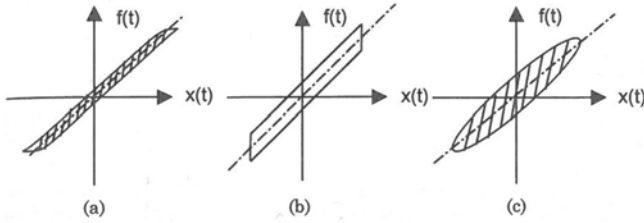


Figure 2.5: Force vs. displacement characteristics for (a) material damping, (b) dry friction and (c) viscous damping, from [59]

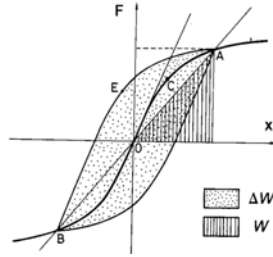


Figure 2.6: Hysteresis of a damping cycle shown in a force vs. displacement diagram from [24]

can be created for various damping mechanisms. Figure 2.5 depicts hysteresis loops for three different mechanisms according to Ewins [59]. While the hysteresis of the viscous damper shows an elliptical shape, other mechanisms clearly differ from this idealized shape. The main advantage of hysteresis models is that these diagrams can be created from experimental data regardless of any nonlinearities. Subsequently, the *energy method* can be applied to determine the damping. Therefore, the energy dissipated within one vibration cycle ΔW (or W_{cycle}) is determined, which corresponds to the enclosed area of the hysteresis loop, see fig. 2.6. In order to calculate the specific damping capacity ψ , W_{cycle} is normalized by a reference energy W_{ref} , which

can be defined in various ways, for example as the total strain, potential or kinetic energy [84].

$$\psi = \frac{W_{\text{cycle}}}{W_{\text{ref}}} \quad (2.29)$$

The specific damping capacity can be translated into a critical damping ratio according to eq. (2.25)

$$\zeta = \frac{\psi}{4\pi} = \frac{W_{\text{cycle}}}{4\pi W_{\text{ref}}} \quad (2.30)$$

and ζ can then be used in a viscous damper system to model the damping of the hysteresis damper. But this implies that for example a realistic, rather complex damping system, which by far does not show an ideal viscous damper hysteresis, is approximated by a viscous damper with the same W_{cycle} . Furthermore, attention has to be paid when eq. (2.30) is applied as this assumption is only valid at resonance. Considering for example dry friction, which has a constant damping regardless of the vibration frequency, a viscous damper approximation of the friction hysteresis would lead to significant errors away from the resonance since viscous dampers show a frequency dependency of W_{cycle} [59].

The energy method is also applied for the determination of aerodynamic damping in unsteady CFD simulations, see section 2.4.4.

2.3 Aeroelasticity

As mentioned in the beginning of this chapter, the influences of steady and in particular unsteady aerodynamics play a key role in the characterization of turbomachinery vibration and aeroelasticity combines relevant aspects of aerodynamics and mechanics to an interdisciplinary field of research. Figure 2.7 shows the Collar triangle representing aeroelasticity as the connection of aerodynamic, inertial and elastic forces. In order to extend the equation of motion for aeroelasticity, a harmonic aerodynamic force $\underline{F}_{\text{aero}}$ is introduced on the right-hand side of eq. (2.16), forming the aeroelastic equation [215]:

$$(\underline{\underline{K}} + i\omega\underline{\underline{C}} - \omega^2\underline{\underline{M}})\underline{\underline{X}} = \underline{\underline{F}}_{\text{ext}} + \underline{\underline{F}}_{\text{aero}} = \underline{\underline{F}}_{\text{ext}} + \underline{\underline{F}}_{\text{aero.damp.}} + \underline{\underline{F}}_{\text{aero.excit.}} \quad (2.31)$$

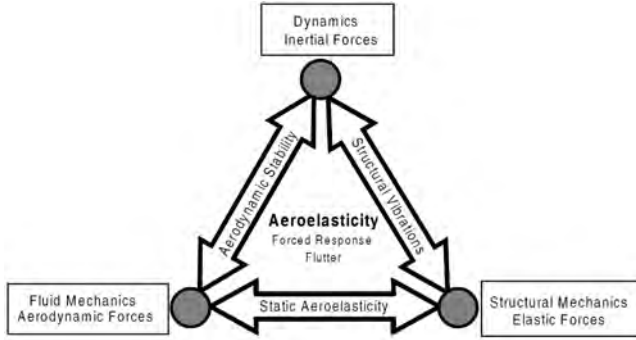


Figure 2.7: Collar triangle, from [112]

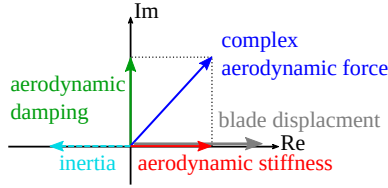


Figure 2.8: Aerodynamic force in the complex plane, adapted from [167] and [216]

Equation (2.31) describes the general form of forced response for turbomachinery with consideration of aeroelasticity. For reasons of completeness, an external force $\underline{F}_{\text{ext}}$ is also implemented, which summarizes external mechanical excitation sources for example from the gearbox, bearings or due to shaft imbalances. In this section, external forcing shall be neglected and only aerodynamic excitations are considered, thus $\underline{F}_{\text{ext}} = 0$. Nevertheless, external forcing becomes important in the present work when the electromagnetic excitation system is introduced in section 5.4. The aerodynamic force can be further distinguished between damping components $\underline{F}_{\text{aero.damp.}}$ and aerodynamic excitation forces $\underline{F}_{\text{aero.excit.}}$. $\underline{F}_{\text{aero.damp.}}$ represents the aerodynamic forces induced by blade vibration. Since a harmonic blade vibration is assumed, the resulting aerodynamic forces are also harmonic and the complex force vector $\underline{F}_{\text{aero.damp.}}$ can be further separated into a real and an imaginary

part, see fig. 2.8. The imaginary part of the aerodynamic force can be interpreted as a damping component, where the sign of the imaginary part determines the type of damping (positive or negative). Furthermore, the real part can be interpreted as an additional stiffness or mass (inertia) introduced by the aerodynamic force, depending on the sign of the real part. In fig. 2.8, the displacement vector only has a real component for illustration purpose. To generalize the statement it can be assumed that the in-phase and -180° parts of the complex damping force add to the stiffness and mass (*added mass effect*), respectively, and the out-of-phase part adds to the damping [35, 167]. Applying this interpretation of the aerodynamic forces of the vibrating blades to the aeroelastic equation, eq. (2.31) can be written as:

$$\left((\underline{K}_{\text{struc}} + \underline{K}_{\text{aero}}) + i\omega(\underline{C}_{\text{struc}} + \underline{C}_{\text{mat}} + \underline{C}_{\text{aero}}) - \omega^2(\underline{M}_{\text{struc}} + \underline{M}_{\text{aero}}) \right) \underline{X} = \underline{F}_{\text{aero.excit.}} \quad (2.32)$$

$\underline{F}_{\text{aero.damp.}}$ is separated in eq. (2.32) into aerodynamic contributions to the mass, stiffness and damping matrices (labeled with *aero*). The parts are subsequently shifted to the left-hand side and added to the members of the corresponding structural matrices (*struc*). A damping matrix for material damping is also added ⁷. The real parts of the damping force can also be concentrated in the stiffness matrix $\underline{K}_{\text{aero}}$, c.f. [153]. Since the aerodynamic mass and stiffness contributions are minor compared to their corresponding structural parts, often only the aerodynamic damping matrix $\underline{C}_{\text{aero}}$ is considered in the aeroelastic equations while neglecting the real part of the aerodynamic damping force.

Equation (2.32) is a general form of the aeroelastic equation and describes the aeroelastic forced response problem. The solution of the equation will yield the displacement shape under aerodynamic excitation. The different sources of $\underline{F}_{\text{aero.excit.}}$ are categorized and discussed in the sections below. In case of flutter, which is considered a self-exciting vibration phenomenon, the aerodynamic excitation force $\underline{F}_{\text{aero.excit.}}$ is neglected [215], and eq. (2.32) reduces to a stability problem, where the summation of the different damping sources determines the flutter stability. Equation (2.32) is written in physical

⁷In eq. (2.32) structural, material and aerodynamic damping are all modeled as viscous damping. While this assumption is often made (for example in [153]), there are also other models used in literature for structural and material damping, see for example [224]

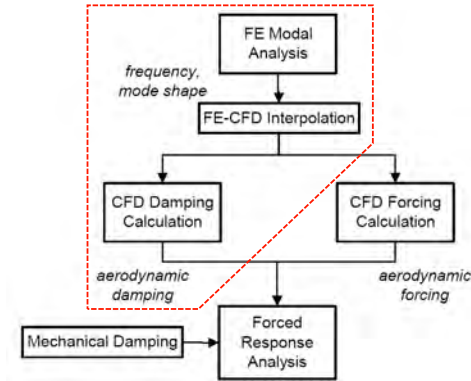


Figure 2.9: Decoupled forced response method from Moffat & He [166]. The Main focus of the numerical part of the present work highlighted in red.

coordinates and a transformation to modal coordinates is often performed, where the assumption is made that the vibration can be described as a combination of a finite number of normal modes [35, 193]. While the modal matrices of stiffness and mass are diagonal for the structural part, the aerodynamic damping matrix can be fully populated, which does not allow for a straightforward decoupling of the aeroelastic equations in modal coordinates. Decoupling can be realized by neglecting the aerodynamic stiffness coupling and applying for example viscous Rayleigh damping for the aerodynamic damping, which results in a diagonal modal damping matrix $\underline{2\omega_r\zeta_r}$, with the individual natural frequency and damping ratio for each mode r on its diagonal [20]. For the consideration of a single mode, the decoupled aeroelastic equation reduces to a SDOF equation [35, 166].

In order to efficiently solve the aeroelastic forced response equation and thus calculate displacement amplitudes for the excitation of a specific mode, Moffat & He [166] proposed a decoupled approach, where the structural and the aerodynamic parts of aeroelastic equation are determined separately, see fig. 2.9. For the structural part, the normal mode of the rotor is determined with a classic modal analysis using a FE solver. The aerodynamic forcing is estimated with an unsteady CFD simulation. The aerodynamic damping

can be estimated as well with a separate unsteady CFD simulation or determined experimentally. Structural and mechanical damping are commonly estimated with experiments. The determination of the aerodynamic damping with CFD and experiments is the main focus of the present work and will be discussed in section 2.4. The advantage of the decoupled approach is that established FE and CFD solvers and methods can be applied separately, which is beneficial in terms of computational time. However, simplifications are required for the decoupled approach such as assuming that the resonance frequency is identical to the natural frequency. Besides, it is assumed that there is no modification of the mode shape due to aerodynamic forces and no mode coupling since an isolated normal mode shape is considered. This assumption is valid for turbomachinery rotors and in particular for compressor and turbine blisks with high stiffness and high mass ratio (=blade mass / mass of surrounding air, c.f. [215]), but can be less acceptable for large fan blades with lower stiffness [35, 36].

There also exist coupled methods, where the FE and CFD equations are solved simultaneously or in an iterative manner. The blade surface thereby acts as the boundary where fluid forces influence the blade displacement and vice versa, see for example [35, 167, 173]. Coupled methods compensate for the disadvantages of the decoupled methods mentioned above but at the cost of significantly increased computational time. Moreover, multiple simulations can be required to find the resonance frequency and thus the point of the highest (most critical) displacement amplitude. This is due to the fact that the exact resonance frequency is not known a priori but rather is a result of the simulation since the resonance frequency is influenced by damping and aero-structural coupling.

In the following, flutter and forced response as the two main phenomena will be described along with some other aeroelastic phenomena causing blade vibrations.

2.3.1 Flutter

Flutter represents a self-excited vibration phenomenon, which is considered to be non-synchronous with respect to the rotation of the turbomachinery. Thus, the vibration frequency is no integer multiple of the rotational speed.

In the occurrence of flutter, the unsteady pressure field on the surface of a turbomachinery blade transfers energy into the blade vibration causing an increase in vibration amplitude. Flow disturbances often initialize relatively small blade vibrations, which trigger the self-sustaining positive energy transfer loop from the surrounding fluid into the blade. In absence of sufficient structural and/or material damping, flutter can cause severe stress amplitudes compromising the structural integrity of the blade by exceeding yield strength limits and ultimately resulting in low cycle fatigue (LCF). In terms of aerodynamic damping, flutter is understood as negative aerodynamic damping with a negative work per vibration cycle W_{cycle} (c.f. section 2.4). If the resulting overall damping, as the summation of structural, material and aerodynamic damping, is zero, the flutter vibration amplitude tends towards a limit (limit cycle oscillation). While the stresses can be sufficiently low to avoid LCF for limit cycle oscillations, high cycle fatigue (HCF) is still of concern at such conditions. Flutter is mostly observed in turbomachinery stages with long and slender blades such as the fan stage and the last stages of low pressure turbines. Typical flutter vibration modes are the first bending, flap or torsional mode, which are present at relatively low natural frequencies for blades with high aspect ratios (= height / chord length). Flutter represents an aeroelastic instability and often the reduced frequency k is used in the design phase as an indicator of flutter probability:

$$k = \frac{2\pi f l_{\text{chord}}}{w} \quad (2.33)$$

The reduced frequency combines structural and aerodynamic properties with the vibration frequency f , the blade chord length l_{chord} and the relative flow velocity w .⁸ If the reduced frequency falls below a critical value, the occurrence of flutter is very likely. The critical reduced frequency for flutter can vary between 0.1 and 1 [215]. Most commonly, flutter is an off-design phenomenon occurring at higher incidence angles but no clear correlation to incidence angles can be found [194]. Nevertheless, different reasons and aerodynamic conditions for negative aerodynamic damping and thus flutter can be identified. Fans, for example, show distinctive operating areas in their

⁸relative velocity upstream of the rotor for fans and compressors and relative velocity downstream of the rotor for turbines [218]

operation maps (flutter regimes), where the risk of flutter is significant, see [149, 152, 178, 179, 190, 194].

2.3.2 Forced Response

In contrast to flutter, forced response represents a vibration, which is driven by excitation mechanisms originating away from the blade row and which would not sustain if this external forcing would stop. To some extent, forced response excitation is always present in turbomachinery due to the inherent rotor-stator interactions causing multiple forcing phenomena. In general, every flow defect in the stationary frame of reference can potentially induce a periodic force on the rotor due to the change in reference frame. Considering a rotor in a multi-stage setup, the potential field of the upstream and downstream stator rows will induce a non-uniform pressure field in circumferential direction, which will create a periodic pressure variation on the rotor blades and hence a periodic forcing. Furthermore, wakes of upstream rows propagating downstream and, in case of transonic flow, trailing edge shock waves of the stator impinging on the rotor introduce unsteady forces on the rotor as well. Inlet distortions due to crosswind or ground running for jet engines [122], boundary layer ingestion [151, 183] or due to compact installation situations with bended inflow channels also cause periodic excitations of the rotor. Since the excitations originate from the rotor-stator interaction, forced response excitation is always synchronous to the rotational speed and its integer multiples and distinct excitation frequencies are present. They can be related directly to the rotational speed using the definition of excitation (or engine) orders, where an excitation order of one corresponds to an excitation frequency equal to the rotational speed. If excitation frequencies are close or correspond to natural frequencies of the rotor, resonances can occur causing a possibly dangerous amplification of the vibration amplitude.

In order to estimate possible resonance conditions, Campbell [30] introduced a diagram, in which the natural frequencies are plotted against the rotational speed. An example of a Campbell diagram is depicted in fig. 2.10. The excitation orders are displayed along with the natural frequencies, which show a dependency on the rotational speed due to stiffening and softening effects. Every crossing of the natural frequency line with an excitation order

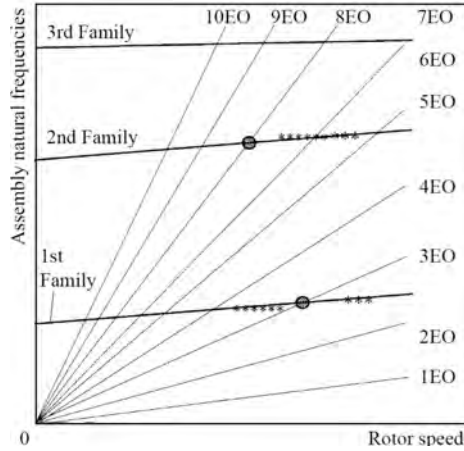


Figure 2.10: Example of a Campbell diagram from [149]

represents a possible resonance point and two of them are shown with circles as examples. There are also flutter regions depicted in fig. 2.10 (asterisks marker) illustrating that flutter also occurs between excitation order lines and that flutter frequencies do (usually) not correspond to excitation orders. Hence, flutter is not driven by forced response excitation mechanisms. In order to fulfill a resonance condition for a forced response excitation, not only the frequencies have to coincide, but also the spatial excitation pattern has to correspond to the mode shape. Hence, not every crossing in the Campbell diagram will lead to resonance. To take the spatial resonance condition into account as well, Wildheim [222, 223] introduced the ZZENF diagram (*zig-zag-shaped excitation line in the nodal diameters versus frequency diagram*). Figure 2.11 represents a typical ZZENF diagram, where the natural frequencies of a rotor and the corresponding number of nodal diameter of each mode is shown. Additionally, Wildheim included excitation lines, which are defined by the following equation:

$$\omega = (jN \pm n_{nd})\Omega \quad j = 0, 1, 2, \dots \quad (2.34)$$

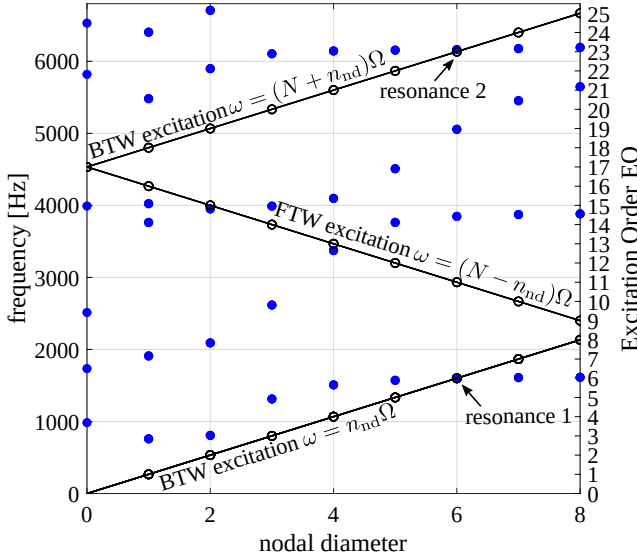


Figure 2.11: Example of a ZZENF diagram for a rotor with an odd number of blades

N is the number of rotor blades, n_{nd} represents the number of nodal diameters, j is an integer value and the resulting excitation frequency is ω .⁹ The ZZENF diagram is created for a distinct angular velocity Ω and thus a distinct rotational speed of the rotor.¹⁰ On the right-hand side of fig. 2.11 ω is translated to excitation orders. Resonance occurs, if the excitation line coincides with a natural frequency, which means that the number of excitation sources corresponds to the number of nodal diameters. This is the case for resonance 1, where an excitation order of six excites a mode with six nodal diameters. The sign of the gradient of the excitation line determines the direction of ex-

⁹the frequency ω and the angular velocity Ω are expressed in $[\text{rads}^{-1}]$ in consistency with Wildheim [222, 223], but the frequency is referred to in $[\text{Hz}]$ in the present work with $f = \omega/2\pi$

¹⁰the nodal diameter n_{nd} is an integer value, see. section 2.1, but is used as a continuous value in fig. 2.11 to visualize the zig-zag lines, c.f. [223]. Equation (2.34) is only valid at integer values of n and these positions are marked with black circles.

citation. A positive gradient represents a backward traveling wave mode excitation and vice versa for a negative gradient¹¹. The absolute value of the gradient is determined by the rotational speed, see eq. (2.34). Due to aliasing effects, excitations with more excitation sources than the maximum number of possible nodal diameter will also excite specific modes, depending on the exact rotational speed and number of sources. See for example resonance 2, where an excitation order of 23 also excites a mode with six nodal diameters. For an odd number of blades, the excitation line is interrupted at the maximum number of nodal diameters, which means for the example in fig. 2.11 that eight and nine excitation sources both will excite a nodal diameter eight mode, but in different directions. Instead of considering the spatial pattern of the excitation sources in the stationary frame of reference, one can also consider the phase difference of the excitation force of two neighboring blades in the reference system of the rotor, which has to coincide with the interblade phase angle σ for resonance.

Forced response is usually attributed with higher values of the reduced frequency k , which means positive values for aerodynamic damping. Hence, the most common failure mechanism for forced response problems is HCF. The magnitude of the excitation force is not constant and depends on the operating point, see for example Kovachev & Vogt [128]. Often, forced response problems can be avoided in the development of turbomachinery by changing the rotational speed in the design point, modifying the natural frequencies by either changing the geometry or the material, or by increasing the meridional distance between the rotor and the excitation source.

2.3.3 Other Aeroelastic Excitation Phenomena

In contrast to forced response and besides flutter, there exist further non-synchronous flow excitation phenomena causing blade vibration. For exam-

¹¹Different definitions of the direction of excitation are used in literature. Wildheim [222, 223] for example uses the direction of the force rotation as reference. In the present work and as already described in section 2.1, the direction of the traveling wave mode is determined with respect to the direction of the blisk rotation. Assuming a rotor passing a stationary excitation force in circumferential direction, a rotating force is introduced to the rotor, which rotates against the rotor direction in the reference system of the rotor. Hence, the definition of the direction in the present work and in many publications is contrary to Wildheim [222, 223]

ple rotating stall, which can cause very strong excitations with an excitation frequency below the rotational speed. While stall flutter is understood as negative aerodynamic damping due to flow separations on all blades in the stall region leading to self-excited vibration, rotating stall is an excitation mechanism itself with rotating stall cells inducing a periodic non-synchronous force to the blades. Nevertheless, both phenomena can occur at the same operating point and can influence each other, see Holzinger et al. [104].

For ducted turbomachinery, acoustic resonance can also play an important role and several experimental and numerical investigations are reported in recent years showing its relevance [29, 65, 99, 105]. In general, there exist acoustic modes in the duct with distinct mode frequencies. These modes can be excited, for example by vortex shedding or flow separations [29, 65, 105], resulting in significant unsteady pressure amplitudes capable to excite blade vibrations if the acoustic frequency is close to a natural frequency of the rotor. Acoustic resonance not only influences forcing but can also modify aerodynamic damping, which will be discussed in the following section.

2.4 Aerodynamic Damping

Aerodynamic damping depends on multiple factors. In the following, the different factors influencing aerodynamic damping are categorized and main factors are mentioned on the basis of various investigations from literature. Subsequently, numerical and experimental methods to calculate and measure aerodynamic damping are introduced.

Three main categories can be identified, which influence the aerodynamic damping of turbomachinery rotors: 1. geometry and material, 2. operating conditions and 3. installation conditions. While some factors within these categories show clear trends of how they influence aerodynamic damping, most factors may be identified to have an impact but no clear correlations can be found. In fact, it is one of the main topics in today's aeroelastic research to find (simple) design guidelines regarding aerodynamic damping.

2.4.1 Influence of Geometry and Material

The geometry along with material properties define the normal modes consisting of the mode shapes and the natural frequencies, both of which have a significant influence on the unsteady pressure field and thus on aerodynamic damping.

For the frequency as well as the reduced frequency, which combines geometrical parameters (frequency and chord length) with an operating condition parameter (fluid velocity), see eq. (2.33), mostly a linear correlation is reported, with an increase in aerodynamic damping for an increasing frequency, see for example [27, 40, 76, 211].

The influence of the mode shape is diverse and can be further separated for blade dominated modes into the influence of nodal diameter and mode family (for example bending or torsional mode family). In the context of flutter investigations, often “S-curves” are computed or measured, meaning the determination of aerodynamic damping for all possible nodal diameter (backward and forward traveling) of one mode family [18, 184, 211, 215]. There are no universal distributions of aerodynamic damping vs. nodal diameter, but for some machine types general observations can be made, as for example for last stage steam turbine rotors, where the aerodynamic damping tends to have a minimum at an interblade phase angle of approximately 90 degree¹² [18, 184]. The characteristic shape of the “S-curves” can be analyzed with the help of the influence coefficient method (see section 2.4.4). It can be shown that a single vibrating blade has constant aerodynamic damping for all nodal diameter and the influence of the neighboring blades introduce sinusoidal variations in the aerodynamic damping with an increasing wavenumber for blades further away from the center blade. Since the magnitude of the influence of the neighboring blades further away decreases fast, mainly the center blade and its direct neighbors determine the distribution of aerodynamic damping for different nodal diameters. With a wavenumber of one for the two neighboring blades, the “S-curve” tends to have a S-like shape [74, 215].

The influence of different mode families is investigated in various ways. For transonic blade rows with high aspect ratios, often the mode displacement in the tip region mainly defines the overall aerodynamic damping

¹²a positive interblade phase angle results in a backward traveling wave, c.f. section 2.1

of the blade row. So-called “tie-dye plots” were introduced in the past, which enable the determination of flutter stability (and thus conclusions on aerodynamic damping) based on the mode shape family of the considered blade section [77, 121, 215, 218]. While these investigations are based on a two-dimensional reduction of the mode shape, more recent publications consider three-dimensional modes and show an influence of the aerodynamic damping with the twist to plunge ratio for jet engine fans [139, 205]. For some cases, relatively small variations in blade geometry can create a significant improvement in aerodynamic damping for multiple modes, as accomplished in Vanti et al. [210], where an aeroelastic optimization process is applied to a compressor rotor showing the general potential for aeroelastic blade design. While the optimization in [210] leads to similar aerodynamic efficiency along with benefits in aerodynamic damping, contradicting results have been reported in Peeren & Vogeler [174], where an aeroelastic optimization was used for a low-pressure turbine rotor. Peeren & Vogeler [174] conclude that “aerodynamic damping and losses/efficiency are competing optimization targets”.

Blade geometry modifications due damages and repair may also significantly change the aerodynamic damping and worsen the aeroelastic performance [92].

A large field of aeroelastic research is also concerned with the influence of mistuning on aeroelastic stability and thus on aerodynamic damping. Mistuning modifies the mode frequencies and mode shapes, see also section 2.5. Various investigations, mostly with focus on flutter stability, show that the influence of mistuning can be beneficial for modes with the lowest aerodynamic damping within a mode family [42, 112, 123, 150, 196]. It can be concluded in general that the spread in aerodynamic damping for one mode family decreases with an increase in mistuning and the aerodynamic damping values converge towards a mean value for different nodal diameter [150]. Furthermore, there are promising studies revealing the possibility to use intentional mistuning in order to increase the aeroelastic stability: [63, 112, 207]. For aerodynamic mistuning, due to changes in blade stagger or altering the blade spacing causing a change in the steady aerodynamic field, different findings have been reported. While some investigations indicate an increase in aerodynamic damping due to

modifications in the blade stagger or blade spacing, others draw opposing conclusions [52, 124, 161, 197].

2.4.2 Influence of Operating Conditions

The following section summarizes effects of the fluid, in which the blades are vibrating, and operating conditions on the aerodynamic damping¹³.

One of the most important parameters influencing aerodynamic damping is the fluid density. It is well established for vibration modes of still standing turbomachinery rotors that air damping linearly increases with air density and hence with air pressure¹⁴ [11, 12, 82, 154]. Regarding aerodynamic damping, similar findings have been made for axial [171] and centrifugal compressors [43, 115, 225]. In each of these studies, the inlet pressure is varied and the aerodynamic is evaluated at the same operating points in terms of rotational speed and volume flow rates. The authors individually conclude that aerodynamic damping linearly increases with inlet pressure and thus with fluid density. The gradient of this linear dependency is specific to each mode investigated. It has to be noted that these authors investigated modes with relatively high reduced frequencies with positive aerodynamic damping and thus positive gradients in the linear density dependency. For modes with negative aerodynamic damping, the gradient also becomes negative, and an increase in density further decreases aerodynamic damping [184, 195]. Moreover, experimental investigations on last stage steam turbines from Rice et al. [184], Heinz et al. [97], Tschürtz [201] and recently on an axial air turbine from Meinzer [156] revealed a linear dependency not only on fluid density but also on the mass flow rate. Due to the definition of the mass flow rate ($\dot{m} = \rho \cdot c \cdot A$) the correlation of aerodynamic damping with mass flow rate includes the dependency on density/pressure and adds a linear dependency on the fluid velocity c .

The dependency on mass flow rate can be considered as a primary factor but there are multiple secondary mechanisms influencing aerodynamic

¹³different operating conditions can also imply different rotational speeds, which can indirectly influence aerodynamic damping by modifications in the natural frequency and mode shape. Nevertheless, in this subsection solely the influence of the flow field on the aerodynamic damping is discussed

¹⁴assuming ideal gas with and a constant air temperature

damping, which can also overcome the general damping influence of mass flow rate. As mentioned in section 2.3.1, there are several publications in the field of flutter research depicting the minimum aerodynamic damping for a complete compressor map. While in general the minimum aerodynamic damping increases with mass flow rate, there are multiple regions of low or negative aerodynamic damping, see for example Schuff & Reisberg [190] or May [152]. These (mostly off-design) regions represent operating points where effects like high incidence, flow separations, shocks or choked blade passages mainly determine the unsteady pressure field of the blades and may cause reduced or negative aerodynamic damping. Furthermore, Reynolds number effects can be observed for low density flow conditions, causing a non-linear dependency of aerodynamic damping on density [73, 85, 190]. Additional non-linear effects may be introduced for blades with high aspect ratios due to a change in static loading and therefore a change in blade twist [190]. This results in modifications of the incidence angle for the same reduced operating points and hence may alter the aerodynamic damping. Basically, every phenomenon causing a change in the steady flow field may also affect the unsteady flow field and ultimately influence aerodynamic damping. Even spontaneous condensation has an impact on aerodynamic damping, as shown by Fuhrer & Vogt [75] for a last stage steam turbine rotor.

2.4.3 Influence of Installation Conditions, Acoustic Resonance and Acoustic Reflection

The category of installation conditions includes all peripheral geometry influencing the aerodynamic damping of a rotor. For many applications, there is no isolated rotor but several rotor-stator stages present in a turbomachinery and in particular the neighboring blade rows can have a significant influence on the aerodynamic damping. The influence of adjacent stages originates from a change in the steady and unsteady flow field of the rotor as well as acoustic reflections of the pressure waves induced by a vibrating rotor [165].

Acoustic reflection is driven by the propagation of acoustic pressure waves in the ducts upstream and downstream of vibrating turbomachinery rotors. In

general, there exist acoustic modes ducts, which, similar to structural modes, show different numbers of nodal diameter and nodal circle lines and different resonance frequencies [81, 146]. If one of these acoustic modes is excited by the unsteady pressure field of a vibrating rotor, the pressure wave propagates without attenuation. This is the case when the excitation frequency is higher than the acoustic cut-off frequency of a corresponding acoustic mode [81, 146]. The case when the excitation frequency is close to the resonance frequency of the corresponding acoustic mode is referred to as acoustic resonance [47, 144]. The cut-off frequency depends on the spatial symmetry of the mode and is hence different for each nodal diameter. Pressure waves with a frequency equal or below the cut-off frequency exhibit an exponential decay in amplitude while propagating (cut-off condition) [81]. The computation of cut-off frequencies is often simplified for 2-dimensional cases or 3-dimensional cases with homogeneous flow fields, examples can be found here [74, 98, 165, 218]. According to [165] or [98], who both refer to [81, 146], the cut-off frequency can be estimated for a duct with uniform flow in radial direction with

$$f_{\text{cutoff}} = \frac{c_s k_m}{2\pi} \cdot \left(\sqrt{1 - \text{Ma}_z^2} \mp \text{Ma}_\theta \right) \quad (2.35)$$

The sign in front of the circumferential Mach Number Ma_θ in eq. (2.35) depends on the direction of the flow velocity with respect to the spinning acoustic mode, which is excited by the structural mode. A positive sign is used if the acoustic mode is spinning in the same direction as the rotation of the flow field (FTW). The wave number k_m can be calculated using Bessel functions, see [98]. It is important to note that for subsonic flow, modes with zero nodal diameter are always cut-on [218]. Reflections of acoustic pressure waves can occur regardless of the cut-on condition, but the exponential attenuation of cut-off modes results in negligible reflection effects for most cases. Nevertheless, there are investigations showing a significant influence of acoustic reflection of cut-off modes, if the spacing between blade rows is small [79].

The influence of a stator row downstream of an axial compressor rotor is investigated by Li & He [143]. The authors show that the aerodynamic damping can change by up to 100 % depending on the rotor-stator gap. Müller et al. [165] investigate the influence of an upstream stator of an

axial turbine and conclude that the aerodynamic damping changes by up to 16% if the axial gap is altered. The authors attribute these changes to modifications in the phasing of acoustic reflections by changing the axial gap between adjacent blade rows. Furthermore, they suggest that the axial spacing between blade rows can be used as an aeromechanical design parameter. In the study of Zhao et al. [226], the influence of acoustic reflections on flutter stability is investigated in a multi-stage compressor. It is shown that in particular the neighboring stator rows have a significant influence on the aerodynamic damping of the investigated rotor. Similar conclusions are drawn in Ekici et al. [53]. Additionally, Zhao et al. [226] ascribe the change in aerodynamic damping to a change in the phasing of the reflected pressure waves, which is in agreement to the findings of Müller et al. [165]. Acoustic reflections not only occur at adjacent blade rows but can also be induced by an abrupt change in volume, for example in the region of a jet engine fan intake. Vahdati et al. [209] and Lee et al. [139] investigate this phenomenon and show that acoustic reflections at the fan intake can cause the so-called *flutter bite*, which represents an area of reduced aeroelastic stability close to the surge line in the compressor map of a jet engine fan rotor. It is concluded in both studies that the phasing of the reflected pressure waves is crucial for the positive or negative effect on aerodynamic damping and that the phase of the reflection can be adjusted with the fan intake length.

Acoustic resonance can also influence aerodynamic damping regardless of reflections from adjacent blade rows or other structures. As presented by Waite [218] for a 2-dimensional cascade of an isolated low pressure turbine rotor, the flutter stability and hence the aerodynamic damping is significantly affected by acoustic resonance for certain interblade phase angles. The author investigates different reduced frequencies and while for some reduced frequencies acoustic resonance has a positive effect on aerodynamic damping, a negative effect is reported for a case with a relatively low reduced frequency. Effects of acoustic resonance are often visible as distinctive spikes in the S-curve (aerodynamic damping vs. interblade phase angle), see also [74, 144, 172].

Besides acoustic reflections and resonance, the radial distance between the rotor tip and the casing has a major influence on the aerodynamic damping of rotors. There are several publications investigating the effect of tip gap size, for example Leichtfuß et al. [141], who show that for a transonic

axial compressor the aerodynamic damping decreases significantly with an increase in tip gap size. Hauptmann et al. [200] present similar findings for an axial turbine as well as Möller et al. [162], where aerodynamic damping was increased with a reduced tip gap compared to the nominal geometry. These investigations might lead to the conclusion that a decrease in tip gap and thus a decrease in tip vortex strength might be beneficial in terms of aerodynamic damping in general. However, there are other investigations, such as Mesbah et al. [159], where contradicting results compared to the aforementioned studies are presented. The authors use CFD simulations to show an increase in aerodynamic damping for the first bending mode of a low pressure axial compressor of 35 % by increasing the tip gap from 0 % to 2 % of the blade span. Numerical results from Besem [15] indicate the existence of an optimum in tip gap size for an axial compressor rotor embedded in a multi-stage compressor, where the aerodynamic damping decreases for lower and higher values of the tip gap. It can be concluded that no clear design guidelines exist for the dimensioning of the tip gap to maximize aerodynamic damping and individual considerations are required. Recently, also the influence of circumferential grooves in the rotor tip region of the casing was investigated for a transonic fan, but no significant effect on flutter stability was discovered [127].

2.4.4 Determination of Aerodynamic Damping with CFD

There are multiple methods to estimate the aerodynamic damping of turbomachinery rotors. One of the most prominent methods in combination with CFD is the energy method, which was originally introduced by Carta [31] in 1967 for an analytic flutter investigation in turbojet engine rotors. Since the energy method is used in the present work as well, an overview of the method is provided below.

The energy method for aerodynamic damping is derived by the energy method for hysteresis damping in general, c.f. section 2.2.3, where the energy of a vibration cycle is related to a reference energy. For aerodynamic damping, the energy transfer between the fluid and the blade structure is considered. Depending on the damping effect (positive or negative), the fluid either extracts energy from the vibrating blade (negative work per cycle) or

transfers energy to the blade (positive work per cycle)¹⁵. In order to compute a damping ratio, the work per cycle of the fluid is normalized by the average kinetic energy of the vibration [31]:

$$\zeta_{\text{aero}} = \frac{-W_{\text{cycle}}}{8\pi W_{\text{KE,ave}}} \quad (2.36)$$

The determination of work per cycle nowadays is realized by conducting CFD simulations, where the harmonic blade vibration is prescribed in the simulations by means of the mode shape of interest. For each investigated mode an individual CFD simulation is required (*traveling wave mode (TWM) method*). Following the decoupled approach from Moffatt and He [166], see also section 2.3, the mode shapes are determined prior to the unsteady CFD simulations by means of a FE modal analysis. Various CFD methods have been applied in the past, such as Euler equation solvers [94, 111], linearized RANS solvers [152, 175], harmonic balance methods [27, 91, 214, 218], unsteady time-marching RANS solvers [74, 92, 214] and others. Subsequently, the work per cycle can be calculated [27]:

$$W_{\text{cycle}} = \int_0^T \int_{\text{blade}} p(t) (\underline{n} \cdot \underline{v}(t)) \, dA \, dt \quad (2.37)$$

In eq. (2.37), the dot product of the local blade force ($p(t) \underline{n} \, dA$) with the local blade vibration ($\underline{v}(t)$) is integrated over the surface of the blade and one vibration cycle T . This formulation is based on the force normal to the blade surface since this force is by far the major contributor to the unsteady aeroelastic work. However, there are studies [208] leading to the conclusion that unsteady viscous forces due to shear stress should also be considered, as it is done for example in [35]. A transformation of eq. (2.37) to the frequency domain can be conducted due to the harmonic blade motion, which allows for the determination of aerodynamic damping with frequency domain based CFD solvers:

$$W_{\text{cycle}} = \Im \left(\pi \int_{\text{blade}} |\underline{h}| \, p_{\tilde{\phi}(p \rightarrow h)} (\underline{n} \cdot \underline{e}_{\underline{h}}) \, dA \right) \quad (2.38)$$

¹⁵The sign of the work per cycle could also be changed to be consistent with the damping effect, but it is common in literature to define a negative work per cycle for positive aerodynamic damping. In this case, a minus has to be used in the calculation of the aerodynamic damping, as it is done in eq. (2.36)

The work per cycle is then computed with the magnitude of the complex blade displacement $\underline{h}$ along with the complex blade force in direction of blade motion ($p_{\tilde{\phi}(p \rightarrow h)} (\underline{n} \cdot \underline{e}_h) dA$). The local force is derived by the product of the unit vector of the blade displacement $\underline{e}_h$ with the complex surface pressure $p_{\phi(p \rightarrow h)}$ and the local surface area dA . The phase of the complex pressure has to be shifted with respect to the phase of the blade motion $\tilde{\phi}$:

$$p_{\tilde{\phi}(p \rightarrow h)} = p e^{-i\tilde{\phi}_h}.$$

The maximum kinetic energy and thus the aerodynamic damping can then be calculated with the natural frequency f of the blade vibration and a scaling factor S_{CFD} , which is required to scale the mass-normalized eigenvectors to a reasonable amplitude for the unsteady CFD simulations:

$$\zeta_{\text{aero}} = \frac{-W_{\text{cycle}}}{8 \pi^3 f^2 S_{\text{CFD}}^2} \quad (2.39)$$

For this decoupled approach, a linear behavior of the unsteady blade forces and the displacement amplitude is assumed. Hence, the work per cycle scales with the square of the maximum displacement, consistent with the kinetic energy, which results in constant aerodynamic damping, independent of the displacement amplitude¹⁶.

Equation (2.39) is used for real modes and in particular in investigations where the influence of the hub on the mode shape is neglected (blade dominated modes). Assuming no influence of the hub, which also implies the assumption of a constant frequency for all modes of one mode family, the mode shapes can be determined by a modal analysis of a single blade fixed at its root. In these (simplified) cases, a traveling wave mode with an arbitrary number of nodal diameter can be created in the CFD simulation by using the same single blade mode shape for all blades of the rotor along with a blade-individual phase-lag according to the nodal diameter (= interblade phase angle). This approach is often utilized for high aspect ratio bladings. In addition, the amount of CFD simulations to determine the work per cycle of individual interblade phase angles for these cases can be significantly reduced by applying the *influence coefficient (IC) method*. In the IC method,

¹⁶It is common to scale the maximum mode displacement to a value between 0.1 % and 1 % of the blade chord length. Very large displacement amplitudes should be avoided since they can introduce nonlinearities in the flow field, such as flow separations due to high incidence, and/or cause numerical instabilities due to invalid mesh elements.

only the center blade is vibrating with a prescribed mode shape and the complex pressure field of a traveling wave mode is created in post-processing by extracting and superimposing the pressure field of the neighboring blades. Different nodal diameter are realized by adjusting the phases of neighboring influences with the corresponding interblade phase angle. The IC method is not utilized in the present work, since mode families are investigated with disk dominated modes. Further information on the IC method and a comparison to the TWM method can be found in Buchwald et al. [27].

For blisk dominated modes, two real modes of a double mode have to be used to correctly construct a traveling wave mode, see eq. (2.12). In this case, where two standing wave modes are superimposed to create a traveling wave mode, the kinetic energy of both modes has to be considered in the calculation of the aerodynamic damping:

$$\zeta_{\text{aero,TWM}} = \frac{-W_{\text{cycle}}}{(8 \pi^3 f^2 S_{\text{CFD,m1}}^2) + (8 \pi^3 f^2 S_{\text{CFD,m2}}^2)} \approx \frac{-W_{\text{cycle}}}{2 \cdot (8 \pi^3 f^2 S_{\text{CFD}}^2)} \quad (2.40)$$

Further information on the consideration of kinetic energy for double modes is provided in appendix A.1.

As already mentioned in section 2.3, it is necessary for the decoupled determination of aerodynamic damping with CFD simulations to assume that there is no influence of the aerodynamic forces on the mode, which can introduce significant errors for large, slender blades with low mass ratio. To overcome this limitation, in particular for jet engine fans or last stage steam turbine rotors, coupled simulations are presented by different authors [108, 138, 207], where the aerodynamic damping is determined with a numerical approach similar to free decay experiments (see section 2.4.5). In general, an initial mode shape displacement is imposed to the rotor blades and then the blade displacement is tracked over time. Using the displacement information, the aerodynamic damping can subsequently be determined using eq. (2.22). A thorough discussion and comparison of the energy method and the coupled free decay method is provided by Chahine et al. [35], who support the use of coupled methods for low mass ratio fans and show that the energy method overpredicts the aerodynamic damping for these cases. However, this approach is not considered for the present work since a compressor blisk with comparably small and stiff blades is investigated.

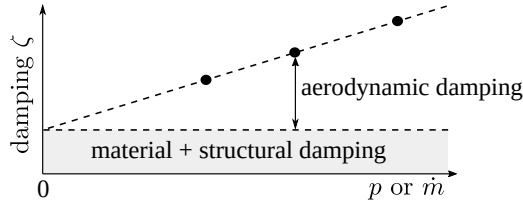


Figure 2.12: Extraction of aerodynamic damping from overall damping measurements using extrapolation. Depiction based on [184]

2.4.5 Determination with Experiments

It is not possible to directly determine the aerodynamic damping of a turbomachinery rotor in operation, because multiple damping mechanisms (material, structural and aerodynamic damping, c.f. section 2.2) are present simultaneously:

$$\zeta_{\text{overall}} = \zeta_{\text{material}} + \zeta_{\text{struc}} + \zeta_{\text{aero}} \quad (2.41)$$

If structural and material damping are assumed to be independent of fluid properties, the linear dependency of aerodynamic damping on fluid density (and mass flow rate) can be utilized to isolate aerodynamic damping from damping measurements. Figure 2.12 exemplifies this method, where the overall damping is measured for three different inlet pressure levels. According to eq. (2.41), the damping value at vacuum conditions represents the summation of structural and material damping, which can be subtracted from all damping measurements in order to obtain the aerodynamic damping. One of the first authors who applied this method to an axial compressor in operation is Newman in 1988 [171] followed by several other authors for different types of turbomachinery and operating conditions (still standing or rotating): [11, 43, 97, 100, 115, 154, 176, 184, 225].

The overall damping can be determined with different methods. In general, two main approaches can be distinguished: Free decay experiments and forced response measurements. Both methods require the measurement of blade displacement, which can be done with blade tip timing (BTT) systems, or the measurement of a signal proportional to the displacement, for exam-

ple with strain gauges or even with the casing pressure as it is shown later in the present work (section 11.5).

Free Decay Method

For free decay measurements, the rotor is excited in a resonance frequency up to a certain displacement amplitude. Subsequently, the excitation is interrupted and the amplitude of the excited mode decays according to the overall damping acting on the rotor. Damping can then be calculated by measuring the displacement amplitude during the decay and applying eq. (2.22). As an alternative, damping could also be estimated by fitting a function in the form of eq. (2.18) to the measurement data, which will provide the damping as part of the fitting solution. The decay method necessitates an additional excitation mechanism, which can be controlled externally.

Half-Power Bandwidth Method

Forced response measurements either can be realized by synchronous excitation due to rotor stator-interaction, see also section 2.3.2, or by non-synchronous excitation using an additional excitation mechanism with the ability to control the excitation frequency independent of the rotational speed. Methods to evaluate the displacement amplitudes of forced response measurements in terms of damping are introduced below.

A straightforward technique to calculate damping from a forced response measurement is the half-power bandwidth method, which is based on the assumption of a SDOF system. As it can be seen in fig. 2.4 b), the width of the amplitude response increases with damping. This characteristic is utilized to estimate damping by evaluating the two frequencies at amplitudes, where the system contains half the power compared to the peak amplitude frequency. These points correspond to $1/\sqrt{2}$ of the peak amplitude, c.f. fig. 2.13. Damping can then be estimated with:

$$\zeta = \frac{f_{\text{high}} - f_{\text{low}}}{2f_{\text{peak}}} \quad (2.42)$$

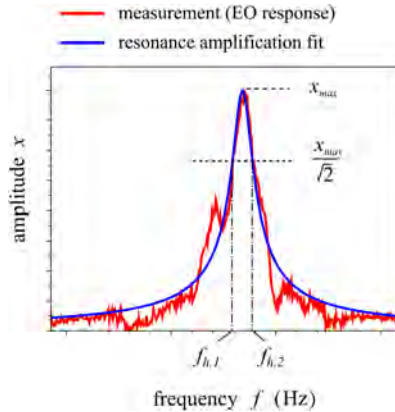


Figure 2.13: Measured forced response amplitude vs. excitation frequency along with half-power frequencies of an amplitude fit, from [89]

A derivation of eq. (2.42) can be found in [33]. It has to be noted, that according to [33] the half-power bandwidth method is only valid for $\zeta < 0.05$, which is sufficient for the consideration of damping in turbomachinery.

Since the half-power bandwidth method is based on a SDOF system, limitations of the method are visible if two mode frequencies are close, in particular for relatively high damping ratios. In this case, the modes are partially superimposed in the FRF spectrum and errors in the damping estimation arise. However, these errors can be estimated, see [220].

FRF Fit Method

Experimental results are often affected by measurement noise, leading to frequency response curves as depicted with red in fig. 2.13. By using the half-power bandwidth method on such noisy data, errors are inevitable. While filters could be used to smoothen the data, the preferred (and more accu-

rate) method is to fit a frequency response function of a SDOF system to the data [97]:

$$|H(\omega)| = \left| \frac{q}{-\omega^2 + \omega_0^2 + 2\zeta\omega_0 i\omega} \right| \quad (2.43)$$

The fitting of eq. (2.43) can be realized using a least-squares fit approach with the parameter ω_0 as the natural frequency, ζ as damping and q as a scalar factor of the input force and its influence on the response with the assumption of a constant excitation force in the small frequency range considered [97]. The frequency ω represents the variable of the function. Since the damping value is a result of the fitting process, no further calculation of the damping is required.

In case of significant mistuning, where multiple modes are participating in a resonance region, more complex system identification methods with the consideration of MDOF systems, as for example presented by Huang et al. [107] or Figaschewsky et al. [62], show very promising results.

2.5 Influence of Mistuning and Coriolis Effect

As discussed in section 2.1, blisks are considered as cyclic symmetric systems. In the ideal or *tuned* case, all sections of the blisk are identical and double modes appear with pairs of identical natural frequencies. In reality, the cyclic symmetry is disturbed by *mistuning* due to geometrical imperfections, inaccuracies in the manufacturing process, inhomogeneities in the material, wear, cracks, or variations in the clamping condition for bladed rotors. As a result, the natural frequency pairs split up and the orientation of the modes and hence the orientation of the nodal diameter lines with respect to the geometry become fixed [59, 118, 125]. The amount of frequency split depends on the strength of mistuning introduced to the structure and also varies for different double modes [118, 219].

Mistuning not only affects the natural frequencies, but also modifies the mode shapes. The clear sinusoidal nodal diameter pattern successively breaks up if mistuning is increased and single modes show contributions of several nodal diameter patterns. In case of strong mistuning, nodal diameter

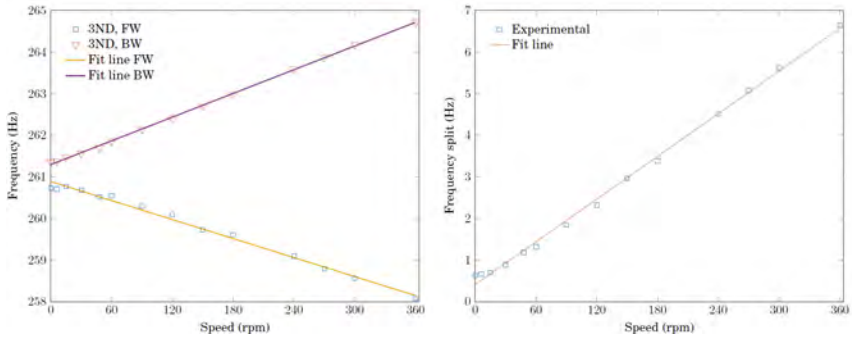


Figure 2.14: Experimental investigation of the Coriolis effect from Ruffini [185]

can no longer be identified and severe localization effects occur, where the vibration energy is concentrated in one or just a few blades leading to increased vibration amplitudes for these blades [125, 219].

Besides mistuning, the Coriolis effect for turbomachinery rotors in operation also causes a frequency split. Coriolis forces appear for rotating structures in vibration, where the vibration displacement has components in radial and/or tangential direction [116]. As a consequence, the two real modes of a double mode both become complex and split up into a forward and a backward traveling wave mode [186]. This effect is impressively shown by Ruffini [185], who performs measurements on an academic blisk excited with piezoelectric elements for different rotational speeds, see fig. 2.14. The measurements show a clear split of a double mode with three nodal diameter into a backward (BW) and a forward (FW) traveling wave. The amount of frequency split is thereby linear with the increase in rotational speed, which is a clear sign of the Coriolis forces causing this split. Since these results are based on experiments, mistuning cannot be avoided completely, as it can be seen at 0 rpm, where a frequency split is already present. While mistuning and the Coriolis effect both split the double mode frequencies, the split due to mistuning is constant and does not change with rotational speed [185]. In fig. 2.14, the constant split can be seen for very low rotational speeds, where the Coriolis effect is negligible. Furthermore, mistuning splits the modes of a double mode resulting in two real modes with different natural frequencies. In

case of excitation and without considering Coriolis effect, the two mistuned modes appear as separated standing wave modes, since their orientation is locked to the geometry. Traveling wave modes may still be observed if the two modes are close to each other and sufficient damping is present to couple both standing wave modes, which lets them appear to be tuned [59]¹⁷. Nevertheless, this is in contrast to the split due to Coriolis forces, where the split creates two complex modes, each representing a traveling wave mode with a different rotational direction. In reality, both effects superimpose as described by Ruffini [185], where two standing wave modes are observed for very low rotational speeds due to mistuning, which transform into traveling wave modes for higher rotational speeds due to an increasing influence of the Coriolis forces. Besides, stress stiffening or spin softening effects due to centrifugal forces also modify the natural frequencies but can be assumed as an additional effect acting equally on both modes and thus do not influence the amount of frequency split significantly. It has to be noted that there are different conventions regarding the direction of the traveling waves depending on the reference frame. In the present work, similar to for example [185, 222], the direction is determined in the rotating frame of reference, resulting in an increase in frequency for backward traveling waves and vice versa for forward traveling waves, c.f. fig. 2.14. A thorough discussion on this topic can be found in [185].

The consideration of the Coriolis effect does not represent solely an academic problem but has relevance to real turbomachinery applications, as it is shown for blisks for example in [3, 83, 219] and later in the present work.

¹⁷Ewins [59] estimates that a split double mode appears to be tuned if the frequency split due to mistuning is less (in percentage with respect to the mean) than the damping of both modes (as percentage of the critical damping)

3 Review of Experimental Research on Vibration and Aerodynamic Damping

An overview of test rigs and experimental research in the field of aeroelasticity is provided in the following chapter with focus on blade excitation and the experimental determination of aerodynamic damping. Regarding the general types of test rigs, two main categories can be distinguished: Test rigs, where the aerodynamic excitation of vibration modes stems from the inherent rotor-stator interaction in turbomachinery and test rigs, where an additional excitation source is implemented, allowing for the excitation of a broader range of modes at multiple operating points.

3.1 Test Rigs with Rotor-Stator Interaction Excitation

A detailed investigation of aerodynamic damping in a three-stage transonic axial compressor is reported by Newman [171] in 1988. He investigates the first bending and the first torsional blade mode of the last-stage rotor using strain gauges. The extraction of the overall damping is realized in the frequency domain by a SDOF curve fit on the measured frequency response curve. Measurements are performed at three different levels of total pressure at the inlet while maintaining the same reduced operating points. This allows for a linear extrapolation of the measured damping to vacuum conditions and the extraction of aerodynamic damping (c.f. section 2.4.5). It is concluded that aerodynamic damping is the largest contributor to the overall damping, even at the lowest inlet pressure condition. The report does not discuss the excitation of the modes in detail, but damping measurements are performed at constant rotational speed and modes are investigated with resonance frequencies of non-integer excitation order values. This leads to the conclusion that the flow field contains sufficient frequency content even to excite non-integer EO frequencies but raises questions regarding the as-

sumption of a constant excitation force, which is required for the SDOF curve fit approach. Nevertheless, the work of Newman represents, to the author's knowledge, one of the first experiments in an operating axial compressor showing the linear dependency of aerodynamic damping on air density by changing the inlet pressure level.

The Purdue three-stage axial compressor represents a prominent test rig for the investigation of aeroelasticity and was initially introduced in 2004 [78]. The compressor is equipped with multiple fast-response pressure transducers as well as tip timing sensors and blisk excitation is provided by the excitation orders of the upstream and downstream stator rows. Murray and Key [170] published a set of measurement data for the second rotor blisk showing that the damping of a first torsional mode doubles at an operating point of high loading compared to nominal loading. Excitation is realized by sweeps of the rotational speed in the resonance region of the mode. Comparisons of damping measurements to CFD simulations are provided by Besem [16], who also points out the significant mistuning of the blisk and the need to implement mistuning in forced response calculations to obtain sufficient agreement to measured blade amplitudes. Subsequently, Huang et al. [107] introduced a system eigenvalue identification method and applied it to the measurement data of the aforementioned mode concluding that aerodynamic damping is overestimated by SDOF curve fitting, if mistuning is dominant. Pressure data obtained in the compressor casing during resonance crossings is used by Murray [169] and more recently by Leng and Key [142] to correlate the unsteady pressure field to blade vibration amplitudes. Processing results from simultaneously acquired blade tip timing data as well, Leng and Key [142] present an impressive agreement between blade deflection and pressure amplitudes and state that unsteady pressure data can be used as a source to monitor blade vibration in a non-intrusive manner.

A resonance crossing with an engine order during the start up is used by Heinz et al. [97] to investigate the aerodynamic damping and vibration amplitudes of a last stage steam turbine rotor with free-standing blades and intentional mistuning. Strain gauges are applied to several blades and the resonance is investigated with different gradients for the increase of rotational speed. Furthermore, the pressure level in the downstream condenser as well

as the mass flow rate is varied and damping is determined using a MDOF curve fit. Due to the variation in pressure level, Heinz et al. [97] are able to extrapolate damping to vacuum conditions and to calculate aerodynamic damping. The authors show that aerodynamic damping linearly correlates not only with the pressure level and thus with the fluid density, but also with the mass flow rate.

Similar measurements of the aerodynamic damping are also performed by Degendorfer [43] for a centrifugal compressor in the ETH test rig RIGI. Degendorfer uses strain gauges to capture blade vibration and investigates resonances due to an interaction of the blisk with a vaned diffuser at different density levels and for air as well as an air- CO_2 mixture with a CO_2 concentration of 46wt%. He concludes that the aerodynamic damping for the air- CO_2 mixture has the tendency to be higher compared to air. Kammerer [114, 115] used the same test rig prior to Degendorfer to determine the aerodynamic damping of a centrifugal compressor with distortion screens upstream of the impeller to create defined excitation orders.

3.2 Test Rigs with an Additional Excitation System

In the following section, test rigs with an additional excitation system are presented. Since a rotating test rig is designed and investigated in the present work, the following overview is focused on similar facilities. A test rig with a rotating rotor offers the possibility to investigate aeroelastic effects in very realistic operating conditions in terms of the flow field as well as the structural loads and vibration modes. Nevertheless, there are multiple aeroelastic investigations of non-rotating cascades published in the last decades. For example the famous standard configurations [22, 68, 69] or the AETR annular cascade at KTH [215]. A thorough review on oscillating cascade experiments is presented in [148]. While cascades represent a simplification by modeling the rotating flow field in the stationary frame of reference, they can offer valuable insight into the flow field of vibrating blade sections due to the easier accessibility. Cascades enable in particular the measurement of the unsteady pressure field on the blades and are thus often used as a first step in the validation of unsteady CFD simulations. Limitations compared to rotating test

rigs are often the investigation of solely rigid body modes as well as uncertainties in the periodicity in particular for linear cascades and sector annular cascades.

3.2.1 Piezoelectric Excitation

Kielb and Abhari [120] present in 2003 the experimental determination of aerodynamic damping for a full-scale rotating axial turbine with a bladed rotor. The test setup consists of a tank containing the turbine and a shock tunnel connected to the containment. The tank can be evacuated to perform measurements in vacuum conditions as well as steady measurements in air for 50 to 100 ms by using the shock tunnel. Piezoelectric patches are mounted on two blades and vibration is measured by means of strain gauges on several blades. The data is post-processed using a SDOF curve fit to estimate damping. Due to the measurements in vacuum, Kielb and Abhari are able to extract the aerodynamic damping from the overall damping measured in air and they can show that aerodynamic damping is the main source of damping for a first torsional as well as a second bending mode of the blades. The investigation represents one of the first measurements of aerodynamic damping for a full scale turbine by means of an external excitation source. However, only two piezoelectric patches are used and significant mistuning is observed, which does not allow for the excitation of traveling wave modes and results in the excitation of isolated blades.

A larger piezoelectric system is used by Gibert et al. [83] to excite an axial compressor blisk of a high pressure stage with 36 blades. While the investigation is conducted in a vacuum spin rig and hence does not allow for the determination of aerodynamic damping, interesting findings are reported with regard to the influence of the Coriolis effect. The blisk is equipped with 24 piezoelectric patches on both sides of the hub and the pattern are shifted against each other to cover the complete circumference of the blisk. Slip rings are utilized to transfer high voltage power to the piezoelectric patches and 12 strain gauges are attached to the blades to measure blade vibrations via a telemetry system. Due to the large number of well positioned piezoelectric patches, the authors are able to excite traveling wave modes. They can show for a mode with four nodal diameter, that a frequency split occurs

for the backward and forward traveling mode, which the authors attribute to the effect of the Coriolis force. As a result of the limitations in maximum rotational speed of the test rig far below design speed, the frequency split is minor but significant nevertheless. Furthermore, a frequency split is also present at rest due to mistuning and both observations are in agreement with the investigations of Ruffini [185] (see also section 2.5). In addition, Gibert et al. [83] demonstrate that a standing wave excitation can excite a traveling wave mode in presence of the Coriolis effect.

3.2.2 Flow Jet Excitation

Excitation of a transonic axial compressor rotor by means of flow jets is done by Crawley [41] in 1983 at the MIT blow down facility to determine aerodynamic damping. The facility allows for 150 ms of steady flow conditions. Pressurized gas through holes in the upstream struts are used to create engine order excitations. The excitation is stopped within 1 ms, which enables the measurement of the free decay response of the rotor with strain gauges. Crawley shows that aerodynamic damping is by far the largest contributor to the overall damping.

A pulsed flow jet excitation system is developed by Wegman et al. [221] within the scope of the FUTURE research project [67] for the TUD transonic axial compressor test rig [103]. The system consists of 15 devices, each containing a rotor with 32 holes and a variable speed to create jet pulse frequencies of up to 3200 Hz by interrupting a flow jet periodically. The maximum excitation time of the system is limited to 60 s due to overheating. The authors show the ability of the system to excite traveling wave modes with up to three nodal diameters and in both directions (BTW and FTW). The system is installed in the casing upstream of a compressor blisk and allows for frequency sweeps as well as free decay experiments. Results of the aerodynamic damping measurements are provided in [67].

A similar system for synchronous and non-synchronous excitation was recently introduced by D'Souza et al. [50] and Kurstak and Souza [133], which is used in a vacuum spin rig. The authors use fast acting valves with a maximum frequency of 1000 Hz and pressurized air to excite traveling wave modes.

3.2.3 Acoustic Excitation

A thorough experimental and numerical study on aerodynamic damping of an axial turbine rotor blisk is conducted by Meinzer [156, 157] utilizing an acoustic excitation system previously developed by Freund et al. [70, 71]. For the excitation system, 16 speakers are positioned in the casing downstream of the rotor creating sound pressure levels of up to 155 dB. Since frequency and phase of the speakers can be adjusted individually, Meinzer is able to excite a large range of different nodal diameter with significant blade displacement amplitudes. The blade displacement is measured by means of a tip timing system. Damping is calculated by using a SDOF fit. Material and structural damping is not measured directly but estimated by an extrapolation of damping measurements to a mass flow rate of zero and can be considered of minor influence to the overall damping. The agreement of the measurements with accompanying CFD simulations is very good for positive nodal diameter, except for the position of the acoustic resonance, which is measured at a nodal diameter of +1 but predicted for a mode with +3 nodal diameter. For negative nodal diameter, the agreement is worse and the standard deviation of the damping measurements is significantly higher, which Meinzer attributes to lower excitation levels. Due to a limited frequency range of the speaker system, the aliasing effect is used to excite negative nodal diameter resulting in lower excitation amplitudes. Furthermore, variations in the rotational speed during the excitation sweep introduce errors in the damping calculation. These errors become more significant for lower sweep rates, which are required to measure the steady blade response. The study impressively shows the possibility of acoustic excitation in combination with a tip timing system to conduct non-intrusive investigations of aerodynamic damping of a turbine blisk in operation.

3.2.4 Magnetic Excitation

The first report of using magnetic excitation was published in 1924 by Campbell [30] and part of the experimental setup is depicted in fig. 3.1. Campbell uses an electromagnet with alternating current (AC) to excite traveling wave modes for a steam turbine rotor. The stationary magnet is placed axially in

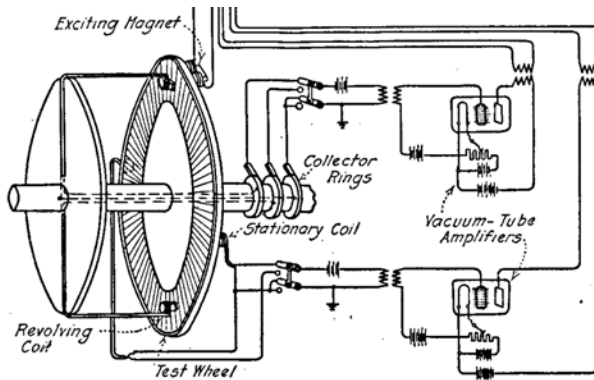


Figure 3.1: Experimental setup for magnetic excitation, from Campbell [30] in 1924

the region of the blades and the speed of the rotor, which resembles a large disk due to the small blades and large hub diameter, can be adjusted individually. The displacement is measured by means of two stationary and two rotating coils, which provide electric current fluctuations proportional to the air gap between the coils and the rotor. Furthermore, this assembly allows for the detection of the traveling wave mode direction. Campbell is able to systematically investigate turbomachinery rotor vibration and clearly identifies and measures modes with different nodal diameter and in both traveling wave directions as well as the dependency of mode frequency on rotational speed. Furthermore, he describes that an increase in frequency of the stationary magnet proportional to the rotational speed is required to excite a mode in a forward traveling manner, if the rotor is rotating, and that the frequency has to be decreased compared to the still standing rotor, for a backward traveling wave of the same mode.

The same principle is utilized by Rice et al. [184] to measure the aerodynamic damping of a last stage steam turbine rotor in operation. The authors place an electromagnet with an E-core in the turbine casing, where ferritic material in the vicinity of the magnet is removed to improve the performance of the magnet. In order to compensate for high temperatures due to high electric currents in the coil, an air supply system is installed along with a ther-

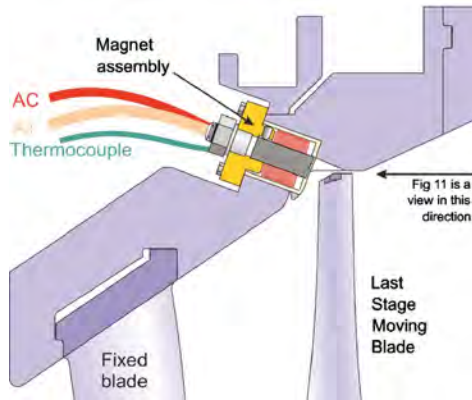


Figure 3.2: Magnet installation of Rice et al. [184]

mocouple to monitor the temperature. Excitation is realized by means of a high voltage square pulse signal, which can be adjusted in amplitude and frequency. Rice et al. perform free decay measurements for several nodal diameters and at two different operating points. Subsequently, mechanical and aerodynamic damping are determined based on strain gauge data by extrapolation of the overall damping to a mass flow rate of zero. Comparisons to CFD simulations show similar trends in the aerodynamic damping for different nodal diameters.

A similar excitation setup was used at the Doosan Skoda Power test facility [17, 18, 177] to also investigate the aeroelastic stability of the last stage of a low pressure steam turbine. Instead of an E-core, a U-shaped¹ laminated magnet core is assembled with two copper coils and placed in the turbine casing facing the shroud of the last rotor. Very high electric currents in the coil of up to 50 A are possible and magnet frequencies of up to 2500 Hz. Due to the high electric currents, a water cooling system is implemented. The rotor is excited by means of sinusoidal frequency sweeps and the excitation frequency is corrected by a control system to compensate for fluctuations in the rotational speed. Strain gauge signals are post-processed with a wavelet-

¹ sometimes also referred to as C-shape

based analysis and SDOF fits in the frequency domain are applied. The authors in [18] conclude that the rotor disk shows low mistuning with variations in vibration amplitude between different blades of less than 15 %. Several operating points and modes are investigated and comparisons to CFD simulations show a good agreement of the minimum aerodynamic damping value. However, the shape of the measured S-Curves cannot be reproduced by CFD, in particular for off-design conditions.

Recently, Doll et al. [46] also successfully used AC electromagnets in the casing of a low pressure steam turbine in order to excite backward traveling wave modes.

3.2.5 Conclusions

In general, all four excitation methods presented above offer the possibility to excite blade vibrations, but with different limitations.

With piezoelectric excitation, reasonably high forces and high frequencies can be realized, depending on the size of the patches and supply voltage. In addition, the excitation works at vacuum. However, all blades or the whole circumference of a blisk hub have to be equipped with piezoelectric patches for traveling wave mode excitation, which can result in significant mistuning and flow disturbances. Furthermore, the power and individually phased excitation signals have to be supplied to the patches in the rotating system, which increases the complexity of the system and introduces additional sources of errors.

Flow jet excitation systems can create significant forces, which can be easily adjusted by increasing or decreasing the flow jet pressure. While large and strong jets are desirable in terms of blade excitation, they may also alter the flow field, which increases uncertainties in the determination of aerodynamic damping, in particular for frequency sweep excitation. Besides, fluid is added, which has to be removed in closed loop test rigs to maintain a steady operating point. Very high excitation frequencies also seem to be challenging since the frequency is limited by the time of the valve switching cycle.

Acoustic excitation decreases the influence on the flow field compared to flow jets, but still, significant pressure fluctuations necessary for sufficient

blade excitation². Interactions of the transient pressure field, induced by the vibrating blades, with the acoustic excitation pressure may occur, which introduce uncertainties. The excitation force can be increased by installing more speakers but there are clear limitations in the maximum force. The frequency range is also limited by the speakers used, but adjustments to the system to change the frequency band seem possible. A main disadvantage of acoustic excitation is that it cannot be used in vacuum.

Electromagnets offer contactless blade excitation, similar to flow jets and speakers, but with the advantage of using an electromagnetic field to transfer the excitation forces instead of using the operating fluid as transfer medium. Hence, there are no disturbances of the flow field as long as the magnets are not placed within the flow channels. Excitation in vacuum is also possible. High excitation forces are realizable, which are proportional to the size of the magnets as well as the number of windings in the coil and the electric current. Nevertheless, very high excitation frequencies are challenging due to magnetic hysteresis effects along with overheating issues, if cooling is not provided. In addition, it has to be considered that very small air gaps between the magnets and the rotor are required for sufficient forcing amplitudes. Obviously, magnetic excitation requires ferromagnetic rotor material, which limits its application.

²see for example Meinzer [156, 157], where a sound pressure level of up to 155 dB is used, which equals to 1125 Pa in sound pressure

4 Objectives and Approach

The motivation of the present work is to create high quality experimental data of aerodynamic damping for the validation of CFD simulations and therefore contribute to the improvement of numerical models as well as to improve the physical understanding of aerodynamic damping. The review of experimental research in chapter 3 shows that while several test rigs exist to measure aerodynamic damping to some extent, uncertainties are present regarding the influence of mistuning, additional structural damping or insufficient excitation. Furthermore, most of the test rigs are limited to the excitation of very few modes and/or the operation at a few operating points. The present work aims at these limitations and presents a test facility with high flexibility regarding mode excitation as well as operating conditions along with state of the art measurement technique.

Based on this motivation, the following objectives are derived:

1. Design and build a test rig along with an excitation system to measure aerodynamic damping for a representative turbomachinery rotor
2. Provide high quality and reliable measurements of aerodynamic damping of several normal modes at different operating conditions and gain physical understanding of aerodynamic damping
3. Validate a CFD model for aerodynamic damping simulations

In order to achieve the objectives, it was decided to rebuilt and extend an existing closed loop axial compressor test rig, which provides good accessibility for sensors and an excitation system. Furthermore, the test rig can be operated at different density levels as well as at vacuum conditions, which enables the extraction of aerodynamic damping.

A subsonic axial compressor blisk, which is designed from scratch for the present work, is chosen as test object in order to minimize structural damping. The structural dynamics of the blisk are measured in detail by means of non-intrusive methods in order to determine mistuning and validate FE

simulations. These measurements are performed in free-free conditions as well as with the blisk mounted in the test rig to identify influences of the clamping conditions on the normal modes.

In operation, the blisk is excited by means of a newly designed electromagnetic excitation system in the hub, which allows for contactless excitation. The blisk vibration is measured by means of a non-intrusive blade tip timing system. Hence, the experimental setup enables the measurement of aerodynamic damping without additional mistuning and without interference with the blade vibration or the flow field. In addition, the blade tip timing system is validated at vacuum using a scanning laser Doppler vibrometer and a derotator to quantify uncertainties in the vibration measurement.

For the measurement of aerodynamic damping, focus is put on the first bending mode family and measurements are performed for the complete compressor map. The calculation of damping is realized with a SDOF fit approach.

Besides damping measurements, detailed steady aerodynamic measurements are conducted in terms of static pressure measurements and pneumatic pressure probes upstream and downstream of the rotor to validate the steady CFD simulations. Unsteady pressure measurements are also performed with high temporal resolution to gain insight in the transient pressure field and the results are used as additional validation data.

5 Experimental Setup and Test Object

In this chapter, the experimental setup is introduced including the test facility, the excitation system and the instrumentation as well as the test object. Furthermore, information is provided on how the test rig and the excitation system is operated and how steady and unsteady measurement data is post-processed.

5.1 Test Facility

The experimental work is conducted at the axial compressor test facility **ACTIVE** (Axial Compressor Test rig for the Investigation of blisk Vibrations with an active Excitation system). The test facility is based on a former axial compressor test rig at the EPFL [132, 137] and is rebuilt at the Institute of Thermal Turbomachinery and Machinery Laboratory (ITSM) of the University of Stuttgart for the purpose of investigating aerodynamic damping as part of the present work. A CAD rendering of the test facility is depicted in fig. 5.1, the main parameters of the facility are summarized in table 5.1. While most of the periphery (piping, heat exchanger, venturi nozzle, compressor outer casing, volute and electric motor) is adopted from the former

Table 5.1: ACTIVE test facility main parameters

electric motor max. power	110 kW
electric motor max. rotational speed	3320 rpm
gear box ratio	1:8.05
compressor max. rotational speed permitted	18500 rpm
total pressure level	5...2500 mbar
total temperature level	20...30 °C
test loop pipe diameter	DN300
compressor shroud diameter	240 mm

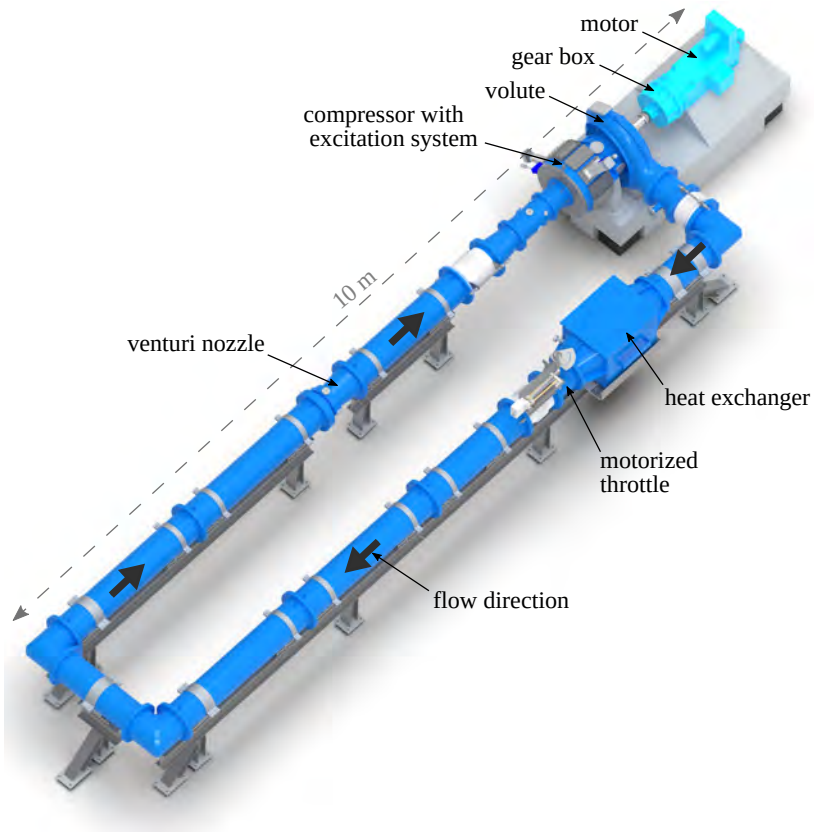


Figure 5.1: ACTIVE, located at the Institute of Thermal Turbomachinery and Machinery Laboratory (ITSM), University of Stuttgart, Germany

setup, the compressor itself as the key part of the facility is designed from scratch. Besides, a new motor power supply is installed to precisely control the rotational speed of the compressor. A state of the art PLC system (Siemens S7-1500 CPU 1512C-1 PN) is implemented along with a LabView program as front end to control the test facility, the measurement technique,

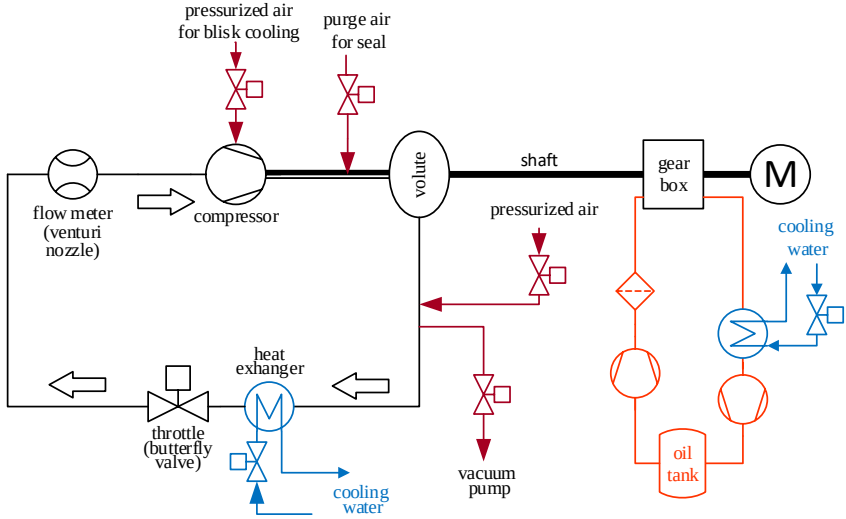


Figure 5.2: Simplified process flow diagram of ACTIVE

the data acquisition systems and the excitation system.

Since vibration issues are reported by Kristukat [132] for the former setup, which limited the operation to a few rotational speeds, the complete powertrain is redesigned for the present work. The redesign comprises a replacement of the gear box with a refurbished planetary gear box along with a new shaft including high-performance ball bearings and multi-plate clutches to connect the shaft and the gear box. In addition, a new test bed concept for low vibration is implemented. Therefore, a heavy test bed (≈ 4 t) was built from polymer concrete, which has very high damping properties to attenuate powertrain vibrations. The test bed is decoupled from the ground by means of air springs, which yields very low natural frequencies of the test bed due to the high mass and thus prohibits interference of test bed frequencies with the blisk modes investigated in the present work. As a result of the major redesign with focus on low vibration, the test facility can be operated at

almost all rotational speeds up to 18500 rpm.¹ A new vibration monitoring system is also installed for safe operation, see section 5.3.

ACTIVE is designed as a closed loop test facility and a simplified process flow diagram is shown in fig. 5.2. The loop is connected to vacuum pumps as well as to shop air via fast acting motorized valves, which allow for very accurate control of the pressure level. With the help of the strong vacuum pumps and low leakage of the test facility, near vacuum conditions at an absolute pressure level of 5 mbar can be achieved and maintained. The main source of leakage is the sealing between the blisk and the shaft bearings. Because of high shaft surface speeds and the requirement to avoid any oil cooling², it was decided to install a dry non-contact seal from flowserve (GASPAC). The seal is purged with pressurized air creating a small leakage flow³, which is easily compensated by the vacuum pumps. The main advantage of this configuration is the ability to use the test facility as a vacuum spin rig as well by removing the piping upstream and downstream of the compressor and mounting blank flanges.

The test loop is mainly built with DN300 pipes and the 90° bends are equipped with guide vanes to minimize secondary flows.

Two detailed views of the compressor are provided in fig. 5.3 and the main test section is highlighted in red. Upstream of the compressor, the pipe diameter is reduced from 300 mm to 240 mm (not shown in fig. 5.3). An elliptical bulb upstream of the test section further reduces the flow area to match the blade height of the blisk and enables a purely axial test section. The main concept of the compressor is to create an axial test section, where an axial compressor blisk can be investigated in an isolated manner as well as with upstream and downstream blade rows, which can be changed easily thanks to the modular design of the compressor. However, an isolated rotor configuration is investigated in the present work as this is the typical configuration in CFD for aerodynamic damping simulations and the work aims at a high degree of comparability between experiment and CFD. Therefore,

¹ the powertrain was balanced at 16900 rpm and the test facility was commissioned for a maximum rotational speed of 18500 rpm at vacuum conditions. The maximum rotational speed in the present work is 16900 rpm.

² oil can accumulate in closed loop test rigs, which could compromise safety requirements, therefore it was decided to avoid any oil systems connected to the bearings or seals

³ approximately 20 l min⁻¹, depending on the rotational speed, measured in a seal performance test rig with 6 bar purge air prior to the installation of the seal

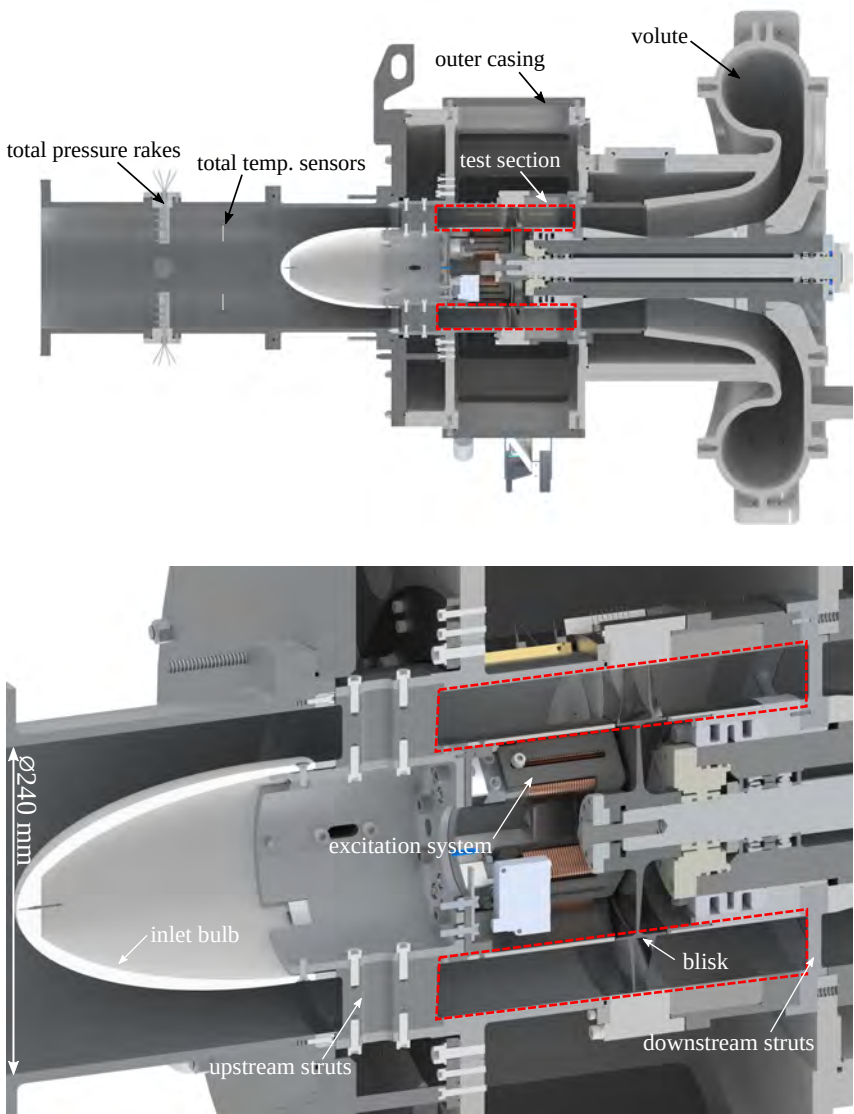


Figure 5.3: CAD render of cut through of the compressor, main test section highlighted in red

the upstream and downstream struts are designed to be far away from the blisk⁴ to minimize any influences on the flow field of the rotor. The excitation system is installed in the upstream hub, which is supported by four struts, located at 0°, 90°, 180° and 270° in circumferential direction, with NACA0017 profiles and a maximum thickness of 12 mm. The excitation of the blisk by the upstream struts can be neglected as no relevant crossings in the Campbell diagram exist for the investigated modes and rotational speeds.⁵ The struts feature channels, which are used as an access for cables to the excitation system and sensors in the hub. Six cylindrical struts with a diameter of 12 mm are located downstream of the blisk at 0°, 60°, 120°, 180°, 240° and 300°, to connect the shaft casing to the compressor casing and ensure high stiffness and low vibration of the powertrain. Downstream of the test section, a large volute collects the flow and guides it towards the heat exchanger. The heat exchanger is connected to an external cooling water loop and the temperature level is controlled using an automated water valve. The widths of the axial gaps between the blisk and the upstream and downstream hub are below 1 mm. A picture of the compressor can be found in fig. 5.4, which shows how the plates of the outer casing are used to loop through the cables of multiple sensors in a sealed manner.

The test facility is controlled by the PLC, where a safety chain is implemented. The operation of the test facility is realized in a Lab View program, which communicates with the PLC but cannot overrule the PLC safety chain. Air is chosen as the working fluid, but due to the possibility to evacuate the test facility along with its design for low leakage, the test facility could be operated with other gases.

5.2 Test Object

The test object is an axial compressor blisk with 17 blades named **ASB1** (Axial Subsonic Blisk) and it is designed within the present study. The design in-

⁴the axial distance between the upstream struts and the blisk is equivalent to 11 times the maximum thickness of the upstream struts and approximately 9 times the thickness for the distance between the blisk and the downstream struts

⁵the possibility of forcing due to the upstream struts was investigated in a preliminary study by Lehmann [140], which was supervised by the author of the present thesis

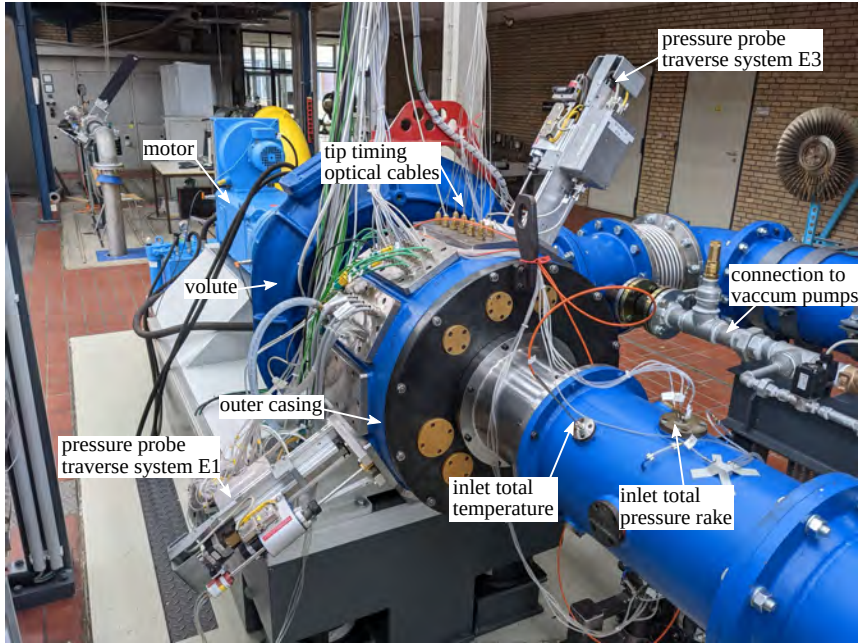


Figure 5.4: ITSM ACTIVE, picture of the compressor

tention was to develop a representative axial compressor blisk, which can be excited sufficiently in multiple normal modes. For the development, CFD as well as FE simulations are utilized and the complete process chain is realized in ANSYS Workbench. The blades feature circular arc camber lines with a NACA 4-digit thickness distribution [134] and a free vortex design in span-wise direction. The design rotational speed is 16200 rpm, where a total pressure ratio of close to 1.1 is reached at a corrected mass flow rate⁶ of 2.4 kg s^{-1} . A tip Mach number of 0.59 is achieved at the design rotational speed. Table 5.2 summarizes the main design parameters of the blisk. The maximum total-to-total isentropic efficiency of the blisk is close to 90 % according to CFD simulations and no flow separations are observed at the design point.

⁶corrected to inlet conditions of $p_{\text{ref}} = 970 \text{ mbar}$ (97 kPa) and $T_{\text{ref}} = 294.15 \text{ K}$, see eq. (5.1)

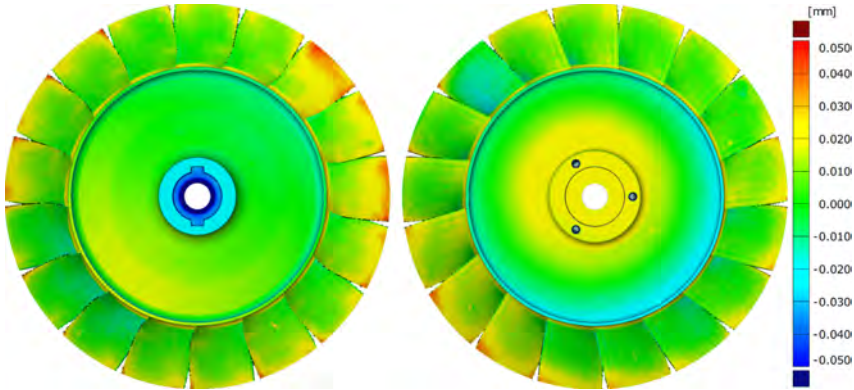


Figure 5.5: Differences between manufactured and CAD geometry. Measured by means of an optical 3D scan.

While the aerodynamic design of the blisk can be considered as straight forward, significant work was put into the structural design of the blisk and the selection of the material. In order to realize the electromagnetic excitation of blisk modes, the blisk has to be ferromagnetic, which limits the choice of possible materials. The blisk also exhibits significant centrifugal loads due to the high rotational speed and additional dynamic loads due to the intended mode vibrations. Furthermore, the reduction of the thickness of the hub contour, see fig. 5.10, was a design objective in order to enable sufficient vibration amplitudes while ensuring the structural integrity of the blisk. As a result, a trade-off in the selection of the material has been made between maximum magnetic permeability, which can be achieved with ferritic stainless steel and high yield strength, which requires duplex or martensitic stainless steel [202]. In the end, martensitic stainless steel 17-4 PH was chosen, as it combines excellent mechanical properties [192] with sufficient magnetic permeability. Different conditions of 17-4 PH are available and the condition H1025 was selected, since this condition maximizes the magnetic permeability [34] and provides a yield strength of 1000 Nmm^{-2} [19]. The blisk is designed to withstand centrifugal loads at an overspeed of 20000 rpm with a safety factor of 2.1 and the maximum permitted vibration amplitudes are

Table 5.2: ITSM-ASB1 axial compressor blisk, main parameters

blade count	17
hub diameter	160 mm
tip diameter	238.8 mm
chord length	44.5 mm
channel height	40 mm
material	17-4 PH condition H1025
mass	1.83 kg
rotational speed @ design point (DP)	16200 rpm
tip Mach number @ DP	0.59
total pressure ratio @ DP	1.1
corrected mass flow rate @ DP	2.4 kg s^{-1}
isentropic efficiency @ DP	0.896
tip gap @ DP	0.5 mm

determined by means of FE simulations and the Haigh diagram, see section 5.5.3.

Manufacturing of the blisk was realized by means of precise 5-axis milling at maximum tolerances of 0.05 mm in order to minimize geometrical mistuning. Differences of the manufactured geometry to the nominal CAD geometry are determined by means of an optical 3D scan, see fig. 5.5, which confirms high accuracy of the manufacturing process but reveals small geometrical differences of the blades nevertheless⁷. The high manufacturing accuracy is also reflected by the fact that the blisk reached balancing grade 2.5 according ISO 1940/1 at design speed without any additional removal of material. The structural integrity of the blisk was validated by the manufacturer with an overspeed test at 20000 rpm. A key connection is used to transmit torque from the shaft to the blisk and the blisk is axially screwed to the shaft to counter axial thrust.

The tip gap is 0.6 mm at rest and reduces to 0.5 mm in average at the design rotational speed of 16200 rpm, see fig. 5.6, which corresponds to 1.25 % of the channel height. As it can be seen by the difference in tip gap between blade

⁷equally distributed differences to the CAD geometry are tolerable in terms of mistuning but differences between blades definitely introduce geometrical mistuning

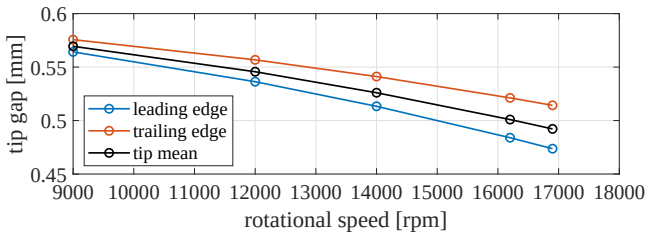


Figure 5.6: Reduction in tip gap due to centrifugal load. Calculated by means of FE simulations. Tip gap at rest = 0.6 mm.

leading and trailing edge in fig. 5.6, there is a minor untwist of the blades in operation, even though the blade profiles are stacked in their centroid. Nevertheless, the untwist is considered to be insignificant in terms of blade angles between different rotational speeds. The aerodynamic performance of the blisk is presented in chapter 9 by means of compressor map measurements and simulations. The structural dynamics of the blisk are discussed in chapter 7.

5.3 Measurement Technique

Multiple sensors and data acquisition (DAQ) systems are installed to operate the test facility and to acquire measurement data. The main parts of the instrumentation are presented in the following section.

5.3.1 Vibration Monitoring and Rotational Speed

In order to ensure safe operation of the test facility, four 1-axis piezoelectric vibration sensors VSP001 from ifm are installed on the motor, on the gear box, near the fixed and near the floating bearings with an uncertainty of $\pm 5\%$ for a measurement range of -50 to 50 g. The sensor data is continuously analyzed in a diagnostic box VSE100 from ifm with a pre-alarm set at a vibration velocity of 4.5 ms^{-1} and a main alarm at 7.1 ms^{-1} in agreement with DIN ISO

10816-3. The motor of the test facility is automatically shut down if the main alarm is triggered or the pre-alarm is activated for a defined number of seconds. The system was validated with portable sensors of an external partner during the balancing of test facility.

The rotational speed is measured by means of an optical sensor AV-ROS-W from avibia as well as with a magnetoresistance sensor GEL 2432 from Lenord & Bauer for redundancy. The later is placed close to a toothed ferromagnetic wheel mounted on the powertrain providing a 500/rev signal, which is post-processed with a fast DAQ module in the PLC. Differences between the two sensors are < 1 rpm. Since all measurements are conducted at the same inlet temperature, no correction of the measured rotational speed to reference inlet conditions is necessary.

5.3.2 Flow Meter and Inlet Conditions

The mass flow rate is measured by means of a venturi nozzle according to DIN EN ISO 5167-1 and 5167-3. The static pressure upstream of the nozzle is measured using a Siemens SITRANS P320 pressure sensor with an uncertainty of $\leq 0.065\%$ full scale ($=\pm 3$ mbar). The differential pressure is measured using a Siemens SITRANS P500 F pressure sensor with an uncertainty of $\leq 0.03\%$ full scale ($=\pm 0.075$ mbar). The exact geometry of the nozzle was determined by means of an optical 3D scan. The fluid properties of humid air are considered according to VDI Wärmeatlas [213], which requires the measurement of humidity. This is done upstream of the venturi nozzle with a HUMTMP sensor from TetraTec, which combines a relative humidity sensor and a Pt100 temperature sensor. The Pt100 sensor has an uncertainty of ± 0.13 K and the relative humidity is measured with an uncertainty of $\pm 1\%$. The calculation of the mass flow rate is conducted continuously in the PLC. The measurement data is acquired by a Siemens analog data acquisition PLC module, which has an uncertainty of 0.1% full scale for the pressure signal ($=\pm 0.25$ mbar), and 0.1% full scale for the relative humidity signal. The temperature is acquired with an uncertainty of ± 0.7 K. The total uncertainty of the mass flow rate due to the uncertainties of the instrumentation adds up to 0.11% according to DIN EN ISO 5167-1.

The mass flow rate measurement was calibrated at the ITSM by means of a

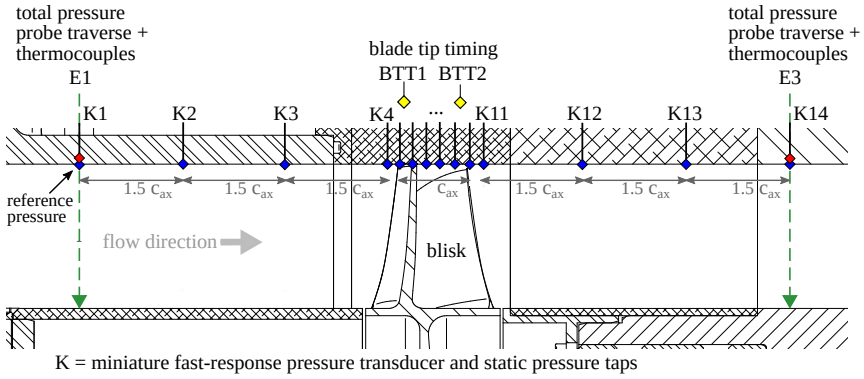


Figure 5.7: meridional cut of test section showing the sensor positions

calibrated flow meter from TetraTec for low mass flow rates as well as with total pressure probe traverse measurements in combination with CFD simulations. Based on the calibration, a calibration factor for the mass flow rate of 0.98 ($\dot{m}_{\text{calib}} = 0.98 \cdot \dot{m}$) was introduced. In order to compare measurements at different pressure levels, the mass flow rate is corrected to $p_{\text{ref}} = 970 \text{ mbar}$ and $T_{\text{ref}} = 294.15 \text{ K}$:

$$\dot{m}_{\text{corr}} = \dot{m}_{\text{calib}} \cdot \frac{p_{\text{ref}}}{p} \cdot \sqrt{\frac{T}{T_{\text{ref}}}} \quad (5.1)$$

The inlet conditions upstream of the test section in terms of total pressure $p_{t,\text{in}}$ and total temperature $T_{t,\text{in}}$ are measured by means of two opposing pressure rakes and two Pt1000 sensors, respectively. The rake and sensor positions are marked in fig. 5.3 and fig. 5.4. The uncertainty of the Pt1000 sensor is $\pm 0.13 \text{ K}$, the uncertainty of the Siemens analog DAQ PLC module is $\pm 0.7 \text{ K}$. The pressure of the total pressure rake is acquired using the pressure scanner described in section 5.3.3.

5.3.3 Casing Pressure and Pneumatic 4-hole Probes

The static pressure at the casing of the test section is measured at 14 axial positions, see fig. 5.7. At each position, four holes with a diameter of 0.5 mm are drilled into the casing, equally distributed in circumferential direction. The four pressure taps are coupled with a flexible tube forming a ring to pneumatically average the pressure measurement at each axial position. This ring is connected to a 16 channel pressure scanner NetScanner 9116 from Pressure Systems with the same type of flexible tube. Two pressure scanner with a measurement range of 5 psi (≈ 345 mbar) and an uncertainty of 0.05% full scale (± 0.17 mbar) are used in total for the test facility. The pressure in the scanner is measured relative to an individual reference pressure, which is the casing pressure at position K1⁸ for all channels. The reference pressure is measured using a Siemens SITRANS P320 pressure sensor and recorded with a NI cDAQ 9208 module resulting in a total uncertainty of 0.105% full scale (± 5.25 mbar). The static pressures at K1 and K14 are used to determine the static pressure ratio of the compressor in the present work:

$$\Pi_{ss} = \frac{p_{s,K14}}{p_{s,K1}} \quad (5.2)$$

In addition, the casing is equipped with miniature fast-response pressure transducers at each axial position, marked with K in fig. 5.7. Upstream (K1 to K3) and downstream (K11 to K14) of the blisk, differential pressure transducers XCE-080 from Kulite are used with a measurement range of 10 psi (≈ 689 mbar) and a diameter of 2.0 mm. The casing pressure at position K1 is used as the reference pressure for all unsteady pressure sensors. In the region of the blade tip, eight pressure transducers XCS-062 also from Kulite are placed with a measurement range of 15 psi (≈ 1034 mbar) and a diameter of 1.7 mm. The sensors are flush mounted in the casing using inserts, see fig. 5.8. The exact axial positions relative to the blade tip are also shown in fig. 5.8⁹. The power supply for the sensors and measurement amplification

⁸K1 shares the same axial position with the pneumatic 4-hole probe in the plane E1. K14 is located at E3.

⁹The circles in fig. 5.8 are scaled to the correct sensor diameter with respect to the blade tip. Kulite XCS-062 sensors are, to the author's knowledge, one of the smallest sensors available in the market. Still, the sensors cover a significant area with respect to the blade tip, which inevitably leads to the measurement of locally averaged values.

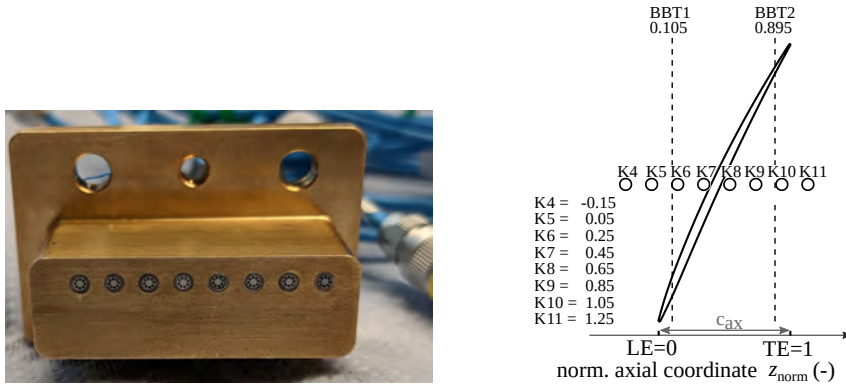


Figure 5.8: Flush mounted unsteady pressure sensors and their position relative to the blade tip

is realized by means of MOM-DMSi modules from GEPA with an accuracy of the power supply of 0.05% and an amplification error of $\leq 0.08\%$. For the DAQ, a 16 bit NI PXIe-6349 card with 32 channels, a maximum simultaneous sampling frequency of 500 kHz and an accuracy of 0.003% full scale is used in a NI PXIe-8800 chassis. The raw signal of the 1/rev and 500/rev sensors are also logged with the same system to correlate the unsteady pressure data with the blade positions in the post-processing, see section 5.5.2. The fast-response pressure transducers were individually calibrated in-situ prior to the experiments including the power supply and DAQ system as described above. The sensors are mounted in a straight line at 335° in circumferential direction to avoid any interference with wakes of the upstream struts or the pneumatic probes. In addition, measurements with the fast-response pressure transducers and the pneumatic 4-hole probes are always performed consecutively.

In order to measure the flow field in radial direction upstream and downstream of the blisk, pneumatic 4-hole probes are installed at E1 and E3, respectively, see fig. 5.7. The probe head, see fig. 5.9, is shaped like a wedge, similar to probes used in [215], which is beneficial for transonic flow con-

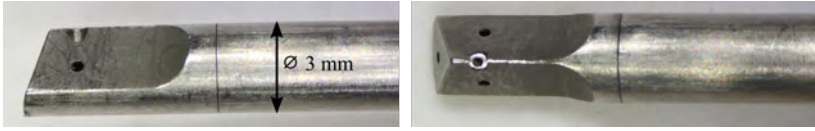


Figure 5.9: pneumatic 4-hole probe head

ditions but can be used in subsonic conditions nonetheless. The shaft of the probe has a diameter of 3 mm and a PTFE sealing in the compressor casing is used to prevent leakage at the probe pass-through. The probes are calibrated at an ITSM in-house wind tunnel for Mach numbers between 0.05 and 0.5 at ambient conditions and yaw and pitch angles between -10° and 10° . The pressure data is recorded using the NetScanner 9116 described above. The pneumatic probe measurements allow for the calculation of the local total and static pressure, the absolute flow angles and the velocity magnitude. Details on the calibration of the probes as well as the post-processing can be found in [163]. The probe measurement is fully automated¹⁰ by means of a Lab View program adopted from the ITSM axial diffuser test facility [9]. The probe traverse system is adopted from the ITSM industrial steam turbine test rig, which is described in [101]. Both traverse systems can be seen in fig. 5.4. The traverse for E1 is located at 240° in circumferential direction in order to avoid interference with the wake of the upstream struts. E3 is located at 30° to avoid influences of the downstream struts. Measurement data below a relative blade height (span) of 0.1 and above 0.9 is not used due to possible influences of the measurement data by the interaction of the probe head with the hub or casing wall.

In addition, four thermocouples of type K with a diameter of 1 mm are positioned in the casing at the E1 as well as E3.

¹⁰The flow field is measured at 17 radial positions, to which the probe is positioned automatically by the traverse system. At each position, a control loop is used prior to the measurement to align the probe with the local flow vector by minimizing the difference in yaw angle between the probe and the flow field. The standard deviation of the measured pressure is evaluated to ensure reliable measurement data before the probe is moved to the next radial position. The measurement time of one radial traverse is approximately 10 min.

5.3.4 Blade Vibration

The blade deflection of the vibrating blisk is measured by means of a blade tip timing system BVM 20036 from Hood Technology. A 12 channel system setup is used, which is based on a NI PXI system with $4 \times 3 = 12$ lasers¹¹, 12 photodetector modules as well as 12 analog to digital converters. The system can be extended to 36 channels. Eleven optical sensors with a tip diameter of 1 mm are embedded in a 3 mm stainless steel casing and recess mounted in the compressor casing¹². The sensors are custom-made for ACTIVE and enable fast and precise assembly and disassembly in combination with Serto fittings. The sensors are mounted in two axial planes, see fig. 5.8, where six sensors are located in plane BBT1 and five¹³ sensors in plane BTT2, respectively. The location of both planes was defined in order to capture the first bending as well as first torsional mode family. The circumferential positions of the sensors are calculated with a proprietary software tool from Hood Technology with the objective to maximize the detection of blisk modes investigated in the present work. The exact positions are provided in appendix B.2. The optical sensors consist of six optical fibers, where the central fiber transmits the laser light, emitted by the laser module, towards the blisk in radial direction. Five fibers surrounding the central fiber receive the laser light reflected by the blade tip and transfer it to the receiver module allowing for the detection of the blade arrival time of every blade at every sensor. With the help of the geometrical information of the blisk along with the 1/rev signal, a theoretical arrival time can be computed and compared to the measured arrival time, which is modified due to the blade vibration. Based on this information, the nodal diameter can be identified for non-synchronous vibrations and a circumferential Fourier-fit can be performed to calculate the blade individual amplitudes. The system is also able to calculate blade individual amplitudes for synchronous vibrations and various excitation orders. These calculations are conducted in the analysis software from Hood Technology, further information is provided in [106]. It

¹¹40 mW, 658nm class 3b (hazardous for eye exposure!)

¹²the working distance between the blade tip and the optical tip timing sensors is 2.6 mm in the present setup

¹³only five sensors are used in plane BTT2 because one channel of the 12 channel system is used for the 1/rev sensor as a trigger

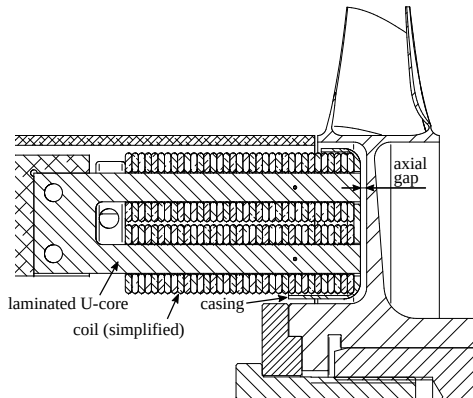


Figure 5.10: CAD drawing of the electromagnetic blisk excitation system

has to be noted that tip timing system does not directly measure true blade displacement amplitudes and a conversion is required to calculate the blade displacement in axial and circumferential direction, see appendix B.1. Since the present work mainly focuses on the first bending mode family, which shows almost completely axial displacements for all modes, see table B.1, only the axial displacement is usually depicted in the following. According to the manufacturer, displacements $\geq 3\mu\text{m}$ can be measured, which is confirmed in chapter 8.¹⁴

5.4 Excitation System and Blisk Monitoring

The excitation of the blisk at traveling wave modes is realized by AC electromagnets, which are developed as part of the present work. The objective of the development is to maximize the magnetic force within the given installation space without any disturbance of the flow field. Therefore, it is decided to use four identical electromagnets, which are placed upstream of the blisk,

¹⁴for the measurements in the present work, an average noise level of the BTT signal of approximately $1\mu\text{m}$ is observed and displacement peaks $\geq 2\mu\text{m}$ can be measured reliably

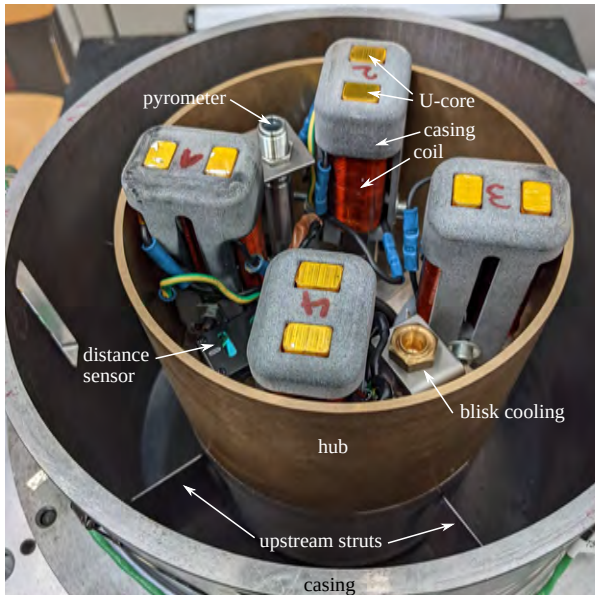


Figure 5.11: Electromagnetic blisk excitation system mounted in the upstream hub

see fig. 5.10 and fig. 5.11. The magnets consist of a U-core, a coil on each leg and a casing to fix the coils to the magnet core.

In general, the magnetic force for electromagnets can be estimated with a simplified equation for U-cores:

$$F_{\text{mag}} = \frac{\mu_0 (N_w I)^2 A}{2 l_{\text{gap}}^2} \quad (5.3)$$

with μ_0 as the vacuum magnetic permeability¹⁵, N_w as the total number of windings on the magnet core, A as the magnet core area facing the blisk and l_{gap} as the width of the air gap between the magnet and the blisk. I represents the electric current in the coils. The derivation of eq. (5.3) can be found in [13] or [113], which shows that since the magnetic permeability of the core

¹⁵ $\mu_0 = 1.25663706214 \times 10^{-6} \text{ NA}^{-2}$

and the blisk are orders of magnitude higher compared to air or vacuum¹⁶, the width of the air gap mainly determines the magnetic force along with the number of windings, the electric current and the size of the magnet core. For the current application and based on the available installation space, each coil has 120 windings in three layers and they are built with 1.6 mm copper wire. The maximum electric current is defined as 10 A in order to limit heat development due to ohmic resistance. The axial gap between the blisk and the magnets is set to 1.6 mm to ensure a safety margin since contact of the magnet and the rotating blisk has to be prevented by all means. A distance sensor IL-S065 from Keyence is installed in the hub, see fig. 5.11, to monitor the axial gap during operation and to stop the compressor in case of significant changes of distance between blisk and magnets.

Additional losses such as eddy current¹⁷ and hysteresis losses¹⁸ have to be considered for AC electromagnets, which increase heat development in the magnets as well as in the blisk and reduce the resulting magnetic force. While the hysteresis losses scale linearly with the frequency, eddy current losses scale with the square of the frequency [8, 66]. In order to reduce eddy current losses in the electromagnets, a laminated U-core was selected. The U-cores are individually manufactured for this application and were built using sheets of VACODUR 49¹⁹ with a thickness of 0.1 mm, which shows a significantly higher magnetic saturation and lower hysteresis losses compared to standard iron-silicon alloy sheets [203, 204]. Eddy currents in the blisk are

¹⁶ $\mu_{\text{vacuum}} = \mu_0 \approx \mu_{\text{air}}$

¹⁷Voltage is induced in the magnet core and blisk due to the alternating magnetic field, which results in electric currents. Due to the shape of the magnetic field, eddy structures appear in the electric field, which create a magnetic field counteracting the initial magnetic field and thus lowering the resulting magnetic force of the electromagnet acting on the blisk [113]

¹⁸Heat is generated in ferromagnetic materials due to the alternating change in magnetic polarity and thus the alternating realignment of the molecules [25, 113].

¹⁹trademark name from VACUUMSCHMELZE GmbH & Co. KG for an iron alloy according to the specification ASTM A801-09 Alloy Type 1 (alloy with 49 % cobalt)

inevitable, but the low electric conductivity of 17-4 PH is beneficial²⁰.

It has to be noted for AC electromagnets that the frequency of the magnetic force is twice the AC frequency, which can be explained by expanding eq. (5.3) with $I = I(t) = \hat{I} \sin(\omega t)$:

$$F_{\text{mag}}(t) = \frac{\mu_0 (N_w \hat{I} \sin(\omega t))^2 A}{2 l_{\text{gap}}^2} = \frac{\mu_0 N_w^2 A \hat{I}^2}{2 l_{\text{gap}}^2} (1 - \cos(2\omega t)) \quad (5.4)$$

Furthermore, the right-hand term of eq. (5.4) in the brackets only varies between 0 and 1 and shows that the magnetic force for AC electromagnets can be separated into a constant (1) and an alternating part ($-\cos(2\omega t)$). This means that the direction of the magnetic force is not changed. In other words, the electromagnets always create an attracting force on the blisk regardless of the direction of the electric and magnetic field [25, 113].

The power supply for the electromagnets is provided by means of four audio amplifiers Pronomic XA-1400 with a maximum output voltage of 100 V. A high output voltage is important to realize sufficient electric currents in the coils at high frequencies since the electrical impedance²¹ and therefore the voltage requirement increases with frequency: $U = I \sqrt{R^2 + (\omega L)^2}$.

The analog input signal for the amplifiers is created by NI cDAQ module NI-9269 and a LabView program. An individual signal in terms of phase, frequency and amplitude can be created for each magnet, further information see section 5.5.3. The individual magnet temperature is monitored by means of thermocouples between the coil and the U-Core (one per magnet) to avoid damages of the insulation and short circuits due to excessive heat. The max-

²⁰A preliminary study was performed using the 2D solver FEMM (www.femm.info) to optimize the magnet design as well as to investigate the influence of different magnet core and blisk materials on the resulting magnetic force. Even though the simulations represent a simplification of the application and simplified loss models are used, general trends could be derived. The simulation results led to the conclusion that if iron would be used as blisk material, which approximately has double the magnetic saturation compared to 17-4 PH (2.5 T vs. 1.2 T), surprisingly, the resulting magnetic force is approximately 75 % lower (strongly depending on the AC frequency). This can be attributed to the higher electric conductivity of iron ($\sigma_{\text{iron}} = 10.44 \text{ MS m}^{-1}$, $\sigma_{17-4 \text{ PH}} = 1.25 \text{ MS m}^{-1}$) and hence higher eddy current losses.

²¹Capacitors could be used to compensate the inductive reactance of the coils and therefore lower the required voltage. This possibility was successfully investigated in a preliminary study and is included in the experimental setup but not used since the amplifiers provided sufficient voltage output for the present study.

imum magnet temperature is set to 60 °C. The two coils on each magnet are connected in parallel with one amplifier and the electric current is monitored for each coil using ACS712 20 A sensors in combination with Arduino modules. The electric circuit of all electromagnets can be opened immediately with the help of relays to perform free decay measurements or for safety reasons.

Because of eddy current and hysteresis losses, the blisk temperature can increase, which is monitored by means of a pyrometer CS-SF15-C1 from Micro-Epsilon installed in the upstream hub next to the excitation system, see fig. 5.11. If the maximum blisk temperature of 40 °C is reached during an excitation cycle, the excitation automatically stops. The blisk can be cooled actively using pressurized air, see fig. 5.11.

5.5 Operation and Data Post Processing

In the following section, information is provided on how the test facility is operated in order to measure compressor maps and how to post process unsteady pressure data. In addition, it is described how the excitation system is operated in terms of frequency and sweep velocity to excite normal modes of the blisk. Finally, the process is shown of how to use time-dependent BTT data to calculate damping.

5.5.1 Compressor Map, Steady Measurements and Vacuum

The closed loop setup allows for the measurement with different inlet conditions. In the present work, compressor maps are measured at a total inlet pressure of 250, 500 and 970 mbar (25, 50 and 97 kPa). The total inlet temperature is set to 21 °C. At each inlet pressure, five speed lines are measured at 9000, 12000, 14000, 16200 and 16900 rpm. The mass flow rate is manually adjusted with a motorized butterfly valve, which is fixed in its position once the target mass flow rate is reached. At each rotational speed, one operating point (OP) with a fully opened butterfly valve is measured. These operating points do not correspond to choked conditions, since the closed loop setup and in particular the heat exchanger introduce pressure losses, which

Table 5.3: Fluctuations of the main parameters for all investigated operating points. $\pm 3 \cdot \text{RSD}$ represents the maximum fluctuation for 99.73 % of the measurement data.

$p_{t,\text{in}}$	250 mbar		500 mbar		970 mbar	
number of OPs	63		51		134	
	mean	(max.)	mean	(max.)	mean	(max.)
$\pm 3 \cdot \text{RSD}(\dot{m}_{\text{corr}})$	$\pm 0.15 \%$	$(\pm 0.23 \%)$	$\pm 0.14 \%$	$(\pm 0.20 \%)$	$\pm 0.15 \%$	$(\pm 0.24 \%)$
$\pm 3 \cdot \text{RSD}(\pi_{\text{ss}})$	$\pm 0.17 \%$	$(\pm 0.22 \%)$	$\pm 0.16 \%$	$(\pm 0.25 \%)$	$\pm 0.24 \%$	$(\pm 0.38 \%)$
$\pm 3 \cdot \text{RSD}(\text{rot. speed})$	$\pm 0.01 \%$	$(\pm 0.02 \%)$	$\pm 0.01 \%$	$(\pm 0.08 \%)$	$\pm 0.02 \%$	$(\pm 0.07 \%)$
$\pm 3 \cdot \text{RSD}(p_{t,\text{in}})$	$\pm 0.15 \%$	$(\pm 0.18 \%)$	$\pm 0.10 \%$	$(\pm 0.15 \%)$	$\pm 0.15 \%$	$(\pm 0.24 \%)$
$\pm 3 \cdot \text{RSD}(T_{t,\text{in}})$	$\pm 0.21 \%$	$(\pm 0.86 \%)$	$\pm 0.20 \%$	$(\pm 0.76 \%)$	$\pm 0.15 \%$	$(\pm 1.32 \%)$

limit the operation of the compressor at very low pressure ratios. Only stable operating points are investigated and a safety margin of 5 % in mass flow rate from the last stable point of the CFD simulations is used. The total inlet temperature is continuously controlled by the PLC with a motorized valve in the cooling water circuit of the heat exchanger. The total inlet pressure is also continuously controlled by the PLC. At each operating point, the standard deviations of the main parameters (mass flow rate, pressure ratio, total inlet temperature and pressure) are observed and measurement data is acquired once the asymptotic trends of the standard deviations converged to their final values. For each operating point, 30 s of data is saved at a sampling frequency of 2 Hz. The mean values of these samples are used in the present work to depict compressor maps, casing pressure distributions and other steady measurement data. The test facility shows a very stable behavior at all investigated operating points. Table 5.3 provides an overview of the operating point stability for all measured operating points. For all OPs, the relative standard deviation (RSD)²² of the main parameters is calculated. Assuming a normal distribution, 99.73 % of the measurement data show a deviation to the mean value in range of $\pm 3 \cdot \text{RSD}$. In table 5.3, the mean and the maximum value of this range is provided over all OPs and for different inlet conditions, showing that in average, the fluctuations of all OPs investigated in the present work are lower than $\pm 0.24 \%$.

²²RSD is equal to the standard deviation divided by the mean value

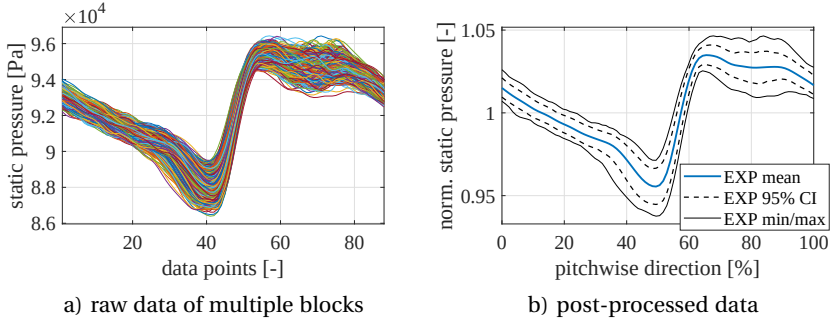


Figure 5.12: Example of phase-locked averaging of unsteady pressure data at $n = 16200 \text{ rpm}$

The pressure of the test facility can be reduced to a total inlet pressure level of $p_{t,\text{in}} = 5 \text{ mbar}$, which is referred to in the present work as (near) vacuum operating condition. The vacuum condition is used for the tip timing validation measurements as well as measurements of the combined structural and material damping of the blisk. The static pressure varies between 4.7 and 4.9 mbar for $p_{t,\text{in}} = 5 \text{ mbar}$, depending on the rotational speed, but the influence of the residual air on damping measurements is assumed to be negligible.

5.5.2 Unsteady Pressure Data

Because of the high temporal resolution of the unsteady pressure data at the casing, the (average) pressure field in pitchwise direction can be reconstructed using the method of *phase-locked averaging*²³. The method is widely used in literature, see for example [7, 21, 155], and will be briefly explained for the present case below. The method allows for the removal of stochastic fluctuations, as introduced for example by turbulence, by averaging the local pressure information for a sufficiently long period of time and hence enables

²³also known as *ensemble averaging*

the direct comparison to steady-state RANS simulations.

The pressure signal of the fast-response pressure transducers, which is acquired with a sampling frequency of 400 kHz, is separated into equally spaced blocks, each corresponding to one blade pitch²⁴. Therefore, the 1/rev signal, which is available with the same temporal resolution, is used as a trigger. An example of the raw data separated into blocks can be seen in fig. 5.12a. In the present work, 1 s of data is used, which is equal to 1×10^6 data points per pressure transducer. The number of points per pitch varies depending on blade passing frequency (BPF) and hence on the rotational speed.²⁵ Since small variations in the rotational speed are possible within the measurement time of 1 s, all data blocks are interpolated to the same number of data points and thus to the same pitchwise position. Subsequently, the average is calculated at every pitchwise position resulting in a mean pressure distribution as depicted in fig. 5.12b. The available data also allows for the calculation of the local standard deviation (SD), which is used to plot a confidential interval of 95 % ($= \pm 2 \cdot \text{SD}$) along with the minimum and maximum values to provide information of local pressure fluctuations. The pressure in fig. 5.12b is normalized by the reference pressure at K1, c.f. fig. 5.7. The processed data is also used to create contour plots of the pressure distribution on the casing above the rotor, see for example fig. 9.4, where the axial resolution corresponds to the number of sensors and the circumferential resolution to the number of data points.

5.5.3 Excitation

The following section discusses the choice in excitation frequency and sweep rate as well as the displacement limits of the blisk.

²⁴it should be noted that the method only works for rotors with equally spaced blades as the assumption is made that the average flow field in all blade passages is the same

²⁵157 data points for $n = 9000 \text{ rpm}$ ($400 \text{ kHz} / (17 \cdot 9000 \text{ rpm} / 60) \approx 157$), 88 data points for $n = 16200 \text{ rpm}$

Excitation Frequency and Phase

For the calculation of the magnet frequency, which is required to excite certain blisk modes, the work of Wildheim [222, 223] is utilized. Wildheim provides an equation to describe all possible resonance frequencies ω ²⁶, which are perceived by a rotating blisk induced by a stationary excitation source:

$$\omega = |\pm (jN \pm n_{\text{nd}})\Omega \pm \omega_{\text{magnet}}| \quad j = 0, 1, 2, \dots \quad (5.5)$$

Equation (5.5) depends on the angular velocity of the blisk Ω , the number of nodal diameter of the considered blisk mode n_{nd} , the number of blades N and the stationary excitation frequency ω_{magnet} ²⁷. It should be mentioned that eq. (5.5) reduces to eq. (2.34) for $\omega_{\text{magnet}} = 0$, which yields the excitation lines in the ZZENF diagram. In Equation (5.5), the excitation in both traveling wave directions and the direct as well as aliased excitation conditions are included, but it is possible to distinguish between the main backward traveling wave excitation condition:

$$\omega_{\text{magnet,BTW1}} = |\omega - n_{\text{nd}}\Omega| \quad (5.6)$$

And the main forward traveling wave excitation condition:

$$\omega_{\text{magnet,FTW}} = |\omega + n_{\text{nd}}\Omega| \quad (5.7)$$

In Equations (5.6) and (5.7), ω_{magnet} is rearranged in order to directly calculate the required magnet frequency to excite a certain mode either in a forward or a backward traveling direction. An example for the excitation of the same double mode (B1ND2) in both directions is shown in figs. 5.13a and 5.13b. In both figures, the first bending mode family (B1), which is the mode family the present work is focused on, is highlighted with blue dots. The corresponding mode shapes are depicted in figs. B.2 and B.3. The modes are named according to the corresponding mode family and the number of

²⁶the frequency ω and the angular velocity Ω are expressed in [rads^{-1}] in consistency with Wildheim [222, 223], but the frequencies are depicted and referred to in [Hz] in the present work with $f = \omega/2\pi$

²⁷In this convention ω is always the excitation frequency perceived by the blisk in the rotating frame of reference and ω_{magnet} is the frequency of the magnetic force in the stationary frame of reference, which is twice the AC frequency to control the magnet, see section 5.4.

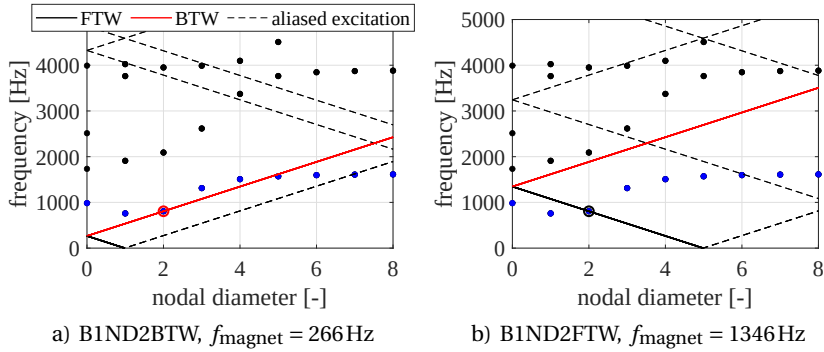


Figure 5.13: Example of the change in magnet frequency for a backward and forward traveling wave excitation of the first bending mode with two nodal diameter at $n = 16200 \text{ rpm}$ (First mode family marked in blue)

nodal diameter. The general notation of modes in the present work is as follows²⁸:

- *B1ND2*: double mode (complex), consisting of two real modes, mode family B1, two nodal diameter (ND2)
- *B1ND2-1*: standing mode (real), first mode of double mode B1ND2
- *B1ND2-2*: standing mode (real), second mode of double mode B1ND2
- *B1ND2BTW*: double mode B1ND2 excited in a backward traveling wave
- *B1ND2FTW*: double mode B1ND2 excited in a forward traveling wave

In addition to the blisk natural frequencies in fig. 5.13, eqs. (5.6) and (5.7) are depicted with red and black lines, respectively, as well as the aliased excitation lines according to eq. (5.5) with dashed lines. It can be seen that multiple excitation lines are present which could potentially lead to excitation of

²⁸Except for modes with zero nodal diameter, which always appear as real, single modes. They are just named according to the mode family and the nodal diameter, for example: B1ND0

multiple modes and the choice in excitation frequency and frequency sweep range is crucial for an isolated and successful experiment. Nevertheless, the depiction in the figures exaggerates potential resonances due to the thickness of the lines and dots. For example in fig. 5.13a it looks like there is a second resonance at B1ND7BTW, which is not detected in the experiment due to an offset in excitation frequency of a few Hertz.

In reality and due to the low overall damping of the blisk modes, the excitation frequency has to almost exactly coincide with the resonance frequency. Furthermore, the example shows that the exciting magnet frequency is in general lower than the corresponding natural frequency for BTW excitation and vice versa for FTW excitation. This becomes even more apparent, if the required magnet frequencies for all modes of the first mode family are considered, as it is done in fig. 5.15. In this figure, eqs. (5.6) and (5.7) are also evaluated for different rotational speeds²⁹. It can be seen that very high excitation frequencies are necessary for FTW excitation in particular for higher nodal diameter. This is challenging for electromagnetic excitation due to eddy current and hysteresis losses in the magnets and in particular in the blisk. Hysteresis losses scale linearly and eddy current losses with the square of the AC frequency and cause a significant decrease in magnetic force for high frequencies.

As shown in fig. 5.11, four electromagnets are positioned equally in the hub with a distance between each of 90° in circumferential direction. The magnets can be used in different configurations, where either just one magnet is used, or two opposing magnets or all four magnets are used at the same time. While the frequency is the same for all magnets, the magnet phase has to be adjusted individually to compensate for the different positions in the hub and the target nodal diameter³⁰:

$$\varphi_{\text{magnet}} = a \cdot n_{\text{nd}} \cdot \varphi_{\text{mag.pos.}}, \quad a = \begin{cases} -1 & \text{(BTW excitation)} \\ 1 & \text{(FTW excitation)} \end{cases} \quad (5.8)$$

²⁹the natural frequencies are determined by means of pre-stressed FE modal analyses at the corresponding rotational speeds

³⁰ φ_{magnet} represents the phase of the magnetic force of each magnet. This phase has to be divided by 2, similar to the magnet frequency, to get the phase of the magnet individual AC signal to control the magnet.

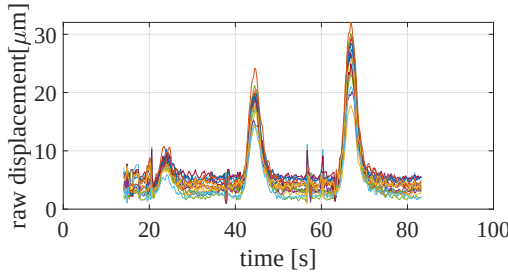


Figure 5.14: Excitation of B1ND3BTW at 6000 rpm and $p_{t,in} = 5$ mbar with one, two and four electromagnets and maintaining the electric current in the coils, resulting in a linear increase of the the blade displacement amplitude. Individual displacement of all blades is depicted.

The phase due to the circumferential positions of the magnets $\varphi_{\text{mag,pos.}}$ is defined between 0 and 2π and is increasing in the direction of the blisk rotation. This is realized in the LabView program and the analog output card mentioned in section 5.4. Multiple preliminary tests were conducted with different numbers of magnets activated and it was observed that by doubling the number of active magnets while maintaining the same electric current in the coils for all magnets, the displacement amplitude also doubles in good approximation. An example of such a test is shown in fig. 5.14. Based on these findings, different numbers of active magnets are used in the present work, if required.

Frequency Sweep Rate

In order to measure the stationary frequency response of the blisk in a resonance, which is necessary for the correct determination of the damping ratio, the excitation frequency sweep rate $\dot{f}$ has to be sufficiently low. The required sweep rate depends on the natural frequency f_0 as well as the damping ratio ζ and Hackenberg & Hartung [89, 90] define a reduced sweep velocity (rate):

$$\kappa = \frac{\dot{f}}{\zeta^2 f_0^2} \quad (5.9)$$

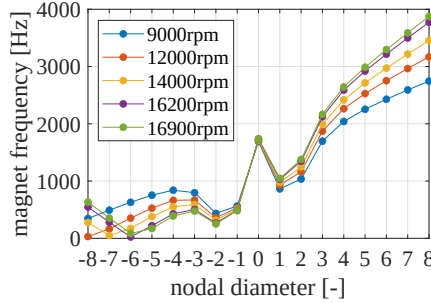


Figure 5.15: Magnet frequency for excitation of the first bending mode family at different rotational speeds. Sign of nodal diameter represents traveling wave direction: > 0 = FTW, < 0 = BTW

The authors state that $\kappa = 1$ is required to measure the stationary response with an error of 1 % in amplitude and 2.5 % in damping ratio³¹. For higher sweep rates, the damping ratio is usually overestimated and beats appear in the frequency response curve. Since f_0 and ζ are not known a priori, an estimated reduced sweep velocity is calculated after the measurements

$$\kappa_{\text{est}} = \frac{\dot{f}}{\zeta_{\text{est}}^2 f_{0,\text{est}}^2} \quad , \quad (5.10)$$

where ζ_{est} and $f_{0,\text{est}}$ are determined for every blade using an SDOF fit, see the section below and section 2.4.5. Hackenberg & Hartung [89, 90] distinguish between three cases:

- $\kappa_{\text{est}} \leq 1$: $\kappa \approx \kappa_{\text{est}} \rightarrow \zeta \approx \zeta_{\text{est}}, f_0 \approx f_{0,\text{est}}$
- $1 \leq \kappa_{\text{est}} \leq 4$: a correction method proposed in Hackenberg & Hartung [89] can be used to determine ζ and f_0 .

³¹Hackenberg & Hartung [89, 90] compare their criterion to similar criteria in open literature in the form of $\dot{f} \leq a \zeta^2 f_0^2$, where Hackenberg & Hartung recommend $a = 1$, Ayers et al. [5] $a = \pi$ and ISO 7626-2:2015 & Ewins [59] $a = 3.6$. Hackenberg & Hartung estimate that the error in amplitude by using $a = \pi$ or $a = 3.6$ increases to 4.9 % and 5.6 %, respectively. It can be assumed that the errors in damping ratio are significantly larger.

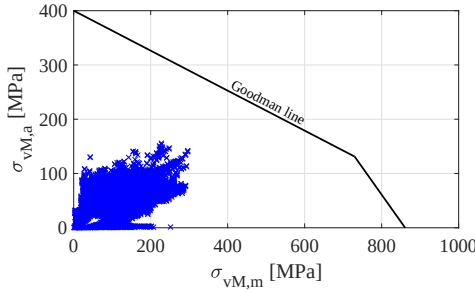


Figure 5.16: Haigh diagram, B1ND3BTW, $d_z = 0.34$ mm in BTT1, $n = 16200$ rpm.

- $\kappa_{\text{est}} > 4$: no straightforward correction of ζ_{est} , $f_{0,\text{est}}$ available

For the present work, all aerodynamic damping measurements are performed with $\kappa_{\text{est}} \leq 1$, which allows for the direct determination of the damping ratio based on the frequency response curves and with maximum errors to the stationary response as mentioned above. For damping measurements at near vacuum, $\kappa_{\text{est}} \leq 1$ was not achieved for all measurements due to the very low damping ratios. Possible errors for the measurement of the structural and material damping are discussed in chapter 10. Moreover, the correction according Hackenberg & Hartung [89] is investigated in section 10.1 for different sweep rates but showed an underestimation of damping in general and is thus not further used for the present work.

Excitation Limits

Since the present work aims at controlled excitation of resonances, it is very important to determine the displacement limits of each mode prior to the experiments to ensure safe operation. This is done by means of Haigh diagrams [147], which are constructed based on material properties from [19, 192]. An example of a Haigh diagram for the mode B1ND3BTW is depicted in fig. 5.16. The mean stresses $\sigma_{vM,m}$ at design rotational speed are determined by means of a linear pre-stress FE analysis. Information on the FE setup is given in section 6.1. The alternating stress amplitudes $\sigma_{vM,a}$ are

Table 5.4: Blade displacement limits for the first bending mode family B1 specified in the plane BBT1 as axial displacement d_z and displacement directly measured by BTT d_{BTT} . A safety factor of 2 is included.

BIND...	0	1	2	3	4	5	6	7	8
max. d_z [mm]	0.30	0.44	0.37	0.34	0.28	0.26	0.26	0.26	0.26
max. d_{BTT} [mm]	0.75	1.10	0.90	0.85	0.73	0.70	0.68	0.68	1.68

calculated based on a pre-stressed linear FE modal analysis. All stresses are calculated as von Mises stresses [6]. The linear modal analysis provides stresses for each normal mode, which scale linearly with the eigenvector length. Hence, by scaling the eigenvector to a defined value the same scaling factor can be applied to the stresses. Complex eigenvectors are defined according to eq. (2.12) and the von Mises stresses are superimposed in the same way³²:

$$\tilde{\sigma}_{\text{vM},m,12} = \sigma_{\text{vM},m,1} + i\sigma_{\text{vM},m,2} \quad (5.11)$$

Subsequently, the magnitude of the stress vector is scaled to meet the criteria in the Haigh diagram and the same scaling factor is applied to the eigenvector yielding the maximum displacement. In the present case, the stresses are scaled such that the maximum stress coincides with the Goodman line and then the resulting maximum displacement amplitude is divided by a safety factor of 2 to compensate for possible uncertainties. In table 5.4, the displacement limits including the safety factor are listed as axial displacement in the tip timing plane BTT1 as well as the displacement directly measured by the BTT system³³. The limits only apply for measurements in BTT1 and care should be taken as significantly lower axial displacement limits exist in other axial positions due to the geometry and mode shapes of the blisk.

5.5.4 Damping

The experimental determination of aerodynamic damping is realized as described in section 2.4.5 by measuring the overall damping and subsequently

³²except for BIND0, which is a real mode

³³ d_z can be converted to d_{BTT} , see appendix B.1. d_{BTT} limits are very useful since this value is directly available in the BTT monitoring system during the experiment.

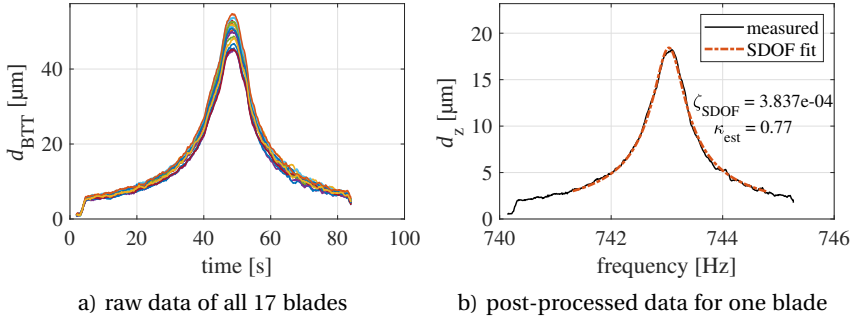


Figure 5.17: Example for the determination of the blade individual damping using a SDOF fit on the frequency response curve measured at BTT1. B1ND2BTW, $p_{t,in} = 500 \text{ mbar}$, $\dot{m}_{red} = 1.4 \text{ kg s}^{-1}$, $\dot{f} = 0.0625 \text{ Hz s}^{-1}$

subtracting the material and structural damping. The combined material and structural damping is thereby determined by means of measurements at vacuum. Comparisons to the extrapolation method are provided in section 11.1. Since a blisk is investigated in the present work, damping measured at vacuum can be completely attributed to material damping. The overall damping is mainly measured by means of frequency sweeps as described in the section above. The first post-processing part of the resonance sweeps is performed in the BTT analysis software, where a waterfall diagram is used to locate and isolate the resonance and to determine the nodal diameter and traveling wave direction of the excited mode as well as its frequency in the rotating frame of reference.³⁴ Subsequently, a non-integral circumferential Fourier fit [106] is applied to calculate the blade individual displacement for the resonance mode of interest. Figure 5.17a shows the resulting data exported from the BTT software as displacement amplitude versus time for all

³⁴the BTT sensors are fixed in the stationary frame of reference, hence the observed frequency has to be transformed to the rotating frame of reference, based on the information of rotational speed and nodal diameter, to obtain the true mode vibration frequency. In addition, aliasing effects are compensated, which arise due to the undersampling of the blade vibration signal with a limited number of BTT sensors.

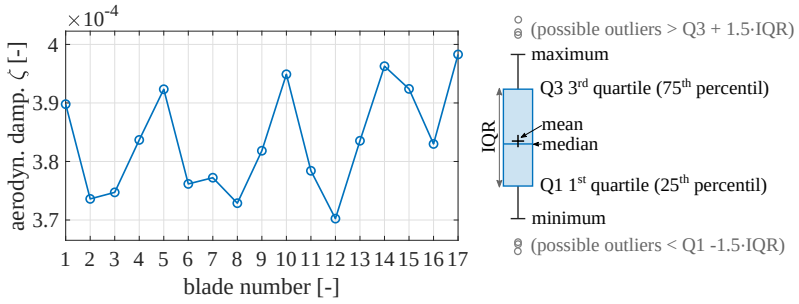


Figure 5.18: Depiction of damping distribution by means of a box-plot. Example for B1ND2BTW, $p_{t,\text{in}} = 500 \text{ mbar}$, $\dot{m}_{\text{red}} = 1.4 \text{ kg s}^{-1}$, $\dot{f} = 0.0625 \text{ Hz s}^{-1}$

blades. This data is then transformed into the frequency domain using the information of the excitation frequency sweep rate and the mode resonance frequency determined by the BTT system: $f = \dot{f} \cdot t + f_d$.³⁵ Following this, the damping is determined for each blade by fitting a SDOF system according to eq. (2.43) to the response curves, see fig. 5.17b. The displacement measured by the BTT system is converted to the blade displacement in axial direction in fig. 5.17b using eq. (B.6), but the SDOF would yield the same damping value without this conversion. The fit is realized in MATLAB using a least-square curve fit³⁶, where the damping ratio is a fitting parameter and is hence a result of the fitting process. In addition, κ_{est} is calculated for each blade.

Figure 5.18 shows the blade individual damping for the raw data of fig. 5.17a. In the present work, the damping distribution for the 17 blades of the blisk is often summarized using box-plots. Box-plots allow for the depiction of the median as well as the data spread and the isolated depiction of outliers. The definition of the box-plot is provided in fig. 5.18 on the right-hand side.

³⁵ t is shifted to have f_d at $t = 0 \text{ s}$

³⁶ MATLAB function *lsqcurvefit*

6 Numerical Models

The following chapter introduces the numerical models used in the present work. For the CFD simulations, a comparison of different CFD domains as well as different mesh resolutions is presented.

6.1 FE Simulations

The finite element (FE) simulations in terms of static and modal analyses are conducted using the commercial software *ANSYS Mechanical 19.5*. For the FE mesh, a sector model as well as a full-annulus model of the blisk are available. The full-annulus model is built with 17 identical copies of the sector mesh to avoid any numerical mistuning¹. The sector model is depicted in fig. 6.1a and consists of 52718 quadratic tetrahedral elements² and 86600 nodes. Figure 6.1b shows the full-annulus model with 896206 elements and 1472200 nodes. The mesh resolution was determined in a preliminary study. The sector mesh is created in *ANSYS Workbench 19.5* with a conformal mesh in circumferential direction (1:1 interface). The full mesh is created with the sector mesh and an APDL script in *ANSYS Mechanical 19.5*. Since the aerodynamic damping simulations are conducted with full-annulus CFD models, the FE simulations are also performed with the full-annulus model to simplify the data exchange from FE to CFD simulations. Nevertheless, results from modal analyses with the sector model and cyclic-symmetric interfaces yield results with negligible difference to the full model in terms of natural frequency and mode shape. In order to model the clamping of the blisk on the shaft, elements are fixed in the hub region, which is marked with red fig. 6.1a.

¹Small differences in the FE mesh between sectors, which are inevitable with most meshing algorithms if the complete blisk is meshed at once, can introduce significant mistuning depending on the mesh resolution.

²type SOLID187 [4]

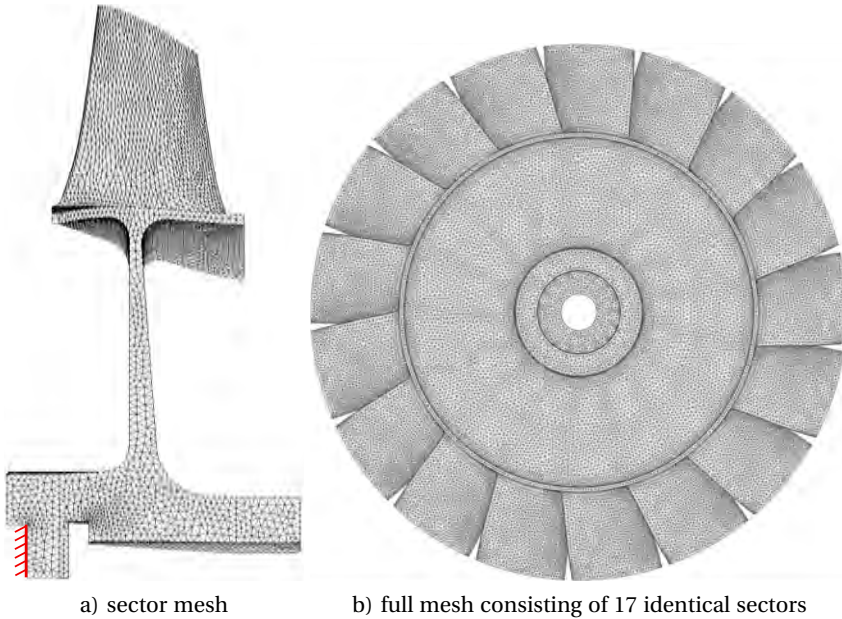


Figure 6.1: Computational meshes for FE simulations

Linear pre-stress analyses³ are performed prior to the modal analyses for different rotational speeds and the FE model allows for the consideration of Coriolis effects. The material properties of 17-4 PH used for all FE simulations are as follows: Young's modulus⁴ $E = 204 \text{ GPa}$, density $\rho = 7800 \text{ kg m}^{-3}$

³The effect of non-linear pre-stress analyses was investigated by Waldherr [219] for a similar blisk, which was also designed by the author of the present work. Waldherr showed that the maximum difference between linear and non-linear analyses for comparable rotational speeds was $< 1.3\%$ in terms of modal assurance criterion (MAC) and $< 0.1\%$ in terms of natural frequency.

⁴Values of the Young's modulus of 17-4 PH between 190 GPa and 210 GPa are reported in literature, depending on the exact composition of the alloy. The chosen value of $E = 204 \text{ GPa}$ is a result of a preliminary study, where the differences in natural frequency between the FE and the experimental modal analysis were minimized for the first mode family

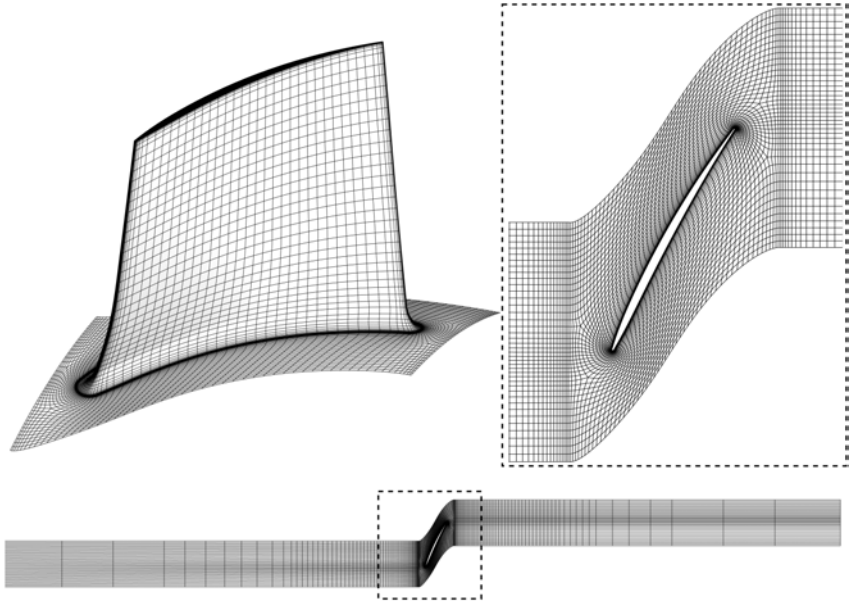


Figure 6.2: Detailed view of the medium CFD sector mesh, CFD model V2.

and Poisson's ratio $\nu = 0.272$. A thorough validation of the FE model is presented in chapter 7.

6.2 CFD Simulations

For the calculation of the flow field as well as the estimation of the aerodynamic damping, computational fluid dynamics (CFD) simulations are performed using the commercial flow solver *ANSYS CFX 2020 R1*, which numerically solves Reynolds-averaged Navier-Stokes (RANS) equations. *ANSYS CFX* is a pressure-based finite volume solver using a coupled multigrid scheme. A thorough introduction to the field of CFD is provided for example in Laurien and Oertel [135]. The steady flow-field is simulated by means of steady-state

RANS simulations. The aerodynamic damping simulations are conducted using the harmonic balance method in the frequency domain, as introduced by Hall et al. [91], along with a moving mesh approach. The harmonic balance method allows for a significant speed-up in the simulation time of unsteady flow-fields with periodic disturbances, such as vibrating blades, compared to simulations in the time domain⁵. Comparisons of the harmonic balance method and unsteady time domain simulations with focus on aerodynamic damping are presented for example in Buchwald et al. [27] or Vogel et al. [214] for *ANSYS CFX*. Furthermore, a preliminary study was conducted for the present case revealing insignificant differences between both methods in terms of aerodynamic damping. The working fluid is air, modeled as perfect gas⁶. The $k-\omega$ SST (shear stress transport) turbulence model by Menter [158] is used for all simulations.

The discretization of the flow field is done with hexahedral meshes generated using the commercial software *NUMECA AutoGrid 11.2*. An example of a sector CFD mesh is shown in fig. 6.2. The mesh consists of several H-blocks along with an O-grid in the vicinity of the blade for an improved mesh refinement and mesh quality in the boundary layer region. The fillet radius at the blade root is included in the CFD meshes. The tip gap features a conformal mesh without an interface. The size of the tip gap is kept constant at 0.5 mm, representing the tip gap at design speed, see fig. 5.6. The axial gaps in the hub upstream and downstream of the blisk are neglected in the CFD simulations, but the rotating and steady parts of the hub are considered with predefined wall velocities. While steady-state simulations are conducted with a sector mesh and a periodic interface in circumferential direction, a full-annulus mesh is used for aerodynamic damping simulations, which consists

⁵The harmonic balance methods makes use of the fact that most flow fields of turbomachinery are periodic in time. As a consequence, the local variables of the unsteady flow field can be approximated with a discrete Fourier series and the RANS equations can be formulated as a steady problem in the frequency domain resulting in a speed up of one or two orders of magnitude compared to standard transient RANS simulations [91]. The number of required Fourier coefficients to sufficiently reproduce the unsteady flow field strongly depends on the type of unsteadiness. For aerodynamic damping simulations in flow fields without varying inlet conditions or periodic flow separations, one Fourier coefficient ("one harmonic") at the fundamental blade vibration frequency is sufficient.

⁶Ideal gas with constant specific heat capacity. The heat capacity ratio γ is 1.4 in the present work.

of 17 identical sectors of the steady-state setup. Total values in terms of pressure and temperature are used as boundary condition at the inlet and a mean static pressure is defined at the outlet. The aerodynamic damping simulations are initialized with converged steady-state simulations and the aerodynamic damping value is continuously evaluated during the simulations to monitor convergence.

The complex displacement of blisk modes is prescribed at the blade surfaces using the superposition of the two real modes of a double mode and user expressions⁷, see appendix C. The calculation of aerodynamic damping is done using the traveling wave mode (TWM) method, which requires one simulation per mode, and the energy method, see section 2.4.4.

In the next sections, different detail levels of the CFD model with regards to the meridional contour of the compressor are discussed followed by a study on the mesh resolution.

6.2.1 Influence of Different CFD Domains

While the test section of the compressor in the region of the blisk represents a completely axial, isolated rotor case, there are structures upstream of the test section (upstream struts and the inlet bulb), as well as downstream (downstream struts and the diffuser), which might alter the strictly axial flow field of the rotor. In order to assess and quantify possible influences, four different CFD sector models are created, which are illustrated in fig. 6.3. V1 is the simplest model, where the CFD domain extends from closely upstream of measurement plane E1 to E3 and only the rotor blade is included, which represents a classic isolated rotor setup. This setup is extended in V2 by significantly increasing the axial length of the domain and adding buffer layers upstream and downstream. Buffer layers⁸ represent an intentional coarsening of the mesh towards the inlet and outlet in meridional direction, which is beneficial in terms of numerical reflections of unsteady pressure waves.

⁷User expressions are used because there was no built-in function to prescribe complex mode shapes at blade boundaries in earlier *ANSYS CFX* versions. This has changed for example for *ANSYS CFX 2020 R1* and a comparison was made between the definition of displacement with user expression and the built-in function, leading to the same results.

⁸also referred to as mesh inflation or grid stretching [164]

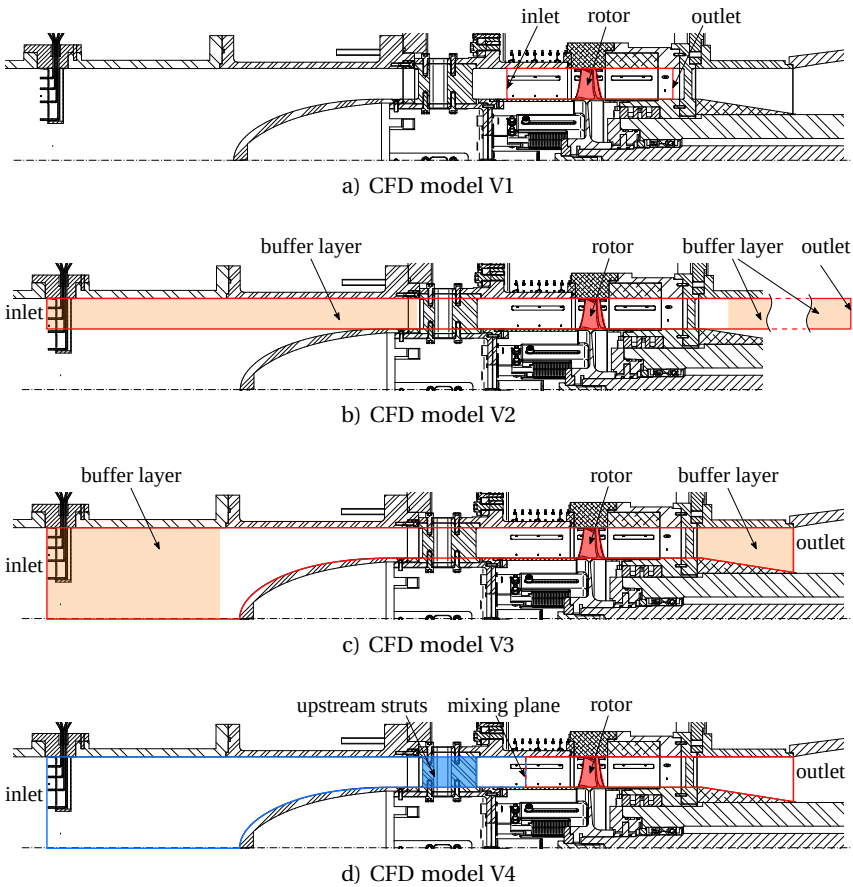


Figure 6.3: Four CFD models with different levels of detail regarding the meridional contour of the compressor

While this is not necessary for steady-state simulations, Müller et al. [164] clearly show that buffer layers are required for aerodynamic damping simulations using *ANSYS CFX*, as the implemented non-reflective boundary conditions do not sufficiently suppress spurious numerical reflections. The inlet of V2 is placed at the position of the total pressure rake in the experiment. For V3, the inlet blub as well as the outlet diffuser are considered in the geometry of the CFD domain. The most detailed CFD model is V4, where the inlet struts are included in a standing CFD domain upstream of the rotor domain. Both domains are connected with a mixing-plane. A fifth CFD model with an additional domain downstream of the rotor including the downstream cylindrical struts was investigated in a preliminary study, but showed very slow convergence due to the wakes of the downstream struts. Since no difference was observed regarding the flow field in the region of the rotor for this CFD model compared to V4, it was not further considered in the present work.

In order to compare the influence of the different CFD models on integral results, a speed line in terms of static pressure ratio is depicted for each model in fig. 6.4a. For reasons of completeness, experimental results are included as well, but a detailed comparison of CFD and experimental results at multiple operating points and inlet conditions is presented and discussed later in chapter 9. To ensure a fair comparison, the same mesh resolution is used for all CFD setups, corresponding to the resolution of the medium mesh in the subsequent section. Figure 6.4a shows that all four CFD models agree well with the experimental results. Furthermore, there are minor deviations between the models with V1 showing the largest difference to the experiment. A comparison of the flow field is provided in figs. 6.4b to 6.4e. The CFD results are averaged in circumferential direction. Due to the inlet position of V1 right at the measurement plane E1, V1 is not able to reproduce the boundary layer distribution present for all other models, see fig. 6.4b. Downstream of the rotor, the Mach number distribution is almost the same for all models, see fig. 6.4c, but there are visible differences in the static pressure and flow angle distribution for V1. All other models show insignificant differences in comparison.

The influence of the different CFD domains on the aerodynamic damping is evaluated by performing aerodynamic damping simulations at several operating points at design speed with the CFD models V2 and V3. V4 is excluded

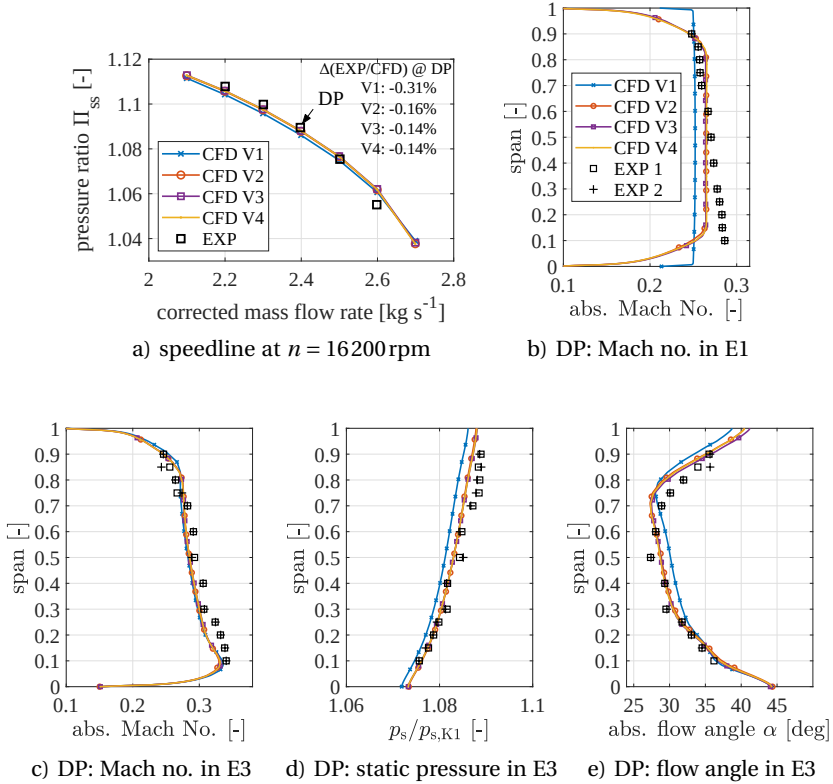


Figure 6.4: Different CFD models: Speedline and detailed comparison of the flow field, $p_{t,in} = 500$ mbar

from the investigation since there was no significant difference in the steady flow field between V4 and V3, which share the same meridional contour, but with the advantage of having no mixing-plane in V3. Furthermore, V1 is not considered for aerodynamic damping simulations due to the lack of buffer layers and thus potential inaccuracies from numerical reflections. The results are presented in fig. 6.5, where the normalized aerodynamic damping

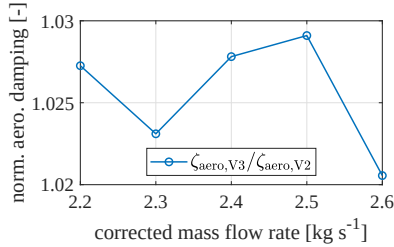


Figure 6.5: Comparison of CFD models V2 vs V3 in terms of aerodynamic damping, $n = 16200$ rpm, $p_{t,in} = 500$ mbar

Table 6.1: CFD meshes for CFD model V2: Number of elements

	very coarse	coarse	medium	fine
sector mesh (steady-state CFD)	135872	298048	650880	1203328
360° mesh (aero. damping CFD)	2309824	5066816	11064960	20456576
average y^+	2.40	0.62	0.62	0.62

is depicted for mode B1ND2BTW. It can be shown that maximum difference between both CFD models is below 3 %.

6.2.2 Influence of CFD Mesh Resolution

The influence of the CFD mesh resolution is investigated with the CFD model V2. Four different meshes are created, see table 6.1, with the intention to approximately double the number of elements for each mesh. Table 6.1 also provides the number of elements for the 360° meshes, which are used for the determination of mesh resolution influence on the aerodynamic damping calculation. All meshes feature an average y^+ -value below 1, except for the very coarse mesh. Mesh refining is realized by evenly increasing the number of elements in all three spatial directions. Details of the medium mesh can be seen in fig. 6.2.

Similar to the investigation of the CFD models, fig. 6.6 presents a speed line for all meshes along with detailed depictions of the flow variables in the mea-

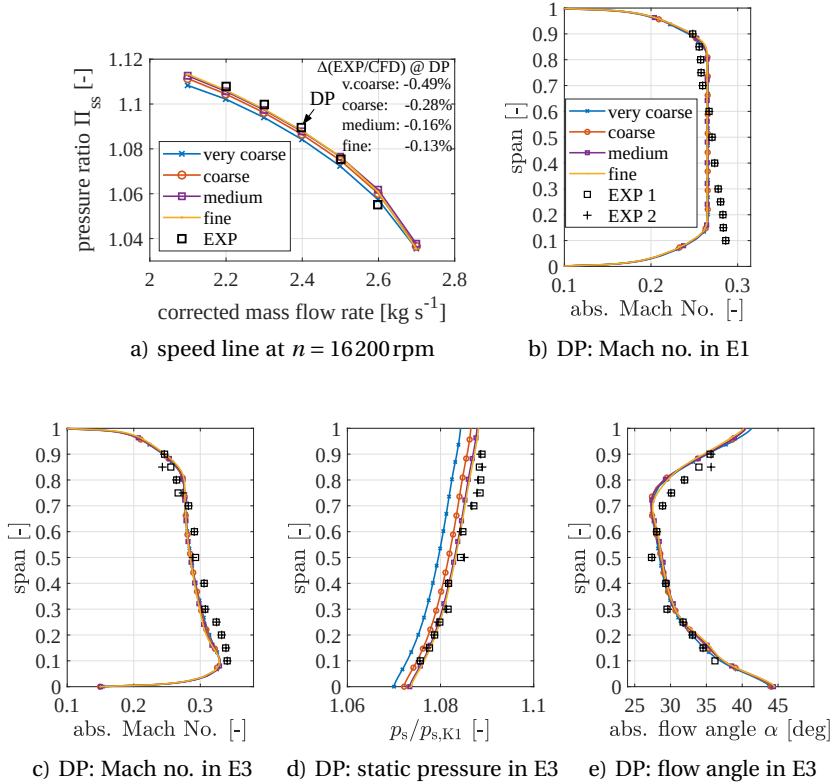


Figure 6.6: Different CFD mesh resolutions for CFD model V2: Speed line and detailed comparison of the flow field, $p_{t,\text{in}} = 500$ mbar

surement planes E1 and E3. The comparison shows insignificant differences between the meshes for the Mach number distribution in E1 and E3 as well as the flow angle distribution in E3. Nevertheless, there is a trend of underprediction of the static pressure distribution in general with decreasing number of elements, see fig. 6.6d, which is, as a result, also visible in the static pressure ratio in fig. 6.6a.

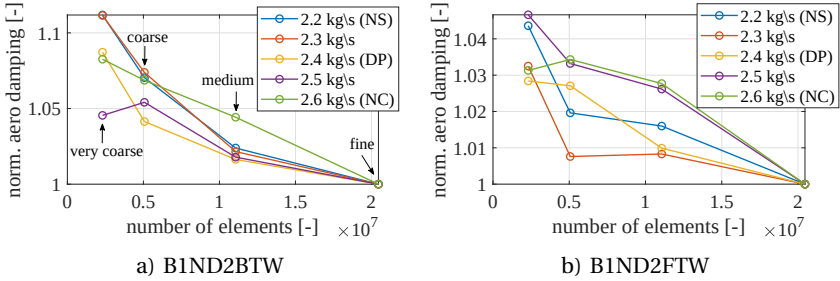


Figure 6.7: Influence of the CFD mesh resolution on aerodynamic damping, CFD model V2, $n = 16200$ rpm, $p_{t,in} = 500$ mbar

The influence of the mesh resolution on the unsteady flow field is investigated by performing aerodynamic damping simulations for two modes with all meshes and at all operating points on the speed line at design speed. The investigation is focused on the CFD model V2. Figure 6.7 shows the aerodynamic damping for the modes B1ND2BTW and B1ND2FTW depending on the mesh resolution. The aerodynamic damping is normalized at each operating point with the value of the fine mesh. It becomes apparent that the aerodynamic damping is over-predicted by coarsening the mesh and that there is no complete mesh independence observable for the chosen mesh resolutions. However, the differences are moderate, in particular for the medium mesh, which shows a maximum difference of 4.4 % for B1ND2BTW and 2.8 % for B1ND2FTW, respectively. Further mesh refinements are not investigated due to limitations in the computational resources.

6.2.3 Influence of Displacement Amplitudes and Number of Harmonics

It is assumed that the aerodynamic damping is independent of the mode displacement amplitude for reasonably low displacement amplitudes. In many studies, 1 % of the blade chord length is recommended as displacement amplitude for the aerodynamic damping simulations. Figure 6.8 shows the nor-

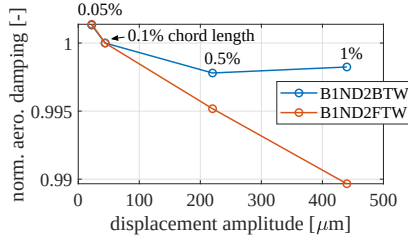


Figure 6.8: Influence of the prescribed mode displacement amplitude, CFD model V2, medium mesh, $n = 16200\text{rpm}$, $p_{t,\text{in}} = 500\text{mbar}$

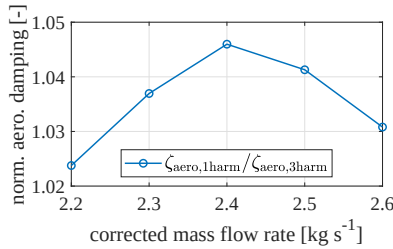


Figure 6.9: Influence of the number of harmonics for B1ND2BTW, CFD model V2, medium mesh, $n = 16200\text{rpm}$, $p_{t,\text{in}} = 500\text{mbar}$

malized aerodynamic damping for different amplitudes and two modes. It can be concluded from this depiction that indeed the simulated aerodynamic damping of the present case is independent of the displacement amplitude within an error margin of 1 %.

In order to quantify the influence of the number of Fourier coefficients in the aerodynamic damping simulations using the harmonic balance method, simulations at different operating points are conducted with one as well as three harmonics of the fundamental blade vibration frequency. The results are summarized in fig. 6.9. It can be noted that the maximum error is approximately 4.5 % for a simulation with one harmonic.

6.2.4 Conclusions and Final CFD Setup

Based on the investigations presented above, it is decided to use CFD model V2 for all following investigations. While V3 better represents the meridional contour of the compressor, the differences in the prediction of the steady-state flow field as well as the aerodynamic damping are negligible. In addition, V3 showed convergence issues for some modes and operating points, with high residuals mainly observed in the region with no hub upstream of the inlet bulb. Convergence issues were detected for V3 in particular for B1ND2FTW, where an acoustic resonance is detected in the measurements, see section 11.2, and it is assumed that this contributes to the numerical instability for this CFD setup.

In terms of mesh resolution and the number of harmonics, a compromise regarding the computational time and the file size is selected with the medium mesh and a setup with one harmonic. Since the aerodynamic damping is simulated for multiple modes and the complete compressor map, a reasonable file size⁹ along with moderate computational effort is required. For the selected setup, a file size of 23 GB per simulation is achieved at a combined error in the range of approximately 10 %, compared to the fine mesh and three harmonics. The displacement amplitude is kept constant for all simulations with 0.1 % of the blade chord length ($=44.5\mu\text{m}$) as there is no significant influence on the displacement amplitude on the aerodynamic damping observed.

⁹the file size linearly correlates with the number of harmonics and the number of mesh elements

7 Structural Dynamics of the Blisk and FE Validation

In order to obtain reliable measurements of aerodynamic damping, it is very important to know the exact mode shapes and the degree of mistuning of the investigated modes in the first place. The following chapter presents a series of measurements, where the structural dynamic of the blisk is investigated at different clamping conditions and under rotation. First, an experimental modal analysis (EMA) is conducted in an isolated manner, where a free-free boundary condition is modeled for the blisk in the experiment. Second, the the EMA is reproduced but with the blisk mounted on the shaft of the test facility, which allows for the determination of the influence of the clamping condition on the normal modes. Finally, results from mode excitation experiments under rotation are evaluated (Campbell diagram). All measurement results are compared to FE simulations to validate the numerical model.

7.1 Experimental Modal Analysis: Free-Free

The first set of measurements was performed in cooperation with Polytec GmbH at the RoboVib laboratory in Waldbronn, Germany. Parts of the measurement results were first published by Buchwald et al. [28] as well as by Waldherr [219]. The experimental setup, as depicted in fig. 7.1a, comprises a three-dimensional scanning laser Doppler vibrometer (3D-SLDV) PSV-500-3D from Polytec and an automated modal hammer SAM1 from NV tech. The vibrometer has three infrared laser with a wave length of 1550 nm ¹. A robot arm is used to initially position the 3D-SLDV. This position is maintained for all measurements. The main advantage of a non-contact 3D-SLDV system as compared to for example acceleration sensors is that no mass and hence no mistuning is added to the blisk. In addition, a large set of measurement

¹ $< 10\text{ mW}$, class 2, laser is invisible. Three green pilot laser are used as well for the measurement laser positioning with a wave length of 520 nm ($< 1\text{ mW}$), also class 2. [181]

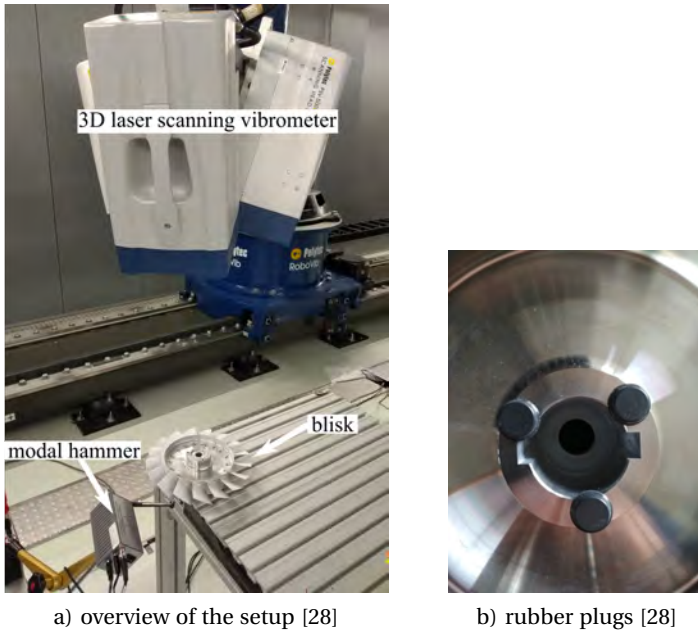


Figure 7.1: Experimental setup for free-free EMA at the Polytec RoboVib laboratory

points can be defined and measured in reasonable time². The blisk is placed onto a table by means of three rubber plugs, see fig. 7.1b, to model an almost free-free boundary condition³. The 3D-SLDV is facing the suction side of the blisk. In order to identify all relevant mode shapes, ten measurement points are defined on each blade, see fig. 7.2. Since a 3D-SLDV is utilized, the mode displacement can be measured in all spatial directions resulting in $10 \times 17 \times 3 = 510$ degrees of freedom per measurement series. For each mea-

²85 minutes for one EMA

³A free-free boundary condition means that no external loads or constraints are present, which can be easily modeled in FE simulations, but is challenging in experiments. Besides rubber plugs, the blisk could also be positioned with strings or springs. The later was investigated in a preliminary study but resulted in lower measurement quality for the present case.



Figure 7.2: ITSM-ASB1, EMA measurement points in red and blue [28]

surement point, the blisk is excited using the modal hammer, which impacts on the hub ring of the disk on the pressure side. Subsequently, the vibration is measured for 20 s yielding a frequency resolution of 50 mHz. The sampling frequency is 25 kHz. The excitation force at the modal hammer is measured as well and the data is acquired using the proprietary software *PSV 9.4.1* from Polytec, which is also used to calculate frequency response functions. The post-processing in terms of normal mode extraction is done in *PolyWave 19.1* from Polytec, where the Poly-LSCF algorithm is applied [87].

In fig. 7.3, the identified natural frequencies of the EMA are plotted along with the corresponding dominant nodal diameter, which are determined with a spatial discrete Fourier transformation (DFT) of the axial mode displacement at the blue measurement points near the blade tips, see fig. 7.2. 99 out of 112 simulated natural frequencies are detected in the measurements in the range of 0 to 10 kHz, excluding the six rigid body modes due to the free-free setup at nodal diameter of 0 and 1 and 0 Hz.⁴ While it is assumed that the excitation at the hub ring is able to excite all modes in the specified

⁴The rigid body modes are also visible in the measurements, but not exactly at 0 Hz due to the non-zero stiffness of the rubber pads. However, these modes are not relevant for the present investigation.

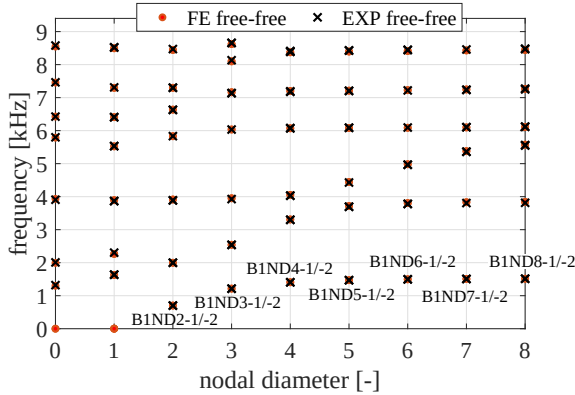


Figure 7.3: Natural frequencies vs. nodal diameter for the free-free boundary condition

range, some double modes cannot be detected due to very low mode splitting⁵. Nevertheless, the agreement between the measured and the simulated natural frequencies, as shown in fig. 7.3, is excellent. In order to quantify the agreement, table 7.1 provides a comparison of the measured and the simulated natural frequencies for the first mode family showing a maximum difference of $\leq 0.3\%$ for the free-free setup. B1ND0, B1ND1-1 and B1ND1-2 do not exist as they become rigid body modes in the free-free setup.

Regarding the spatial DFT, two examples are shown in fig. 7.4 for B1ND2-1 and B1ND8-1. In the plot at the top left of fig. 7.4, the axial displacement clearly shows a displacement pattern corresponding to a nodal diameter two mode, which is confirmed by the spatial DFT components of the displacement on the top right. For B1ND8-1, the dominant nodal diameter component still corresponds to the expected nodal diameter but there is a second component present, which arises due to mistuning effects breaking up the

⁵Mistuning causes a frequency split of the double modes resulting in a double peak in the frequency response curve, which helps to identify and distinguish double modes. If the frequency split is very low and/or damping is too high, only a single peak is visible.

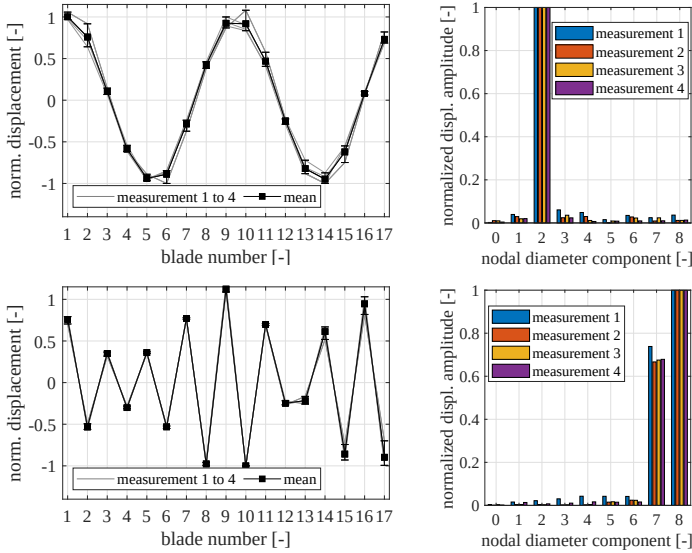


Figure 7.4: Mode displacement and nodal diameter component of B1ND2-1 (top) and B1ND8-1 (bottom)

distinct nodal diameter pattern⁶. Furthermore, fig. 7.4 shows the high reproducibility of the EMA, which is conducted four times in total.

The results of the spatial DFT can be utilized as an indicator for mistuning and fig. 7.5 gives an overview of the nodal diameter components for the first mode family in the free-free setup⁷. Most of the modes only show one distinct nodal diameter component and they can be considered as tuned in the first approximation. However, the effect of mistuning increases with higher modal density and in particular the DFT of the modes with seven and eight nodal diameter comprises more than one significant nodal diameter com-

⁶these mistuned modes are also referred to as modified cyclic symmetry modes compared to cyclic symmetry modes in the tuned case [125]

⁷Mode numbers instead of the mode names are used to simplify the figures. The corresponding mode names can be taken from table 7.1

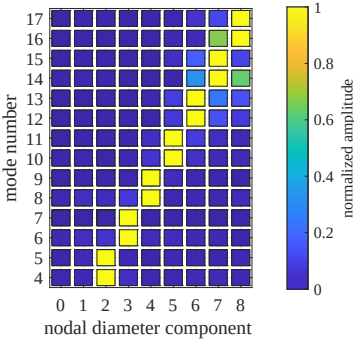


Figure 7.5: Nodal diameter components of mode family B1 free-free

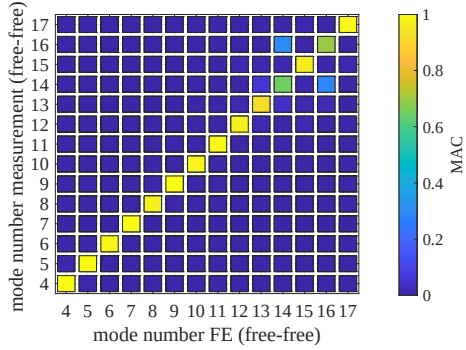


Figure 7.6: Mode family B1 free-free: Comparison of measurement and FE simulation with MAC

ponent. The mistuning of the higher modes becomes even more apparent by comparing the experimentally determined mode shapes to the tuned FE simulation results. Therefore, the modal assurance criterion (MAC) [2] is used

$$\text{MAC} = \frac{\left| \underline{\phi}_1^\top \underline{\phi}_2 \right|^2}{\left(\underline{\phi}_1^\top \underline{\phi}_1 \right) \left(\underline{\phi}_2^\top \underline{\phi}_2 \right)}, \quad (7.1)$$

where a MAC value of 1 (or 100 %) means that two modes are linearly dependent and a MAC value of 0 (or 0 %) that both modes are orthogonal to each other. The MAC comparison for the first mode family is depicted in fig. 7.6. Since the orientation of the normal modes in the FE simulations is arbitrary and does not necessarily match with the measured modes, the simulated double modes are individually rotated with $\phi_{\text{rot}} = \Re(\tilde{\phi}_{12} e^{i\varphi})$ to ensure a fair comparison. It can be seen in fig. 7.6 that the tuned FE model captures the experimentally determined normal modes very well ($\text{MAC} \geq 0.97$) up to mode 12 (B1ND6-1), but the agreement worsens due to the mistuning of the higher modes with a minimum of $\text{MAC} = 0.65$ for mode 14 (B1ND7-1).

Table 7.1: Mode family B1, free-free and mounted conditions. Comparison of numerical and experimental results.

mode no.	mode	blisk mounted			free-free		
		$f_{\text{FESim.}}$	f_{EXP}	Δf	$f_{\text{FESim.}}$	f_{EXP}	Δf
		[Hz]	[Hz]	[-]	[Hz]	[Hz]	[-]
1	B1ND0	941.1	975.9	-3.6%	-	-	-
2	B1ND1-1	692.0	-	-	-	-	-
3	B1ND1-2	692.0	-	-	-	-	-
4	B1ND2-1	702.2	700.2	0.3%	701.8	699.5	0.3%
5	B1ND2-2	702.2	700.9	0.2%	701.8	700.3	0.2%
6	B1ND3-1	1216.0	1211.2	0.4%	1216.0	1212.1	0.3%
7	B1ND3-2	1216.0	1211.8	0.3%	1216.0	1212.7	0.3%
8	B1ND4-1	1408.9	1404.6	0.3%	1408.9	1405.8	0.2%
9	B1ND4-2	1408.9	1405.4	0.2%	1408.9	1406.5	0.2%
10	B1ND5-1	1471.9	1467.9	0.3%	1471.9	1469.2	0.2%
11	B1ND5-2	1471.9	1468.7	0.2%	1471.9	1469.8	0.1%
12	B1ND6-1	1498.0	1493.9	0.3%	1498.0	1495.1	0.2%
13	B1ND6-2	1498.0	1494.6	0.2%	1498.0	1496.0	0.1%
14	B1ND7-1	1510.0	1505.0	0.3%	1510.0	1506.3	0.2%
15	B1ND7-2	1510.0	1507.0	0.2%	1510.0	1508.3	0.1%
16	B1ND8-1	1514.8	1511.6	0.2%	1514.8	1512.9	0.1%
17	B1ND8-2	1514.8	1513.8	0.1%	1514.8	1515.0	-0.0%

7.2 Experimental Modal Analysis: Blisk Mounted

One of the main advantages of ACTIVE is the optical accessibility of the mounted blisk by removing the piping upstream of the compressor. This is used to perform an EMA with the exact clamping conditions as used in operation. An overview of the experimental setup is shown on the left in fig. 7.7, where the SLDV system is facing the suction side of the blisk in axial direction. In order to simplify the measurement setup at the ITSM, it was decided to use the same base measurement system PSV-500 from Polytec as for the free-free measurements, but with only one measurement head, reducing the system to a 1D-SLDV. Since all modes of interest and in particular the first

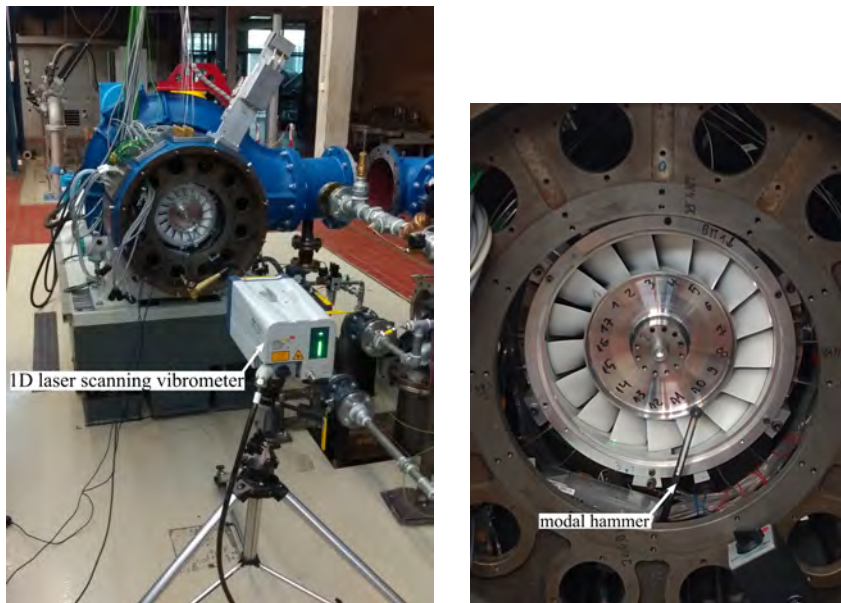


Figure 7.7: Experimental modal analysis of the blisk ITSM-ASB1 mounted in the test facility.

mode family show significant displacements in axial direction, the 1D EMA setup is considered to be adequate. The blisk is excited using the same modal hammer, see fig. 7.7 right, and at the same circumferential position as in the free-free measurements, but this time on the suction side due to limited accessibility of the pressure side. The configuration in terms of sampling frequency (25 kHz) and frequency resolution (50 mHz) is maintained. The measurements are repeated two times in total and the differences between the measurements are insignificant, hence only the results of one measurement is shown in the following.

Figure 7.8 depicts all natural frequencies detected in the measurements along with the results from the FE simulation and the free-free measurements. 86 natural frequencies are identified in the measurements compared to 118 in

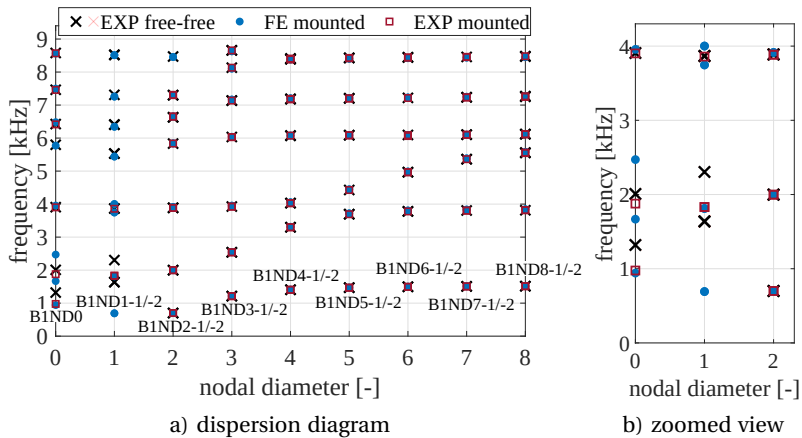


Figure 7.8: Natural frequencies vs. nodal diameter for the blisk mounted on the shaft in comparison to the free-free measurement

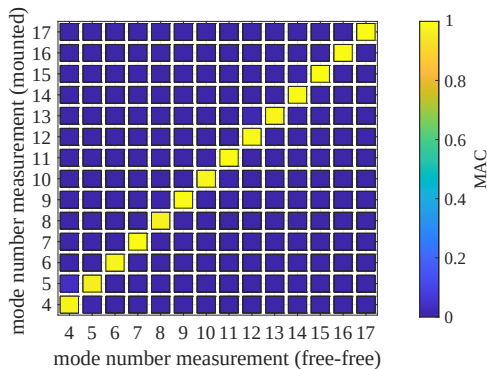


Figure 7.9: Comparison of the first mode family B1 with different clamping conditions. Mode numbers are translated in table 7.1

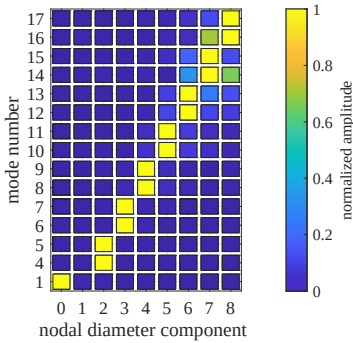


Figure 7.10: Nodal diameter components of mode family B1, blisk mounted

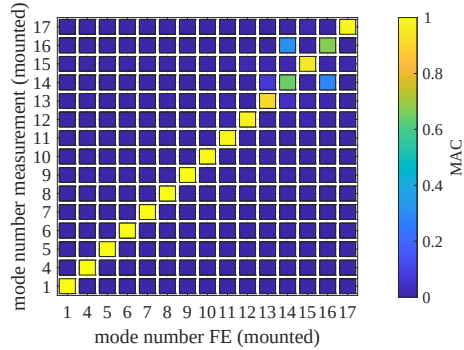


Figure 7.11: Mode family B1, blisk mounted: Comparison of measurement and FE simulation by means of MAC

the FE simulation for the mounted blisk. This is partially due to the same reason as explained for the free-free measurement, where not all modes of the double peaks are identified, but it is apparent that in particular modes with zero and one nodal diameter are not detected in the EMA with the blisk mounted. The shaft bearings introduce significant damping to these modes, which would necessitate a stronger excitation for the measurement. However, the ND1 modes of the first mode family is also not detected using strong magnetic excitation later in the rotating operation. A change in the excitation position was not investigated for the EMA.

Comparing the measurements to the free-free measurements, it can be concluded that the clamping condition of the blisk only has a significant influence on modes with zero and one nodal diameter and below 4 kHz. This is in agreement to similar investigations, for example [228], where an influence of the shaft and bearings is identified only for modes with zero and one nodal diameter. For all the other modes, the agreement to the free-free measurement is very good. Focusing on mode family B1, all modes are detected except for B1ND1-1 and B1ND1-2. The difference in the measured frequencies for different clamping conditions is negligible, see also table 7.1. In addition, the MAC comparison in fig. 7.9 shows an excellent agreement ($\text{MAC} \geq 0.97$)

between the different measurements.⁸. Hence, the mistuning pattern is not changed and the nodal diameter components in fig. 7.10 are very similar to the free-free measurement (fig. 7.5).

The FE simulation with a fixed boundary condition, see fig. 6.1a, is able to reproduce the measured natural frequencies well and even the nodal diameter zero mode B1ND0 is captured. A quantitative comparison for the first mode family is provided in table 7.1, where a maximum frequency error between the measurement and the simulation is shown with $\leq 0.4\%$, except for B1ND0 with -3.6% . Since the comparison of measurements (free-free and mounted) shows very high MAC values, the MAC comparison of FE simulation and measurement for the mounted blisk does not change compared to free-free setup, c.f. fig. 7.6 and fig. 7.11. The FE mode shapes of the mounted blisk are shown in figs. B.2 and B.3.

7.3 Campbell Diagram

So far, the normal modes have been investigated with the blisk at rest. In order to determine the influence of centrifugal loads on the natural frequencies, a Campbell diagram is calculated by means of FE simulations and depicted in fig. 7.12 for the first mode family. The FE simulations are conducted with and without the consideration of Coriolis forces and the blisk is fixed at shaft position (mounted blisk FE model). In addition, fig. 7.12 shows all resonance frequencies determined by means of electromagnetic excitation of the blisk in vacuum and at various rotational speeds. In contrast to the EMA measurements, where standing modes are investigated, traveling wave modes are excited for the Campbell diagram. First, it can be seen that all natural frequencies of the first bending mode family increase with rotational speed and that the agreement of the measurements and the FE simulations is very good, comparable to the EMA results. A significant offset is only visible for B1ND0, which was also detected in the EMA. This offset is attributed to the simplified modeling of the clamping conditions at the blisk hub, c.f. fig. 6.1a, which modifies the natural frequency. Nevertheless, the influence

⁸B1ND0 is not included in the comparison since this mode does not exist in the free-free condition. The MAC value is only calculated with the axial displacement ($10 \times 17 = 170$ DOF) as a 1D-SLDV is used for the measurement series with the mounted blisk.

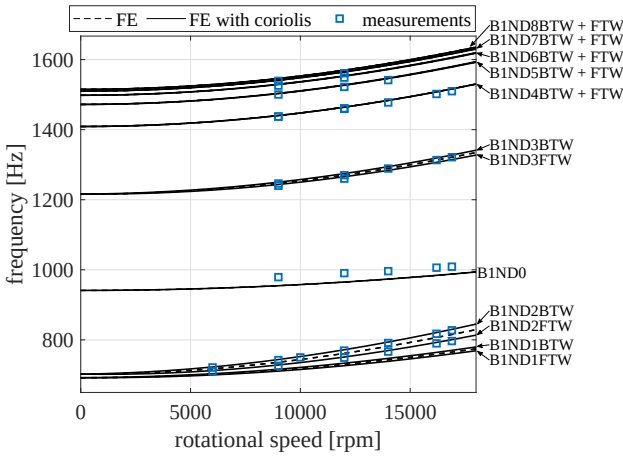


Figure 7.12: Campbell diagram for ITSM-ASB1 for the first mode family B1

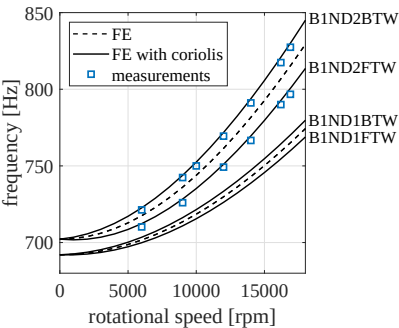


Figure 7.13: Campbell diagram for ITSM-ASB1 with focus on B1ND2

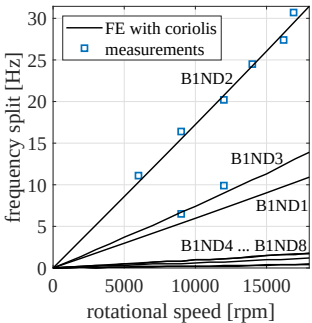


Figure 7.14: Frequency split of mode family B1 due to Coriolis effect

on the mode shape is assumed to be insignificant, as shown in fig. 7.11. Second, the Campbell diagram clearly shows a split of the modes, in particular for B1ND1, B1ND2 and B1ND3, into forward and backward traveling modes. A detailed view on the frequency split of B1ND2 is shown in fig. 7.13. The frequency split in the measurements can be explained by the influence of the Coriolis force, see section 2.5, as the FE simulations with activated Coriolis effect agree very well with the measured frequencies. Furthermore, the amount of frequency split is evaluated and depicted in fig. 7.14, showing a linear increase, which is a clear sign of the Coriolis effect [185]. It should be noted that the backward traveling wave modes always have a higher natural frequency compared to the forward traveling wave modes, which is in agreement to the findings of Ruffini [185].

7.4 Conclusions

The comparison of the measurements and FE simulations for various clamping conditions and under rotation lead to the conclusion that the FE model is considered validated since only minor differences are present. While there is minor influence of mistuning visible for mode B1ND6-2 and more significantly for modes with 7 and 8 nodal diameter, the tuned FE model sufficiently reproduces the mode shapes for all other modes of the first bending mode family.

The Campbell diagram measurements impressively show the influence of the Coriolis effect on the natural frequencies, in particular for B1ND2 and B1ND3. The FE simulations are able to capture this effect almost perfectly. It is assumed that this investigation represents one of the first measurements, which show for an axial compressor blisk at real operating conditions that Coriolis forces have to be considered as they introduce frequency splits of the double modes of up to 30 Hz in the design rotational speed for the present case. The measurements also provide clarity on the direction of traveling wave modes if the double modes are split by the Coriolis effect. The backward traveling wave mode shows a higher frequency in the rotating frame of reference compared to the forward traveling wave. This is sometimes not clearly stated in other publications and care should be taken if only FE simulation results are used, where the reference frame can be changed easily. The

direction is not only determined by the BTT system but also validated using a laser Doppler vibrometer and a derotator, as described in the following chapter.

8 Validation of the Blade Tip Timing System

Since the experimental determination of the aerodynamic damping in the present work is mainly based on the blade tip timing (BTT) measurements¹, a validation of the BTT system is presented in the following.

There are several publications regarding the validation of BTT systems and most of them rely on concurrent strain gauge measurements, see e.g. [43, 96, 126, 154, 187]. While strain gauges are a well established measurement method, it was decided for the present work to avoid any additional mistuning by equipping the blisk with strain gauges and rather use a non-intrusive BTT measurement system. Therefore, a unique experimental setup was created, where the blisk is operated at vacuum ($p_t = 5 \text{ mbar}$) and the blade vibration is measured by means of a 1D-SLDV, which is operated through an optical derotator. As shown in fig. 8.1, the piping upstream of the compressor is removed and a blank flange with a glass window is installed acting as an optical access to the blisk. A second blank flange is mounted downstream of the compressor allowing for the evacuation of the compressor itself. The 1D-SLDV is placed approx. 3 m upstream of the compressor on the axis of rotation. Since the laser vibrometer can only be used for stationary objects, an optical derotator PSV-A-440 is used, which is synchronized with the blisk rotation and enables the measurement of rotating objects. The derotator consists of a rotating Dove prism² and the synchronization is realized using the 500/rev signal of the test facility. As a result, the 1-SLDV system can be used in the same way as for stationary objects while the laser is continuously following the rotating object. In addition to the 1-SLDV, a second laser vibrometer is coupled into the derotator and always placed at the same position on the blisk (blade 1) to provide a reference measurement for the determination of the blade individual phase.

¹unsteady pressure data can be used as well, see section 11.5

²A Dove prism is used to invert monochromatic images. The Dove prism is rotated at 50 % of the rotational speed of the blisk as it doubles the rotation of an image due to its design [160, 182].

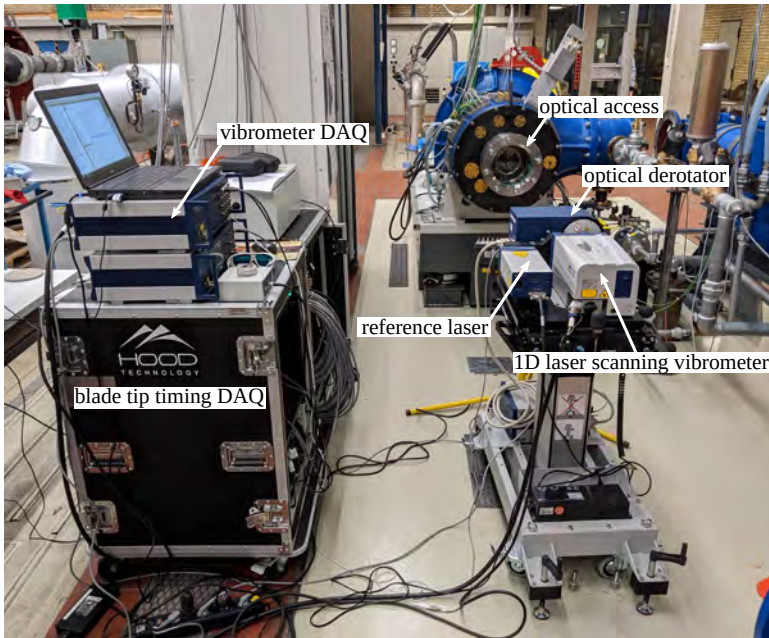


Figure 8.1: Experimental setup for BTT validation measurements

While there are a few publications using a SLDV with a derotator to measure blade displacements of academic blisks at comparably low rotational speeds [64, 88, 102, 228] and in one publication in combination with a BTT system [44], the investigation below represents, to the authors knowledge, the first application of a SLDV to validate a BTT system for a realistic compressor blisk mounted in the compressor and at relevant rotational speeds. One of the main advantages of the derotated 1D-SLDV system is that the frequency spectrum of the rotating blisk can be measured in the rotating frame of reference with a high sampling rate and a high frequency resolution providing the true frequency content of the blisk. This is in contrast to the BTT system, where blisk frequency spectra are strongly reduced in their size with significant aliasing effects due to the limited number of BTT sensors and hence undersampling of the blade vibration.

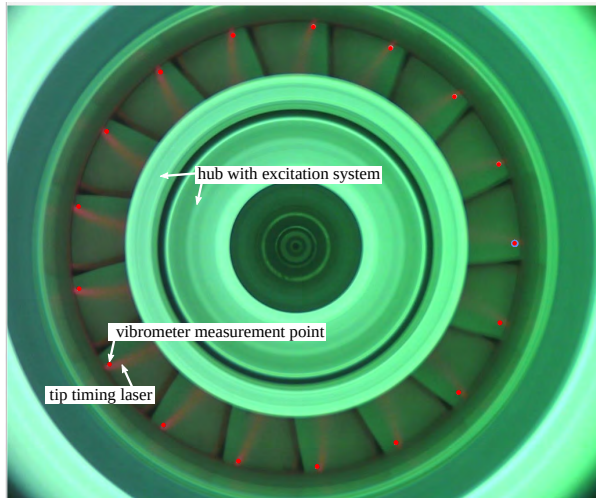
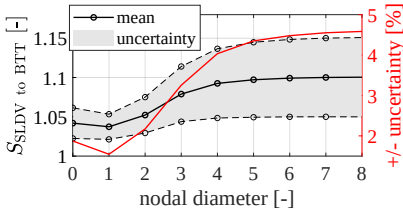
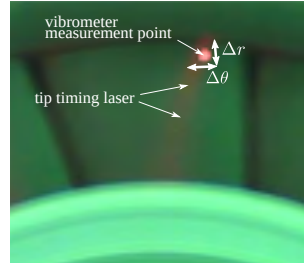


Figure 8.2: Image of the blisk at 6000 rpm in the rotating frame of reference, captured using a standard video camera of the SLDV through an optical derotator. Positions of vibrometer measurement points highlighted in red. BTT1 lasers appear on almost all blades due high exposure time of the video camera.

In order to obtain reliable measurement data, the SLDV has to be aligned accurately with the rotational axis. The alignment is checked with the help of a video camera, which is implemented in the SLDV and thus its video is also derotated by the derotator capturing the image of the blisk in the rotating frame of reference, see fig. 8.2. The video image is also used to position the measurement points in the SLDV software. As shown in fig. 8.2, one measurement point is defined on each blade close to the blade tip. The measurement points are positioned in a way such that they coincide with the tip timing plane BTT1 in axial direction. This is achieved with the help of the tip timing laser, which appear on all blades due to the high exposure time of the video



a) SLDV to BTT scaling factor for the first mode family B1 including uncertainty



b) detailed view of a blade in the SLDV video

Figure 8.3: Uncertainty of measurement point positioning assumed with ± 1 mm (uncertainty ranges and size of the measurement point exaggerated for better visibility in the picture on the right)

camera compared to the rotational speed.³

Since the SLDV measurement points are placed approx. 2 mm radially inwards to avoid any influence of the blade tip, the SLDV displacement measurements have to be scaled to enable a fair comparison to the BTT measurements. The scaling factor is calculated by means of the FE simulations as the ratio of the axial displacement at tip and the axial displacement at the SLDV points. All displacements shown in the following section represent axial displacement amplitudes at the blade tip in the axial plane BTT1, regardless of the measurement system. Furthermore, the placement of the SLDV measurement points is done manually in the video image, where small deviations in the position of the points are inevitable. Therefore, the influence of this uncertainty is evaluated in fig. 8.3a for the first mode family with a range of the SLDV to BTT scaling factor due to an uncertainty of ± 1 mm in radial and circumferential direction, see also fig. 8.3b. The plot shows an increasing un-

³Assuming a frame rate of 30 fps for the video camera and thus an exposure time of 33 ms, the blisk at $n = 6000$ rpm rotates more than three times during the exposure of one image. As a consequence, the laser light of the six tip timing sensors in BTT1 reflected by the blades is visible on all blades. The intensity of the laser on the blades in fig. 8.2 varies due to the brightness of an additional external light source placed next to the compressor to increase the visibility of the blisk in the video.

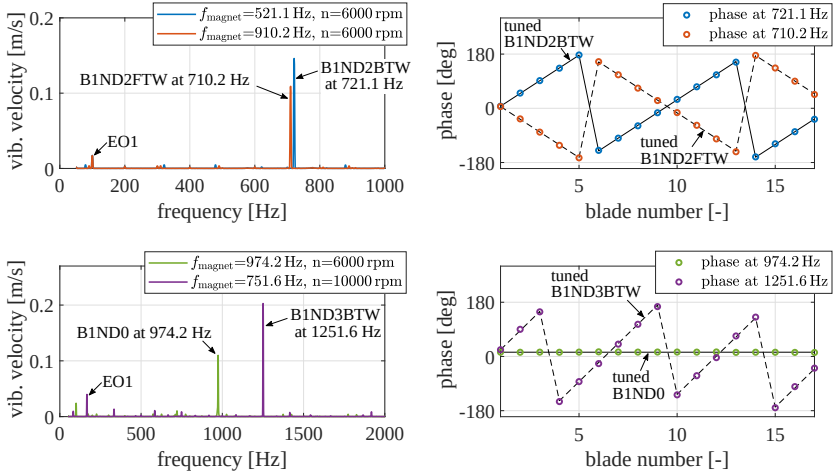


Figure 8.4: SLDV measurements, consecutive excitation of four different modes. Measured phases are compared to the theoretical phase distribution of tuned modes.

certainty for higher nodal diameter, which can be explained by the increasing displacement gradient on the blades for blade dominated modes with several nodal diameter, c.f. figs. B.2 and B.3. The maximum uncertainty in the measured blade displacement is calculated for B1ND8 with $\pm 4.6\%$.

In a first step, the derotated 1D-SLDV system is utilized to analyze the frequency content of the blisk when a certain mode is excited. This is realized by exciting the blisk at a constant frequency and consecutively measuring the displacement at all 17 blades.⁴ Figure 8.4 shows the results of this investigation, where four different modes are excited at two different rotational speeds. In the upper left part of fig. 8.4, the mean frequency spec-

⁴The blisk is excited with a constant frequency for 65 s and each blade is measured three times for 1.28 s resulting in a frequency resolution of 781.25 mHz. The measurement results represent the average of the three measurements.

trum⁵ is depicted for two measurements, where B1ND2 is excited in a forward and a backward traveling wave mode, respectively, at a rotational speed of 6000 rpm. The frequency spectra clearly show that both modes can be excited sufficiently and in an isolated manner, meaning that no other modes are unintentionally excited as well. Only a small contribution of the first excitation order is also visible originating from minor imbalances of the powertrain. Furthermore, the comparison of both frequencies again shows the frequency split due to the Coriolis effect, see also section 7.3. In order to validate the traveling wave direction, the blade individual phase is determined for both measurements at the peak frequency and depicted on the upper right part of fig. 8.4. In addition, the theoretical phase distribution for a tuned blisk is plotted for both modes. First, it can be seen that both modes show insignificant differences to the theoretical phase distributions leading to the conclusion that no significant mistuning is present for B1ND2 regardless of the traveling wave direction. This is in agreement to the results of the experimental modal analysis in chapter 7. Second, the measurements confirm that the backward traveling mode⁶ has an increased frequency in the rotating frame of reference due to the Coriolis effect and vice versa for the FTW mode. The lower part of fig. 8.4 depicts the frequency spectra and phase for the excitation of B1ND0 at 6000 rpm and B1ND3BTW at 10000 rpm. Similar to B1ND2, both modes can be excited without disturbances and both modes show a clear and almost tuned phase distribution.

In a second measurement series, the sensitivity and accuracy of the BTT system is investigated by exciting the same mode to different amplitudes and comparing the measured displacement of the BTT and SLDV systems. The results of this investigation are presented in fig. 8.5, where B1ND2BTW is excited using excitation frequency sweeps at a constant rotational speed of $n = 6000$ rpm. The sweep rate is 66.7 mHzs^{-1} , which is too fast for precise damping measurements ($\kappa > 4$) at near vacuum condition. However, the investigation aims at the comparison of displacement measurements rather than damping extraction and the sweep rate was selected to limit the measurement time. Furthermore, this will show how accurately the BTT system measures transient responses of the blisk and it is assumed that the detected

⁵mean of all blades

⁶with respect to the direction of the blisk rotation

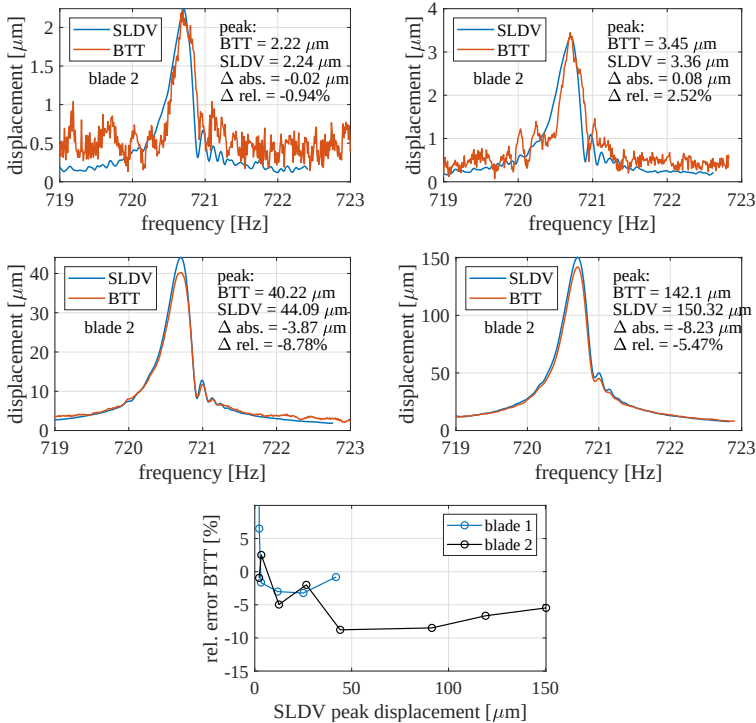


Figure 8.5: Sensitivity analysis of the BTT system at different vibration amplitudes of B1ND2BTW, $n = 6000$ rpm.

differences in the displacement measurements apply to the subsequent damping extraction as well. The first four plots in fig. 8.5 show the response of the same blade to different excitation levels. Due to the fast sweep rate, beats are visible. In the first plot, it can be seen that the noise level of the BTT measurements is approx. $1\mu\text{m}$, whereas the SLDV clearly resolves the beats even at such low amplitudes. Nevertheless, the BTT system is able to accurately capture displacement peaks of $2\mu\text{m}$. For higher displacement

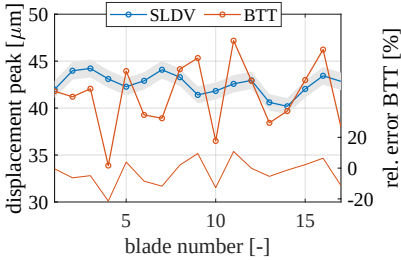


Figure 8.6: Consecutive peak measurement of B1ND2BTW at $n = 6000\text{rpm}$ with excitation frequency sweeps.

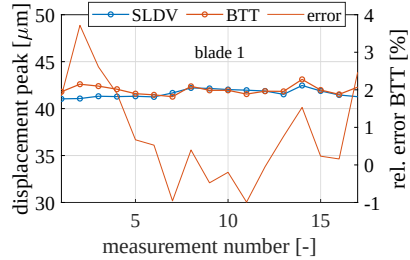


Figure 8.7: Reproduction measurements of B1ND2BTW at $n = 6000\text{rpm}$

amplitudes, the influence of the BTT noise becomes insignificant and the agreement of the frequency response between the BTT and SLDV system is sufficient. The last plot in fig. 8.5 summarizes the differences in peak amplitude between both systems and it can be concluded that the BTT system underestimates the maximum displacement by up to 8% for blade 2 and the investigated mode. The very small differences for the low peak amplitudes should be considered with caution as noise has a significant influence. Data for blade 1 is also included in fig. 8.5 for lower displacement amplitudes since blade 1 is always measured by the reference laser. Unfortunately, clipping occurred in the DAQ system of the reference laser for high displacement amplitudes in this investigation limiting the available data.

Further comparisons are provided in the appendix D for B1ND0, B1ND2BTW and B1ND3BTW, see figs. D.1 to D.3, showing a maximum difference of < 20%.

While the first sets of measurements focused on blade 1 and 2, the differences in peak amplitudes for all blades are investigated in the following for the same mode as investigated above. For this purpose, 17 excitation sweeps are performed while a different blade is measured using the SLDV every time.⁷

⁷It is not possible to measure all blades simultaneously with the derotated SLDV system during an excitation sweep, as only one point can be measured at the same time, whereas the BTT system is able to track all 17 blades at once.

The results of the measurements are summarized in fig. 8.6 as peak displacement amplitudes of both systems for all blades. The grey shadow around the SLDV data represents the SLDV uncertainty for this mode as described earlier. The SLDV measurements expose a variation in the blade individual peak amplitude in the range of approx $5\mu\text{m}$, which shows that there is still some mistuning present for B1ND2BTW, assuming a constant excitation amplitude for all 17 measurements. However, this can be considered of minor influence for the aerodynamic damping measurements as the phases of the blades show a tuned behavior, see fig. 8.4. The comparison with the BTT measurements reveals that the BTT system overestimates the blade individual variation with a mean relative difference to SLDV of 7.0 % and a maximum difference of -21.4% for blade 4. The mean peak amplitude of all blades differs by 2.9 % between both systems.

Since the reference laser always acquires data for blade 1, the 17 measurements can be evaluated for this blade to obtain information on the reproducibility of the results of BTT and SLDV. In fig. 8.7, the peak displacement amplitude is depicted for all 17 measurements. First, the figure shows that the electromagnetic excitation system reliably produces similar displacement amplitudes if the coil current is maintained. Second, it can be concluded that the differences between BTT and SLDV do not change significantly for multiple measurements within a range of approx. 5 %.

For the last part of the validation of the BTT system, free decay measurements are performed, where B1ND2BTW is excited to a certain displacement amplitude followed by an abrupt stop of the excitation. The history of the displacement of blade 2 is shown in fig. 8.8a with a sufficient agreement in the shape of the curves between BTT and SLDV. In order to quantify the differences between the results of both measurement systems, damping is determined by fitting an exponential decay function to the data, c.f. section 2.2.1. As depicted in Figure 8.8b, the difference in damping between BTT and SLDV is negligible. The free decay measurements were repeated for different initial amplitudes and a maximum difference of 5.25 % was determined, see fig. D.4 in appendix D.

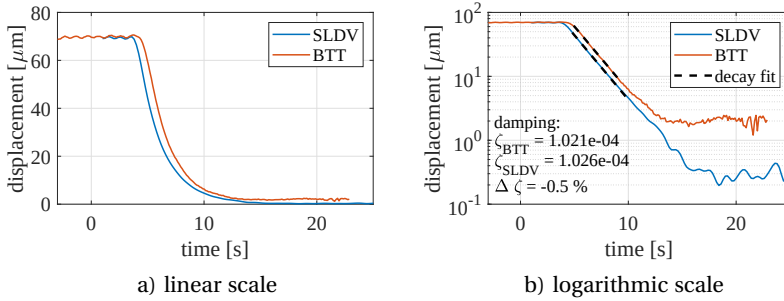


Figure 8.8: Free decay measurement of B1ND2BTW at $n = 6000$ rpm with damping analysis, blade 2. BTT data shifted by 1 s for better visibility.

8.1 Conclusions

The measurements show that the electromagnetic excitation system is able to excite normal modes in an isolated manner and with high reproducibility. Furthermore, the unique measurement setup enabled the validation of the blade tip timing system at different rotational speeds and for various modes by means of constant excitation, frequency sweeps and free decay measurements. The maximum error observed for the BTT system is -21.4% but significantly less for most of the modes and blades. The noise level of the BTT signal is approximately $1\mu\text{m}$ for the present experiments and blade displacements $\geq 2\mu\text{m}$ are measured reliably.

In order to further minimize the error of BTT system and as a result of the validation study, the blade trigger setup is changed for the following aerodynamic damping measurements in the present work. Every time a blade passes a BTT laser, a signal peak is created due to the laser light reflected by the blade tip. The width of this peak depends on the width of the blade tip and the rotational speed and the peak can be triggered at the rising or the falling edge in order to calculate the blade arrival time. While the measurements presented above were performed with the BTT system triggering at the falling edge due experience of measurements at other test rigs, the trigger

is changed to the rising edge of the peak for all following measurements. The result of the change in trigger position is presented in fig. 8.9. With the

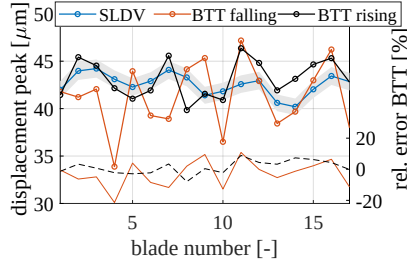


Figure 8.9: Consecutive peak measurement of BIND2BTW at $n=6000$ rpm by means of excitation frequency sweeps. Comparison of BTT trigger at rising and falling signal edge.

trigger at the rising edge, the amplitude distribution measured by the BTT system shows a more similar distribution to the SLDV result. In addition, the mean relative difference in peak amplitude reduces to 3.6 % with a maximum difference of 8.9 % for blade 11. The reason for the differences can partially be attributed to the high (but slightly different) intensity of the lasers in combination with clipping effects in the BTT analog DAQ system, which causes higher errors for a falling signal edge trigger setup. Small geometrical differences on the blade tips due to inevitable tolerances in the manufacturing process can also induce spurious reflections of the laser light leading to uncertainties in the BTT measurements. Due to the limited availability of the derotator system, which was rented for a few days, the validation study was not repeated with the changed trigger setup. However, it is assumed that the reduced errors for the rising edge trigger also apply for other modes and rotational speeds.

The comparison in fig. 8.9 also shows that the BTT system still overestimates the blade-to-blade variation of the peak displacement amplitude. Since these variations are often used to estimate mistuning, the uncertainty should be considered when using BTT results.

9 Steady Flow Field Measurements and CFD Validation

In order to validate the steady-state CFD simulations of the flow-field, which is the basis for subsequent aerodynamic damping simulations, five speed lines at constant rotational speed are measured, covering a large part of the compressor map. In addition, detailed flow field measurements using pneumatic pressure probes upstream and downstream of the rotor are conducted and compared to CFD results in the following along with comparisons of the casing pressure based on phase-locked data from miniature fast-response pressure transducers in the compressor casing. Only stable operating points are investigated. The speed lines are measured at constant inlet conditions regarding total pressure and total temperature. Therefore, the pressure and temperature is constantly controlled in the test facility. While the total temperature is always kept constant at $T_{t,in} = 294\text{ K}$, three different total pressure levels at the inlet are defined: $p_{t,in} = 250\text{ mbar}$, 500 mbar , 970 mbar .

Figure 9.1 shows the measured compressor maps along with the CFD results. The speed lines are depicted in terms of static pressure ratio at the casing, c.f. eq. (5.2). The mass flow rate is corrected to reference conditions. For $p_{t,in} = 970\text{ mbar}$, see fig. 9.1, the measurements are repeated on a second day, showing the excellent reproducibility of the measurement results. In general, the agreement of CFD and experimental results is very good for all five rotational speeds and in particular for $p_{t,in} = 970\text{ mbar}$ and $p_{t,in} = 500\text{ mbar}$. At the lowest inlet pressure level of $p_{t,in} = 250\text{ mbar}$, see fig. 9.1c, a slight clockwise rotation of the measured speed lines is visible, leading to an over-prediction of the static pressure ratio at high mass flow rates with CFD and vice versa for operating points at low mass flow rates of each speed line. Detailed flow field measurements for all three inlet conditions are presented in figs. 9.2 and 9.3 for the operating points A, B, C, NS (near stall), DP (design point) and NC (near choke) as exemplary marked in fig. 9.1a. The Reynolds numbers for these operating points and the different inlet conditions vary between 0.76×10^5 and 5.20×10^5 , see table E.1 in appendix E. The experimental data is compared to circumferentially averaged CFD results.

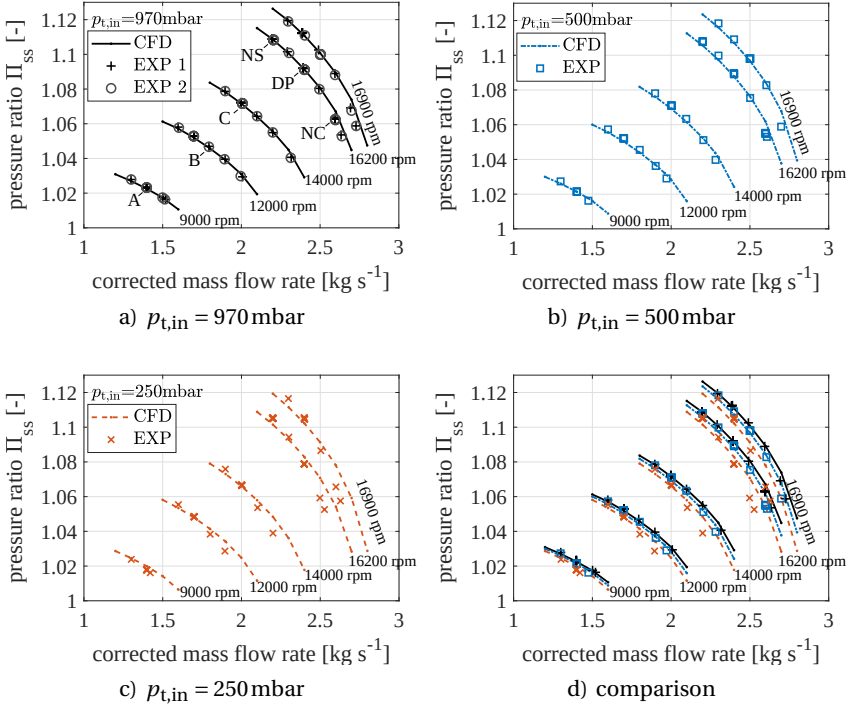


Figure 9.1: Measured and simulated compressor maps for different inlet pressure levels. Mass flow rate corrected to $p_{\text{ref}} = 970 \text{ mbar}$ and $T_{\text{ref}} = 294.15 \text{ K}$.

The flow field for different rotational speeds is depicted in fig. 9.2. As shown in fig. 9.2c, the flow field upstream of the compressor in E1 is completely axial without any swirl for all rotational speeds and inlet conditions. The Mach number distribution in E1, see fig. 9.2b, shows a very good agreement at all operating points down to a relative blade height (span) of 0.4, where the measured Mach number tends to be higher towards the hub compared to CFD, which predicts a symmetric Mach number profile. The deviations between the different pressure levels are insignificant, both in CFD as well

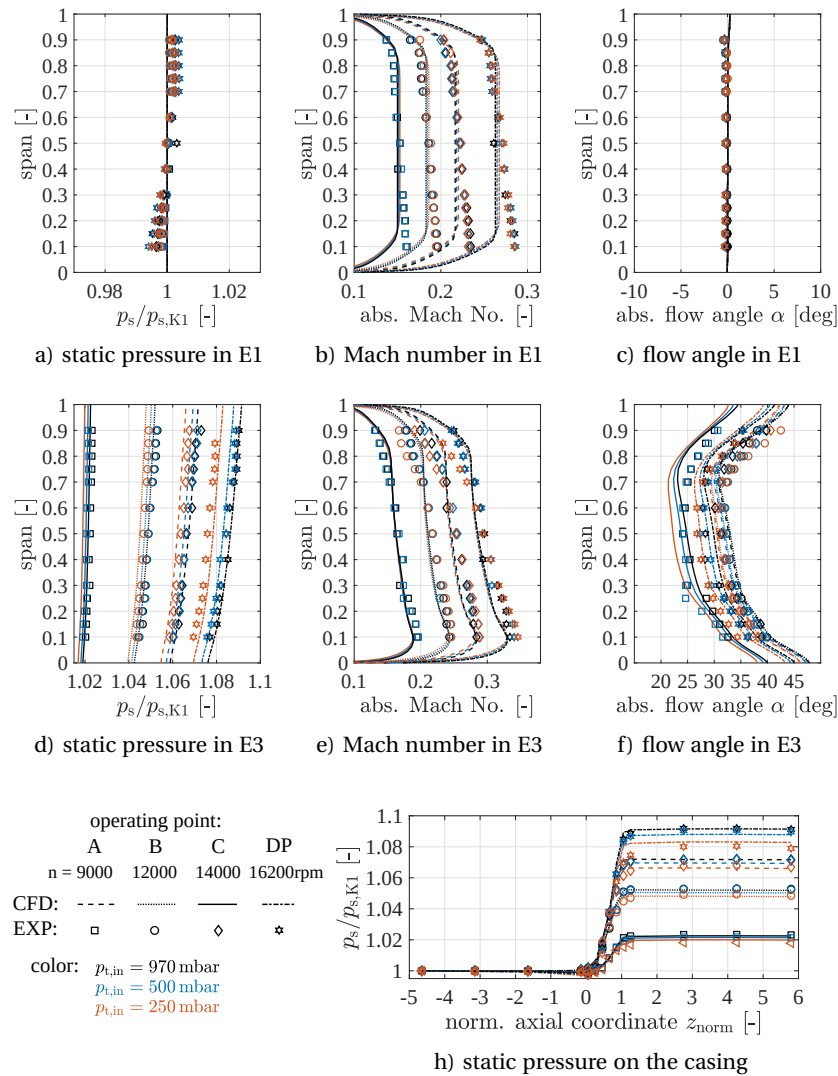


Figure 9.2: Comparison of flow field measurements and CFD simulations at different rotational speeds. Comparison in a plane upstream (E1) and downstream (E3) of the rotor, for four operating points (A, B, C, DP)

as in the measured data. The measured pressure profile is slightly rotated compared to CFD, see fig. 9.2a. The Mach number profile downstream of the rotor in E3, see fig. 9.2e, is captured sufficiently well by the CFD simulations for all rotational speeds. Again, there are no significant differences between the different pressure levels, except for the region between $\text{span}=0.7$ and 1, where the tip vortex in combination with the boundary layer at the casing causes a deficit in Mach number. This deficit is more pronounced for the cases with the lowest Reynolds numbers with $p_{t,\text{in}} = 250 \text{ mbar}$ in the measurements, which is not reproduced by the CFD simulations. The pressure as well as the flow angle distribution in E3 is captured well by CFD with only minor differences. There is also a very good agreement between measurements and CFD for the pressure distribution on the casing in axial direction, as depicted in fig. 9.2h. This figure also shows, for CFD as well for the measurements, the reduced pressure increase at operating points with lower inlet pressure levels and thus lower Reynolds numbers, which can be attributed to the Reynolds (number) effect¹. This effect is not only visible in the casing pressure, but also for the complete pressure profiles in E3 as it can be seen in fig. 9.2d.

A second set of measurements is presented in fig. 9.3 at three different operating points at design rotational speed. Again, the flow field in E1 is completely axial, see fig. 9.3c, with an increased Mach number distribution towards the hub and a slight decrease towards the casing compared to CFD, see fig. 9.3b. This tendency is maintained through the rotor and thus also visible in E3, see fig. 9.3e. Nevertheless, fig. 9.3f shows that the agreement between measurements and CFD in terms of flow angles downstream of the rotor is very good.

A more detailed comparison of measurement and CFD results at the design point is presented in fig. 9.4, where the casing pressure is depicted for four different axial positions.² In general, the CFD simulations agree sufficiently well with the mean pressure distribution of the measurements with minor differences, mostly within the measurement range. Furthermore, it can be seen how the spread of measurement data increases towards the trail-

¹ relative increase of the boundary layers for cases with lower Reynolds numbers causing a decrease in pressure ratio and efficiency [32, 45]

² normalized axial coordinate, leading edge at $z_{\text{norm}} = 0$, trailing edge at $z_{\text{norm}} = 1$

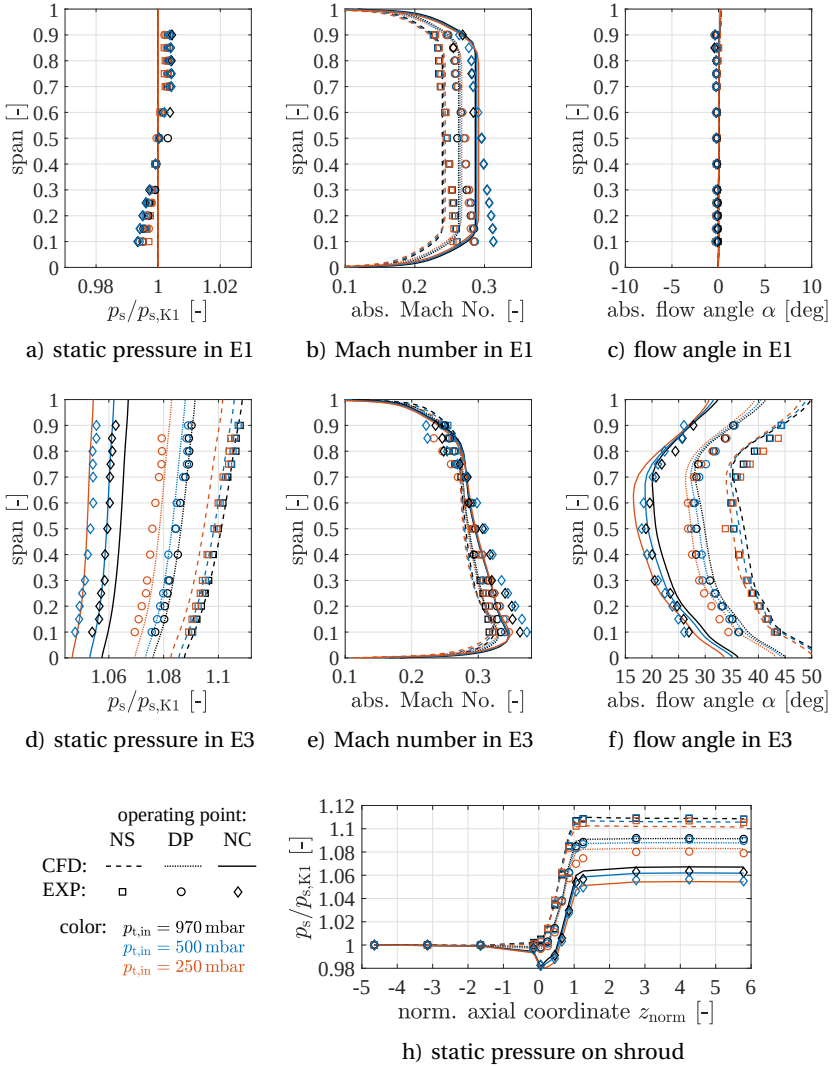


Figure 9.3: Comparison of flow field measurements and CFD simulations in a plane upstream (E1) and downstream (E3) of the rotor, for three operating points at $n = 16200$ rpm (NS=near stall, DP=design point, NC=near choke)

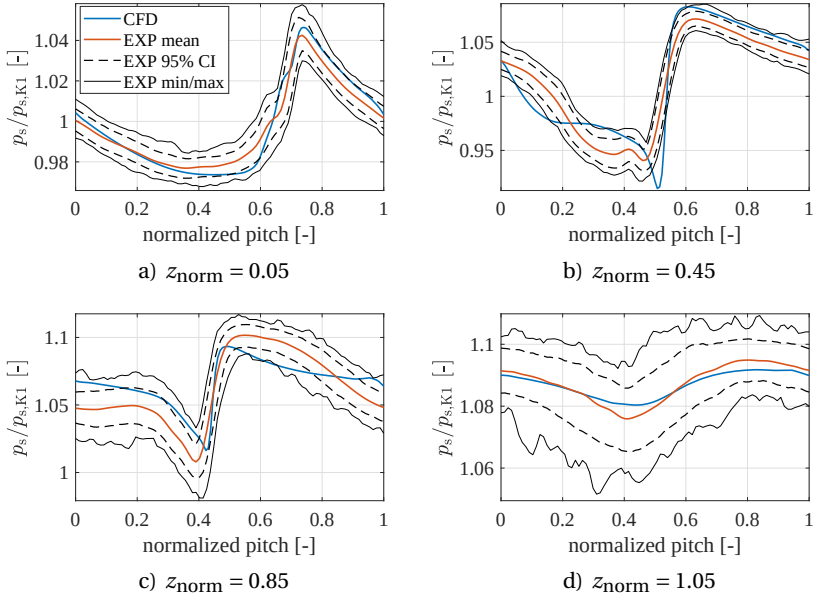


Figure 9.4: Comparison of CFD and phase-locked averaged pressure distribution at the casing, operating point DP, $p_{t,in} = 500 \text{ mbar}$

ing edge, which can be attributed to the increasing turbulence of the flow field introduced by the tip vortex.

At the operating points NS, DP and NC, the mean of the phase-locked pressure data at eight axial positions in the casing is used to construct contour plots of the casing pressure distribution. In addition, fig. 9.5 shows the corresponding CFD results. The position of the blade is indicated with a black profile and CFD and measurement results share the same pressure scale. First, the general agreement of the flow field between measurements and CFD is visible at all three operating points. Second, it becomes evident how the point where the tip vortex detaches from the blade suction side moves from the leading edge towards the trailing edge with an increase in mass flow rate (from NS to NC). Comparing the exact position of the tip vortex detachment

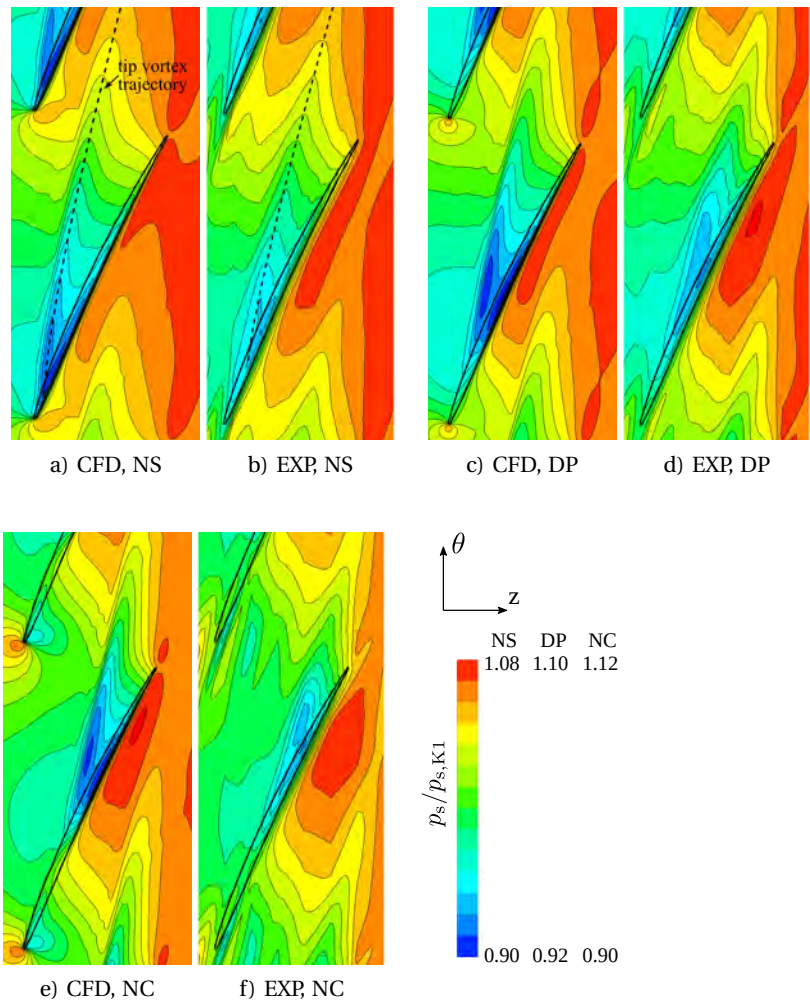


Figure 9.5: Comparison of casing pressure distributions CFD vs. measurements at different operating points, $p_{t,in} = 500 \text{ mbar}$

between measurements and CFD, it becomes apparent that the tip vortex detaches slightly further downstream in the measurements for all three OPs. Besides, the tip vortex trajectory is not fully captured by CFD as it is slightly less bended in axial direction. The tip vortex regions of very low pressure are more pronounced in the CFD, which is assumed to be due to the time averaging of the measurement data as well as uncertainties in the steady-state RANS approach.

Figure 9.6 shows similar pressure contours for the three different inlet pressure levels at the design point. By normalizing the static pressure with the corresponding pressure value at K1, the same scale can be used despite the different pressure levels. It appears that the general pressure distribution is maintained for all three cases, in CFD as well as for the measurements. Nevertheless, the pressure increase is reduced for the case with the lowest inlet pressure level, see fig. 9.6c and fig. 9.6g, respectively. Besides the differences between CFD and measurements discussed above, which mainly concern the tip vortex, no additional differences arise by changing the inlet pressure level.

9.1 Conclusions

A thorough measurement campaign was presented and compared to CFD and it can be concluded that the steady-state CFD model is considered validated, as it captures the integral pressure increase as well as the local flow features. Nevertheless, there are minor differences between measurements and CFD, mainly the increased velocities in the hub region at all operating points and inlet conditions. As this increase is not visible in the total pressure distribution³ it is assumed that the compressor geometry upstream of the axial test section (inlet bulb) causes an acceleration of the flow in the hub region leading to the asymmetry in the Mach number profile. However, the CFD models investigated in section 6.2.1, which include the inlet bulb⁴, do also not reproduce the measured Mach number profile in E1, see fig. 6.4. The study above also shows that the Reynolds number effect is clearly visi-

³not shown in the present work

⁴CFD model V3 and V4, see fig. 6.3

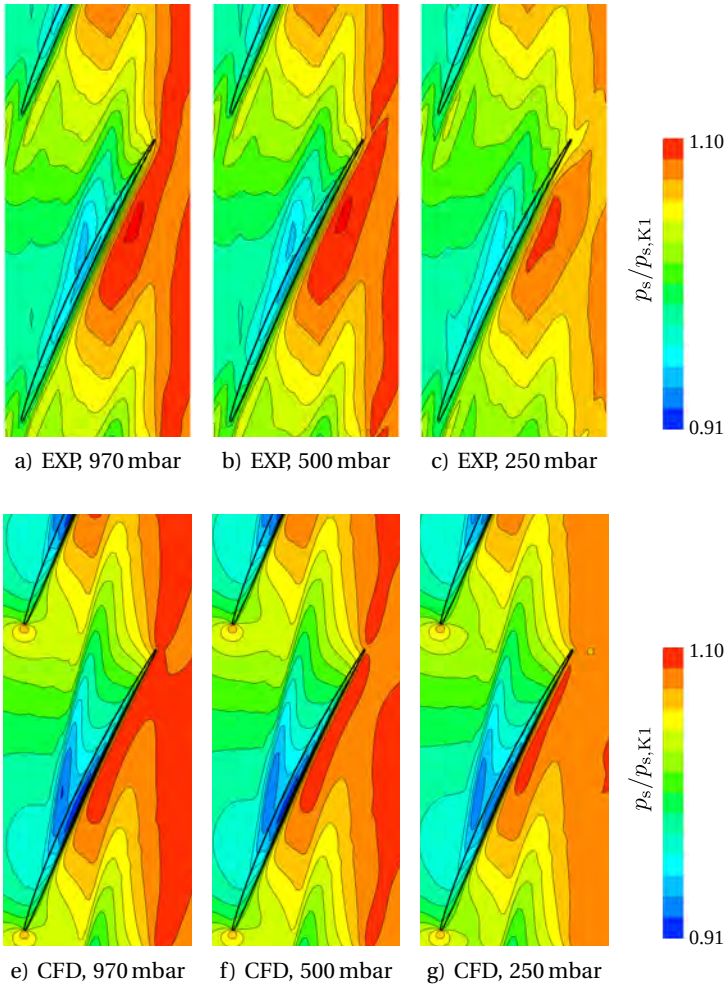


Figure 9.6: Comparison of casing pressure distributions CFD vs. measurements for different inlet conditions ($p_{t,in}$), Design point (DP)

ble, but the flow conditions are very similar for all three inlet pressure levels nonetheless. This leads to the assumption that the effect of different pressure levels on the investigation of aerodynamic damping is mainly driven by the influence of fluid density and changes in the flow field due to different Reynolds numbers will play a minor role.

10 Preliminary Studies for Damping Measurements

In the following chapter, different experimental parameters, such as the sweep rate and the excitation amplitude, are investigated for different modes and operating conditions in order to quantify their influence on the measured damping values and estimate uncertainties. In addition, ring down and sweep excitation experiments are compared. Based on these investigations, the material damping is determined in the last part of this chapter at vacuum conditions and for different rotational speeds. This is done for several modes of the first bending mode family, which are further investigated in terms of aerodynamic damping in chapter 11.

10.1 Sweep Rate

As discussed in section 5.5.3, the right choice of the excitation frequency sweep rate is important for the correct measurement of the stationary blisk response and hence for correct damping measurements.

The influence of the sweep rate is investigated for B1ND2BTW at $n = 9000$ rpm and at vacuum conditions as well as at $p_{t,in} = 500$ mbar. The sweep rate is varied and the results in terms of overall damping are depicted in figs. 10.1a and 10.1b in grey. The estimated reduced sweep rates κ_{est} are also included. Furthermore, the peak blade displacement is presented in the figures on the second axis. It can be seen for both operating conditions that the overall damping ratio as well as the peak blade displacement converges with a decreasing reduced sweep rate and a stationary state is reached at approximately $\kappa_{est} = 1$, which experimentally confirms the results of Hackenberg & Hartung [89, 90]. Moreover, damping is generally overestimated for $\kappa_{est} > 1$, which is also in agreement to the findings in [89, 90]. For these cases, the correction methodology of Hackenberg & Hartung is tested based on the graphs provided in [89]. As shown in blue in figs. 10.1a and 10.1b, the

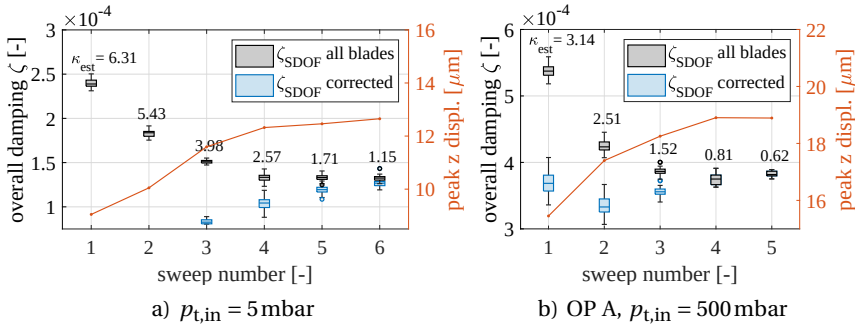


Figure 10.1: Investigation of different excitation frequency sweep rates for B1ND2BTW at $n = 9000$ rpm

corrected damping ratios generally underestimate the converged damping values. A possible reason for the errors of the correction method could be the noise of the measurement data, but no further investigation was conducted since the sweep rate can be adjusted individually in the present experimental setup to meet the requirements. Nevertheless, some measurements at vacuum conditions, see section 10.4, are conducted with $2.2 \geq \kappa_{\text{est}} > 1$ due to the very low damping ratios, which result in very low frequency sweep rates¹.

10.2 Excitation Level and Reproducibility

For the aerodynamic damping CFD simulations, a linear behavior of the tran-

¹A frequency sweep of 2 Hz was performed in 180 s for example for sweep number 6 in fig. 10.1a, resulting in a reduced sweep rate of $\kappa_{\text{est}} = 1.15$ for an overall damping ratio of 1.326×10^{-4} . Other modes show approximately half the structural and material damping causing a significantly higher measurement time as the required frequency sweep rate depends on the square of the damping ratio, see eq. (5.9). However, the measurement time is limited by thermal aspects (blisk and magnet) and a reduction in frequency range is sometimes difficult for practical reasons as the exact peak position varies with temperature and rotational speed

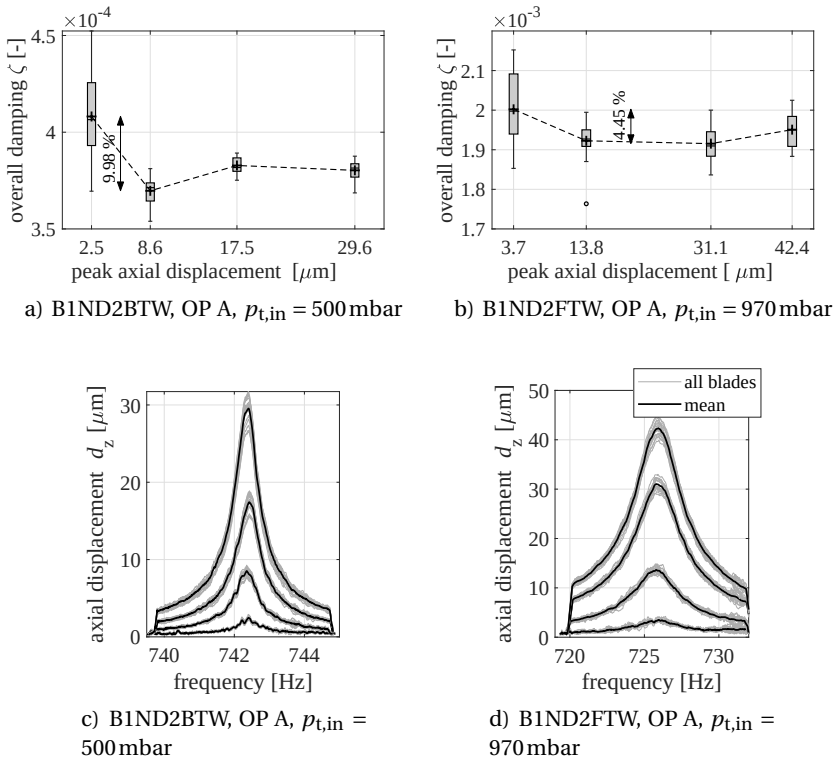


Figure 10.2: Influence of displacement amplitude on damping measurement

sient flow field is assumed and confirmed in section 6.2.3, where the amplitudes of the transient pressure field scale linearly with the blade displacement amplitude, resulting in a constant aerodynamic damping ratio, independent of the prescribed displacement amplitude. This linear behavior is also investigated experimentally by exciting modes to different displacement amplitudes while maintaining the operating conditions. Results for B1ND2BTW and B1ND2FTW are presented in figs. 10.2a and 10.2b, where the measured overall damping is depicted for different maximum blade dis-

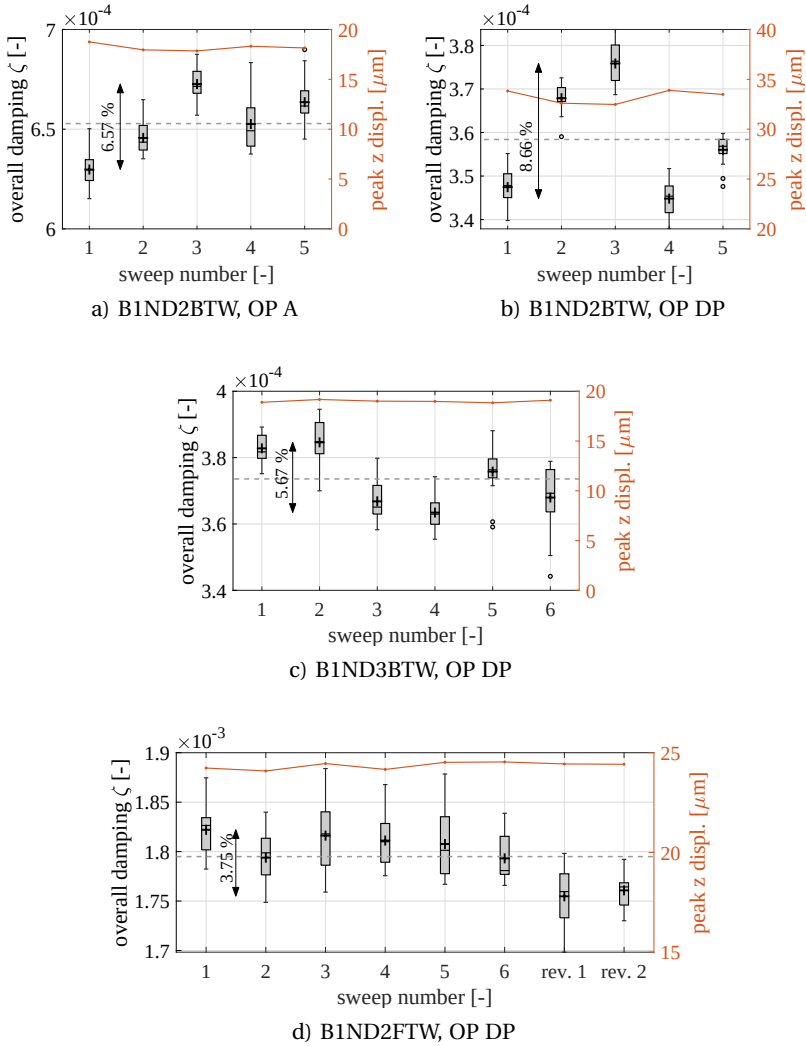


Figure 10.3: Reproducibility of damping measurements at $p_{t,in} = 500\text{mbar}$

placement amplitudes (mean amplitudes of all blades). The corresponding frequency response curves are depicted in figs. 10.2c and 10.2d, respectively. Both modes are excited to very low displacement peaks with $2.5\mu\text{m}$ and $3.7\mu\text{m}$, respectively, close to the minimum level detectable by the BTT system of approximately $2\mu\text{m}$, as confirmed by the BTT sensitivity analysis, see fig. 8.5. Thereafter, the peak displacement is increased step-by-step. The results show that for very low excitation amplitudes, the spread in overall damping increases, which can be attributed to the increasing influence of noise in the BTT signal compromising the SDOF fitting and thus the determination of damping. However, considering the mean values of each measurement, the results show that the mean of the overall damping ratio extracted from the BTT measurements only varies by 9.98 % and 4.45 %, respectively, even though the peak displacement amplitude is varied by a factor of approximately 12 for both modes. This range of variation in damping for different peak displacement amplitudes is confirmed for several other modes and operating conditions, see fig. F.1 in the appendix F.

In order to show the reproducibility of the damping measurements, different modes are excited several times using a constant excitation force and maintaining the operating conditions. The determined overall damping ratios of this investigation are depicted in figs. 10.3a to 10.3d. The dashed line represents the mean of all measurements and the length of the double arrow represents the maximum deviation of the blisk mean damping of one measurement. In addition, the peak displacement amplitude is included for all measurements on the second axis. First, the results in terms of peak displacement amplitude show the ability of the excitation system to accurately reproduce the peak displacement amplitude for all investigated modes, if the excitation force is kept constant. Second, it can be seen that the maximum deviation in terms of mean damping between the consecutive measurements is between 3.75 % and 8.66 % for B1ND2FTW and B1ND2BTW, respectively. Figure 10.3c presents the reproducibility of B1ND3BTW by means of six consecutive sweeps showing a maximum difference of 5.67 %. For B1ND2FTW, two excitation frequency sweeps are reversed in their direction, see *rev.1* and *rev.2* in fig. 10.3d. While all sweep measurements are performed using a positive frequency gradient, the excitation is started at a frequency higher than the resonance frequency and then continuously lowered for these two mea-

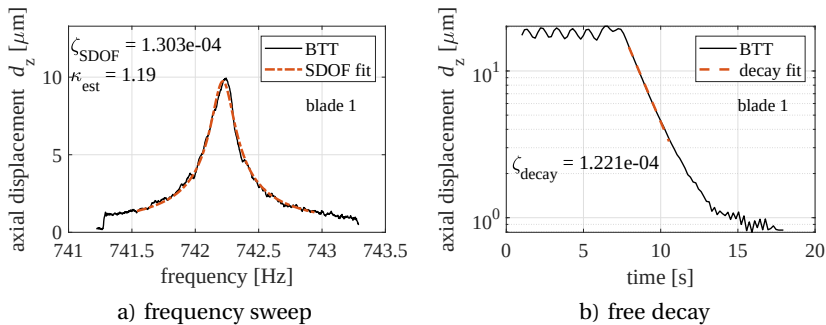


Figure 10.4: Excitation frequency sweep and free decay: Comparison of both damping measurement methods for the same mode. B1ND2BTW, $n = 9000$ rpm, vacuum

surements. The sweep rate is not changed, regardless of the sweep direction ($\kappa_{\text{est}} = 0.12$). It can be seen that the maximum difference between both sweep directions in terms of damping is 3.75 %, which is considered insignificant for the present investigation as the uncertainty in reproducibility for the other modes is in the same range. The results thus also confirm the assumption of a stationary response for $\kappa_{\text{est}} < 1$, where excitations in both sweep directions yield the same frequency response curve and hence the same damping value.

10.3 Free Decay versus Excitation Sweep

The experimental determination of damping can either be realized by means of excitation frequency sweeps or free decay measurements, see 2.4.5. Both methods are used to determine the combined structural and material damping of B1ND2BTW at vacuum conditions and 9000 rpm. The frequency sweep response of blade 1 is depicted in fig. 10.4a and fig. 10.4b shows the displacement curve for the free decay measurement of the same blade. With both methods, the damping can be determined sufficiently well with an accept-

able difference of 6.7%.² The frequency sweep measurement results in a higher damping value, which can be attributed to the slightly too high reduced sweep rate. While these measurements at vacuum show sufficient agreement, the differences increase significantly with higher damping ratios and the accuracy of the free decay measurements worsens dramatically. Several free decay measurements were performed at $p_{t,in} = 500$ mbar and $p_{t,in} = 970$ mbar for different modes and the differences to frequency sweep measurements are $> 100\%$. This is due to the fact that the decay fit requires a sufficiently long observation time of the blade displacement to accurately determine damping. For aerodynamic damping measurements, where the overall damping is at least three times, often an order of magnitude higher than the damping in vacuum, the free decay often occurs within one second. Since the temporal resolution of the post-processed BTT blade displacement data is between 94.8 and 170.7 ms for the present investigation³, often not enough data points are available to accurately determine damping. This disadvantage could be compensated by increasing the initial displacement amplitude for modes with higher damping to increase the decay time. However, higher damping necessitates higher excitation forces, which are limited by the excitation system. As a consequence, frequency sweep excitation is preferred for aerodynamic damping measurements as lower excitation forces are required to accurately determine damping in the present work.

10.4 Material Damping of Mode Family B1

The determination of the combined structural and material damping is essential for aerodynamic damping measurements since these contributions have to be subtracted from the overall damping measurements to obtain aerodynamic damping. For this reason, the combined structural and material damping is measured for modes of the first mode family with nodal

²mean difference of all blades is 10.9 % for this comparison

³BTT sensor data of 256 blisk revolutions is used for each circumferential Fourier fit (CFF) in the BTT post-processing software to calculate the blade individual displacement for a certain nodal diameter. For $n = 9000$ rpm, 256 blisk revolutions are equal to 1707 ms. The CFF is performed block-wise for the complete measurement time with an overlap of 90 %, resulting in a temporal resolution of 170.7 ms and 94.8 ms for 9000 rpm and 16200 rpm, respectively.

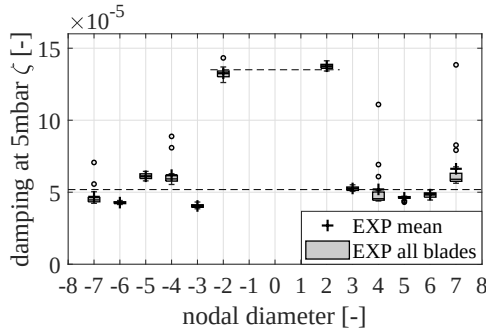


Figure 10.5: Material damping measured at vacuum, $n = 9000$ rpm

diameters of two to seven and at 9000 rpm. Since a blisk is investigated, no structural damping is present and damping at vacuum can be completely attributed to material damping. The measurements are performed at vacuum conditions. The static pressure upstream of the blisk varies between 4.7 and 4.9 mbar ($p_{t,in} = 5$ mbar), depending on the rotational speed, and the influence of the residual air in the test facility is assumed to be insignificant and is hence neglected. The modes are excited both in BTW and FTW direction and the results are summarized in fig. 10.5. B1ND0 is excluded from the investigation because of uncertainties in the measurement as discussed below in section 10.5. B1ND1 was not detected in the preceding experimental modal analysis and hence not further considered as well as B1ND8, which cannot be excited sufficiently using the current excitation system. The results in fig. 10.5 show that there are two levels on which the material damping values concentrate. The mean damping ratio of B1ND2BTW and B1ND2FTW, which is marked with the upper dashed line, is 1.35×10^{-4} and the difference between both modes is insignificant. For modes with three or more nodal diameters, the damping decreases and the mean damping ratio for these modes is calculated as 5.18×10^{-5} , marked with the lower dashed line. There are minor differences visible for the individual modes between BTW and FTW excitation, which are attributed to uncertainties in the measurement. Due to the very low damping values, $\kappa \leq 1$ is not always achieved. Furthermore, comparably low displacement amplitudes are realized for higher nodal diameter

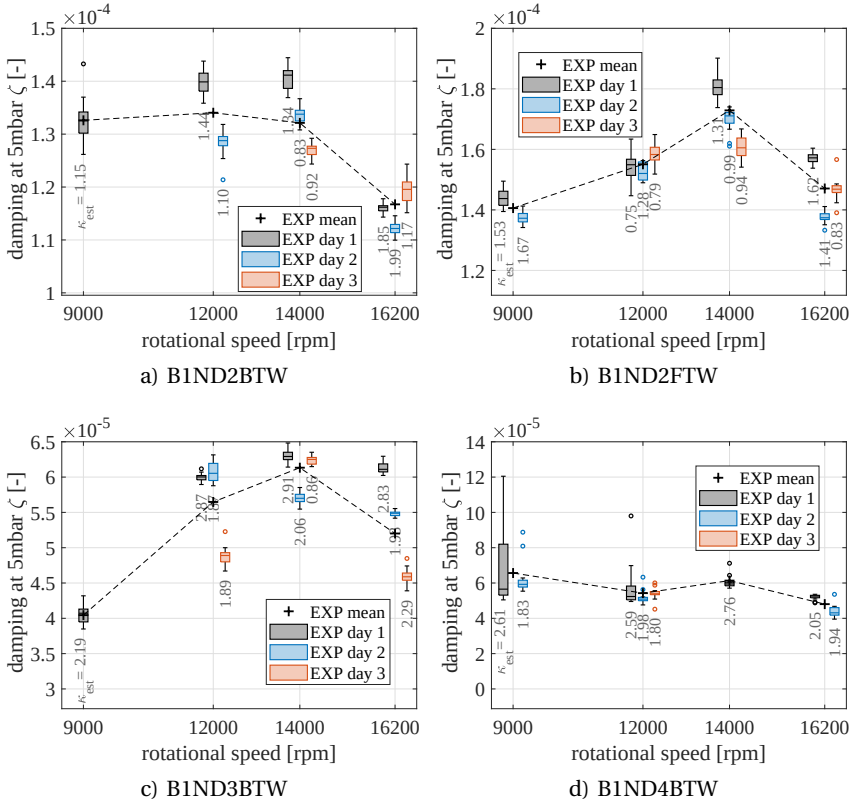


Figure 10.6: Material damping measured at vacuum for five different rotational speeds

modes, see discussion in chapter 12, which increases the influence of noise on the SDOF fit and hence increases measurement uncertainties. Nevertheless, a clear pattern is visible for modes with $n_{nd} \geq 3$ and the lower dashed line represents an estimate of the material damping for these modes in good approximation.

In a second step, the same measurements are performed for four rotational speeds of the compressor map, see also fig. 9.1. Focus is put on B1ND2BTW, B1ND2FTW, B1ND3BTW, B1ND4BTW as for these modes the aerodynamic damping is measured for the complete compressor map and compared to numerical results in chapter 11. No measurements are performed for 16900rpm as this rotational speed is close to the design rotational speed and the same material damping as for the design rotational speed is assumed. The results are presented in figs. 10.6a to 10.6d. Some of the measurements are repeated on a second and third day with slightly varying reduced sweep rates, which are also included in the figures. Figure 10.6a shows for B1ND2BTW that there are variations in the determined damping between the measurement days, in particular for 12000rpm and 14000rpm, which can be explained by differences in the sweep rate as well as uncertainties in the repeatability as discussed above. In general, the mean values of the material damping of B1ND2BTW are almost constant with an approximate value of 1.32×10^{-4} up to 14000rpm. For 16200rpm the mean damping is reduced by approximately 12%.

In case the same double mode is excited in FTW direction, as shown in fig. 10.6b, the material damping is very similar to the BTW excitation for 9000rpm. An increase in damping visible for 12000rpm and 14000rpm, which is repeatable, and a decrease for 16200rpm back to almost the same value as for 9000rpm. The reason for this increase in damping for the two rotational speeds is an increase in measurement noise in the BTT signal. Figure 10.7 presents the measured response curve of one blade for B1ND2FTW at 14000rpm in black. The least-square fitted SDOF response curve is shown in red, along with a response curve in blue, where the damping is manually adjusted to 1.4×10^{-4} .⁴ Due to the significant measurement noise, it appears that both response curves are a fair representation of the measured response. This comparison reveals the limitations in accuracy of the SDOF fit for BTT signals with significant measurement noise and explains the differences in damping for certain rotational speeds mentioned above.

The same explanation is valid for the differences in damping for B1ND3BTW, see fig. 10.6b, where a combination of slightly too high reduced sweep rates and measurement noise leads to deviations in the damping measurement.

⁴the forcing is also adjusted accordingly to obtain the same peak displacement

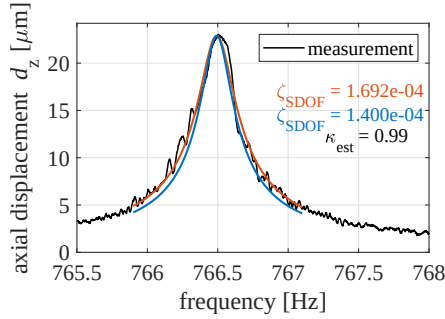


Figure 10.7: Comparison of SDOF response curves, B1ND2FTW, $n = 14000$ rpm, vacuum

Nevertheless, a damping ratio between 4×10^{-5} and 6×10^{-5} for the material damping of B1ND3BTW is verified for different rotational speeds and multiple measurements.

B1ND4BTW shows only minor differences for different rotational speeds and a mean damping ratio of approximately 5×10^{-5} to 6×10^{-5} is observed with high certainty.

10.5 Conclusions and Discussion of B1ND0

The investigations presented in this chapter experimentally confirm that a reduced sweep rate of $\kappa \leq 1$ is required to measure the stationary response of the blisk and hence enable the correct measurement of damping. Furthermore, the reliability of the excitation system and the damping determination method is shown using reproducibility measurements, which reveal an uncertainty of $\leq 10\%$ for multiple modes and operating conditions. Additional measurements with varying displacement amplitudes also show uncertainties in the same range. Given the uncertainty of reproducibility, which also applies for the damping measurements with different amplitudes, it can be concluded that the measured damping is independent of the peak displacements.

A comparison of free decay and excitation frequency sweep measurements shows that the uncertainty for free decay measurements increases significantly with higher damping ratios due to limitations in the temporal resolution of the post-processed BTT data, which prevents reliable damping determination. It is therefore decided to use excitation frequency sweeps for all damping measurements in the present work.

The measurements at vacuum and 9000rpm show that the material damping for various modes of the first bending mode family is very low in general, but a clear distinction can be made between B1ND2, with a damping ratio of 1.35×10^{-4} and modes with a higher number of nodal diameter with an average damping of 5.18×10^{-5} . While there are some variations visible, which are most likely due to uncertainties in the measurement, the results indicate that there is no difference in the material damping between BTW and FTW excitation of the double modes. The observed differences in material damping for different rotational speeds are attributed to uncertainties in the measurement. However, they only have a minor influence on the aerodynamic damping measurements as aerodynamic damping is by far the biggest contributor to the overall damping for the investigated modes in chapter 11.

Furthermore, it can be noted that there are only insignificant fluctuations in the rotational speed during the excitation experiments. Thus, no active adjustment of the excitation frequency to the instantaneous rotational speed is required, as it is for example implemented for turbine experiments in Bessone et al. [18].

Multiple measurements are also conducted for B1ND0 and it is observed that the combined vacuum damping is almost two orders of magnitude higher compared to all other modes of the first mode family. It is assumed that the significant increase in damping originates from the fact that the compressor shaft participates in the vibration of B1ND0 and the additional damping is caused by the shaft bearings. Hence, the measured damping at vacuum comprises of material as well as structural (bearing) damping. However, the measured damping values vary between 1.3×10^{-3} and 3.4×10^{-3} for different rotational speeds with no clear trend. In addition, a dependency of the combined structural and material damping on the bearing temperature is observed, with a vague trend of increasing damping for increasing temperatures. Both effects superimpose and cause a significantly worse repro-

ducibility of the measurements compared to modes with $n_{\text{nd}} \geq 2$. Due to this uncertainty in the determination of the combined structural and material damping, no reliable measurements of the aerodynamic damping are achievable. In particular if one considers the ratio of structural, material and aerodynamic damping for this mode. While aerodynamic damping represents the major part of the overall damping for all other modes, the combined structural and material damping determines the overall damping almost completely for B1ND0. As a consequence, even small differences in the vacuum damping measurements introduce large errors in the subsequent aerodynamic damping measurements. In the present case, the uncertainty of the combined structural and material damping measurements are larger than the predicted aerodynamic damping values with CFD. For this reason, B1ND0 is not investigated experimentally in terms of aerodynamic damping.

11 Results of the First Bending Mode Family

In the following chapter, the results of the aerodynamic damping measurements of the first bending mode family are presented and compared to the results of accompanying CFD simulations. First, the pressure level is varied for multiple modes and at two different operating points. Second, the aerodynamic damping for four different modes is investigated for the complete compressor map. Third, unsteady pressure measurements are investigated for constant as well as frequency sweep excitation. In the last part of the chapter, conclusions are drawn and the maximum amplitudes, which are realized with the current excitation system, are discussed.

The reduced frequencies for the first bending modes differ between 0.96 and 3.66 in the investigated range of rotational speeds, see table 11.1. As discussed in the last chapter, B1ND0 is excluded from aerodynamic damping measurements due to uncertainties in the determination of the material and structural damping. B1ND1 was not detected in the experimental modal analysis and B1ND8 could not be excited sufficiently well. The mode shapes of the first bending mode family can be found in figs. B.2 and B.3 in the appendix B.

Table 11.1: Reduced frequencies k for the first bending mode family and different rotational speeds. B1ND0, B1ND1 and B1ND8 are not investigated in the present work in terms of aerodynamic damping.

B1ND...	(0)	(1)	2	3	4	5	6	7	(8)
9000 rpm	3.97	1.72	1.81	2.98	3.43	3.57	3.63	3.66	3.67
16900 rpm	2.12	0.92	0.96	1.59	1.83	1.91	1.94	1.95	1.96

11.1 Aerodynamic Damping at Different Pressure Levels

The influence of the air density on the aerodynamic damping is investigated at operating point A as well as in the design point by varying the inlet pressure level. The measurements are performed at different total inlet pressure levels resulting in different static pressure levels. It is assumed that the air density scales linearly with the static pressure. The temperature is not varied. The results at OP A are presented in fig. 11.1, where the measured overall damping is depicted at four different pressure levels, including measurements at vacuum. Most damping measurements were conducted several times to compensate for reproducibility uncertainties and the number of excitation sweeps contained in the measurement data is provided below the box plots. The figures also include the aerodynamic damping predicted by the CFD simulations. Since the overall damping is presented, the material damping determined by the vacuum measurements is added to the corresponding CFD data. In addition, a linear fit of the mean measurement data, excluding the vacuum measurement, is included in the figures (solid black line) along with a linear extrapolation to vacuum conditions based on the fit (dashed black line). This allows for the comparison of the material damping of the individual mode based on the extrapolation method with the direct vacuum measurement.

In general, a clear linear trend is visible for all investigated modes, where the overall damping and hence the aerodynamic damping¹ increases linearly with static pressure and thus with air density. This trend is agreement with the general understanding of aerodynamic damping in open literature, see section 2.4.2. The comparison of the damping at vacuum shows that for B1ND2BTW, B1ND2FTW and B1ND3BTW, see figs. 11.1a to 11.1c, the difference in damping ratio between the extrapolation method and the direct measurement is $< 1 \times 10^{-4}$. It becomes apparent, that for these modes, the spread in the damping measurements can be considered insignificant and the linear fit yields reasonable agreement. For B1ND4BTW, B1ND5BTW and B1ND6BTW, see figs. 11.1d to 11.1f, the measurement data spread is more

¹ with the legitimate assumption of a constant contribution of material damping regardless of the pressure level

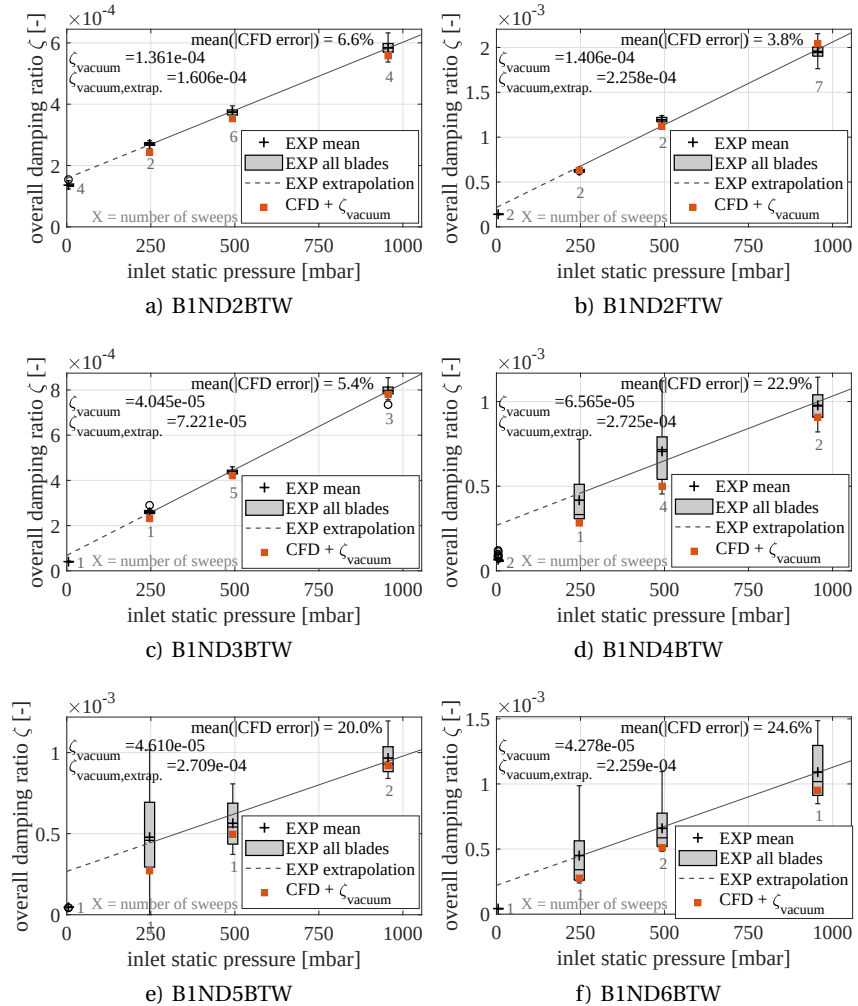


Figure 11.1: Damping measurements vs. CFD for different inlet pressure levels, OP A ($n = 9000\text{rpm}$, $\dot{m}_{\text{corr}} = 1.4\text{kg s}^{-1}$)

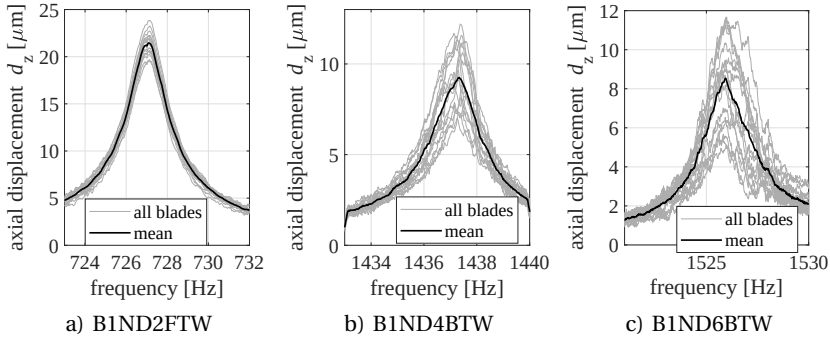


Figure 11.2: Axial blade displacement at BTT1 for sweep excitation of different modes, $p_{t,in} = 500 \text{ mbar}$

pronounced and the accuracy of the extrapolation method worsens. The main reason for the larger spread in extracted aerodynamic damping for modes with a higher number of nodal diameter is the increasing influence of measurement noise in the BTT data, which becomes more dominant for lower displacement amplitudes. An example is provided in fig. 11.2, where the frequency response is depicted for B1ND2FTW, B1ND4BTW and B1ND6BTW. While B1ND2FTW shows an almost perfect SDOF-like resonance curve with negligible disturbances, significant influence of noise is visible for B1ND4BTW and B1ND6BTW, which results in variations of the aerodynamic damping determination by means of a blade-individual SDOF fit. However, the mean blade response still resembles a tuned SDOF response curve sufficiently well. Based on the fact that noise can be reduced almost completely by averaging the response of all blades and that no significant mistuning was detected for B1ND4BTW it is assumed that mainly noise in the BTT data causes the spread in the determined damping. For B1ND6BTW, minor mistuning was identified in the EMA, which increases the spread in

measured damping in combination with noise in the BTT data.²

Considering the CFD results, also a clear linear trend is found. Furthermore, the agreement of the CFD results to the measurement data is very good, in particular for B1ND2BTW, B1ND2FTW and B1ND3BTW with differences < 10%. The mean error of the CFD simulations for all investigated modes in OP A is 13.9%, where the CFD simulations mostly under-predict the measured aerodynamic damping.

The results of the investigation of B1ND2BTW, B1ND2FTW, B1ND3BTW and B1ND4BTW in the design point are presented in fig. 11.3. The same conclusions as for OP A can be drawn since a clear linear trend of the measured overall damping and simulated aerodynamic damping is detected as well. Besides, the agreement of CFD simulations is also very good at this operating point with a mean error of 8.8% in comparison to the measurements.

On the basis of the measurements, the percentage of the material damping on the overall damping at ambient conditions³ can be determined very accurately. For B1ND2BTW at OP A, 23.3% of the overall damping are attributed to material damping (19.7% at DP). For all other modes, the portion of material damping is $\leq 7\%$, with a minimum of 3.3% for B1ND4BTW at the DP. Hence, aerodynamic damping is larger than material damping by a factor of 4.3 to 30.3.

11.2 Aerodynamic Damping for the Complete Compressor Map

In an extensive measurement campaign, the aerodynamic damping of four different modes (B1ND2BTW, B1ND2FTW, B1ND3BTW and B1ND4BTW) is measured for the complete compressor map, see fig. 9.1b, which consists of five speed lines and 23 operating points. The campaign comprises 275 excitation measurements in total, including reproducibility measurements with

²It should be mentioned that the mean value in the box-plot depiction (marked with a cross) represents the mean of the blade-individual damping values and not the damping determined from the mean blade response. Insignificant differences are observed though, if damping is determined directly from the mean response.

³ $p_{t,in} = 970 \text{ mbar}$, $T_{t,in} = 294.15 \text{ K}$

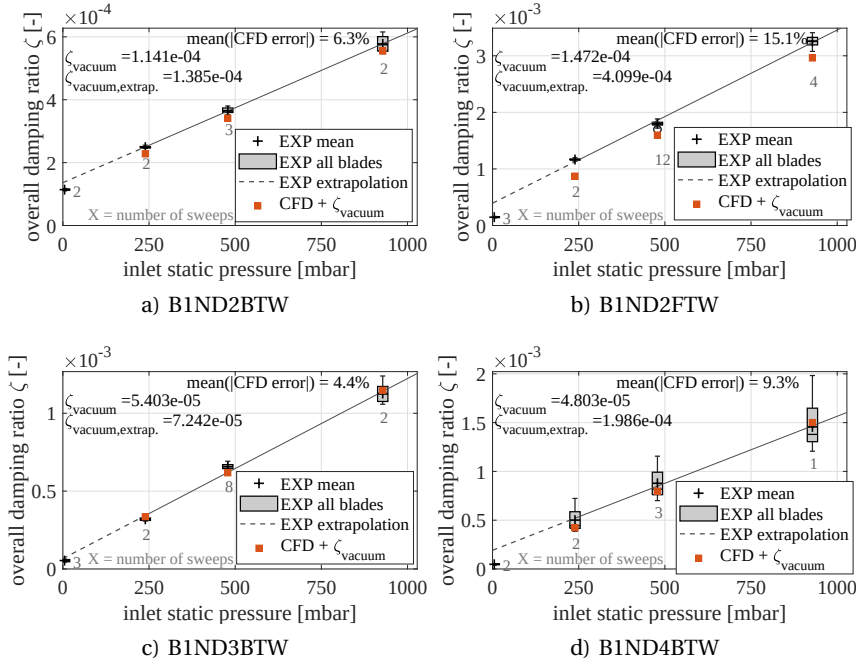


Figure 11.3: Damping measurements vs. CFD at different inlet pressure levels, OP DP ($n = 16200 \text{ rpm}$, $\dot{m}_{\text{corr}} = 2.4 \text{ kg s}^{-1}$)

different displacement amplitudes and measurements on a second day. All measurements are performed at $p_{t,\text{in}} = 500 \text{ mbar}$ in order to reduce the required magnetic excitation force due to the lower overall damping in general in comparison to an inlet condition of $p_{t,\text{in}} = 970 \text{ mbar}$. However, a very linear behavior of the aerodynamic damping is shown in the section above which leads to the assumption that findings in the following can be transferred to other pressure levels with high confidence.

The measurements allow for the determination of general trends of aerodynamic damping with regards to the operating conditions. Furthermore, the results represent a large base of CFD validation data. Consequently, the aero-

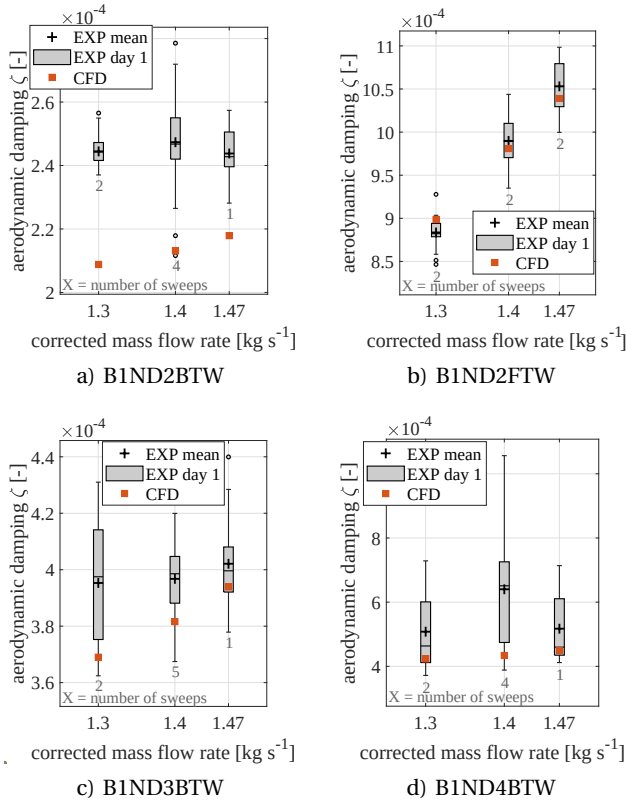


Figure 11.4: Comparison of measurements and CFD in terms of aerodynamic damping at 9000 rpm, $p_{t,\text{in}} = 500$ mbar

dynamic damping predicted by CFD simulations is included in all figures enabling a direct comparison.

Figures 11.4 and 11.5 present the aerodynamic damping for speed lines at 9000 rpm and 16200 rpm, respectively, and for all four modes. Similar figures for all investigated rotational speeds can be found in figs. F.2 to F.5 in the appendix F.

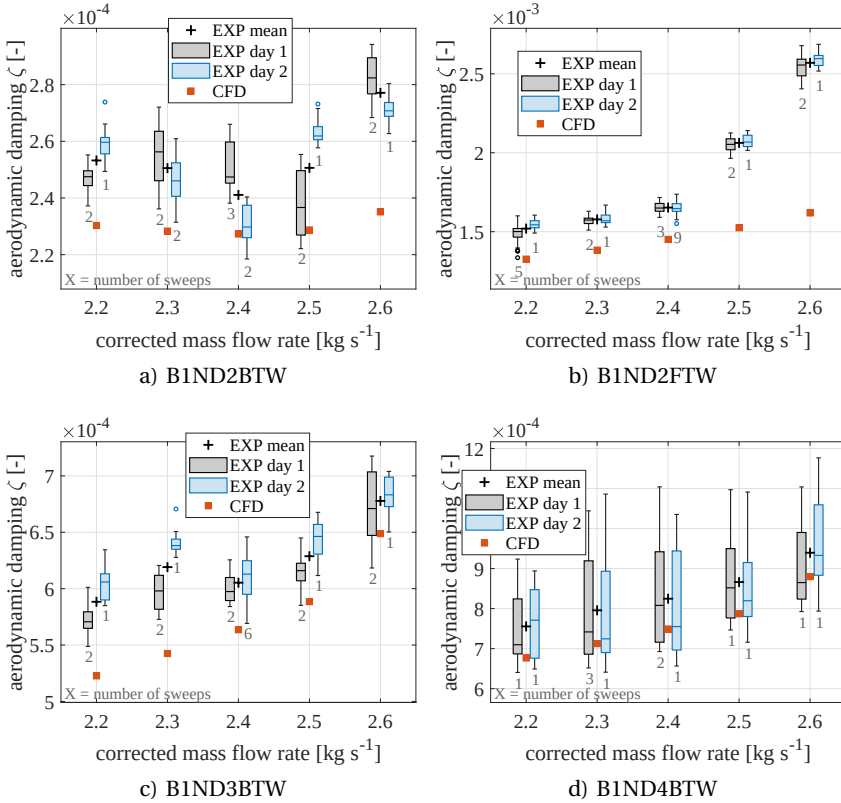


Figure 11.5: Comparison of measurements and CFD in terms of aerodynamic damping at 16200 rpm, $p_{t,in} = 500$ mbar

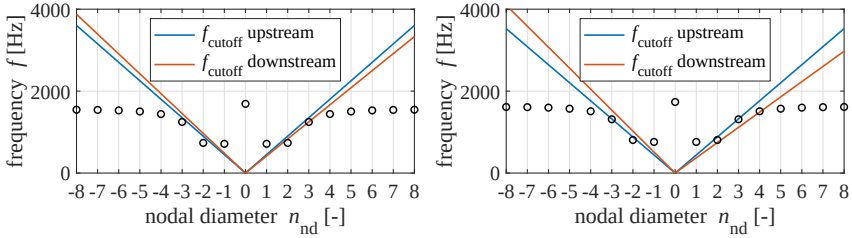
At 9000 rpm, the measured aerodynamic damping for each mode is almost constant for all mass flow rates, except for B1ND2FTW, see fig. 11.4b, where the aerodynamic damping increases for higher mass flow rates. Comparing the absolute values of aerodynamic damping for example at OP A ($\dot{m}_{corr} = 1.4 \text{ kg s}^{-1}$), it can be seen that the aerodynamic damping of B1ND2FTW (9.90×10^{-4}) is almost four times higher compared to

B1ND2BTW (2.47×10^{-4}). The CFD simulations are able to capture the measured aerodynamic damping well, in particular for B1ND2FTW. The maximum difference between CFD and the mean of the measured aerodynamic damping is observed for B1ND4BTW with 16.7 % at 1.3 kgs^{-1} .

At 16200 rpm, the measurements are reproduced on a second measurement day, see fig. 11.5.⁴ The comparison of both measurement days reveal a very good reproducibility of the results, with insignificant differences within the range of general reproducibility uncertainty. The same satisfactory reproducibility is observed for 12000 rpm, 14000 rpm and 16900 rpm, see figs. E2 to E5. It can be observed that at 16200 rpm, the aerodynamic damping continuously increases with increasing mass flow rate for all modes except for B1ND2BTW. This trend is present in the experimental as well as the CFD results. The spread in measured damping ratio is more pronounced for B1ND4BTW, see fig. 11.5d. This can be explained by lower excitation amplitudes in general for this mode, which tend to increase uncertainty and variations in the blade individual measurements due to the influence of measurement noise. The CFD simulations show a sufficient agreement for all BTW modes, with a general trend of under-prediction. The maximum CFD error for the BTW modes is observed for B1ND2BTW at $\dot{m}_{\text{corr}} = 2.6 \text{ kgs}^{-1}$ with 15.2 %. For B1ND2FTW, the difference between CFD and the corresponding mean measurement results is approximately 12 % between 2.2 kgs^{-1} and 2.4 kgs^{-1} , but the error increases significantly for higher mass flow rates up to 37.0 % at 2.4 kgs^{-1} .

It is assumed that an acoustic resonance condition downstream of the rotor is approached for the higher mass flow operating points causing this significant increase in aerodynamic damping. This assumption is based on the distribution of the cut-off frequency as depicted for the design point in fig. 11.6b. The figure presents resonance frequencies with black circles for all modes of the first bending mode family excited in FTW (positive nodal diameter) and BTW (negative nodal diameter) manner. In addition, the mode-individual cut-off frequencies are included according to eq. (2.35), for the ducts upstream and downstream of the blisk. Equation (2.35) is evaluated using averaged values of Mach number from CFD upstream and downstream at 95 % span and thus provides a simplified indicator of the

⁴The measurements were performed at the same corrected mass flow rates on both days. For illustration purposes, the box plots are slightly shifted to the left and right in the figures.



a) OP A ($n = 9000 \text{ rpm}$, $\dot{m}_{\text{corr}} = 1.4 \text{ kg s}^{-1}$) b) OP DP ($n = 16200 \text{ rpm}$, $\dot{m}_{\text{corr}} = 2.4 \text{ kg s}^{-1}$)

Figure 11.6: First bending mode family B1: Cut-on and cut-off conditions for different operating points, upstream and downstream of the rotor blisk. Modes are cut-on if $f > f_{\text{cutoff}}$.

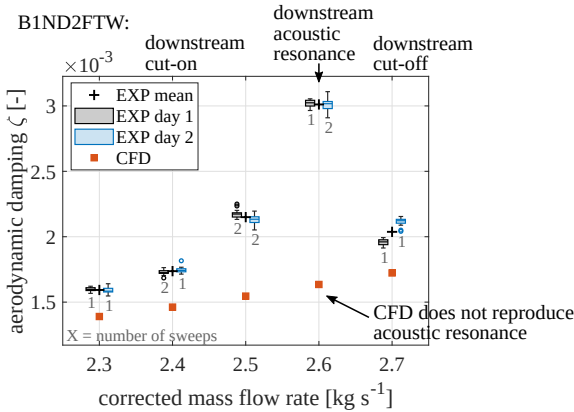


Figure 11.7: Acoustic resonance in the downstream duct influencing aerodynamic damping of B1ND2FTW. $n = 16900 \text{ rpm}$, $p_{t,\text{in}} = 500 \text{ mbar}$

acoustic resonance conditions in the compressor. It can be seen in fig. 11.6b that B1ND2FTW ($n_{nd} = +2$) is cut-off in the upstream duct ($f < f_{\text{cutoff}}$) and cut-on downstream of the blisk. Since the gradient of the f_{cutoff} curve downstream of the rotor increases for positive nodal diameter modes with higher mass flow rates⁵, the cut-off frequency approaches the resonance frequency of B1ND2FTW. As a consequence, the acoustic resonance condition ($f \gtrsim f_{\text{cutoff}}$) is approached leading to a continuous increase in aerodynamic damping towards the resonance. The same mechanism is assumed to be present for the B1ND2FTW at 16900 rpm, see fig. 11.7. For this speed line, not only the influence of the acoustic resonance can be observed but also the abrupt decrease in aerodynamic damping once the cut-off condition is reached at 2.7 kg s^{-1} .

The aerodynamic damping CFD simulations clearly do not capture the influence of the acoustic resonance. Since the applied CFD solver and method⁶ is able to simulate the propagation of pressure waves, it is assumed that the simplified CFD domain of the CFD model V2, see section 6.2.1, is not sufficient to reproduce the acoustic resonance condition for this mode. While the downstream geometry of the compressor consists of a cylindrical duct with cylindrical struts and a diffuser part followed by a large volute, see for example fig. 5.3, a simplified cylindrical duct is used in the CFD model V2. CFD model V3 at least comprises the diffuser part, which might contribute to the correct modeling of the acoustic resonance, but was not used for the final aerodynamic damping simulations due to convergence issues. The results clearly indicate that more detailed CFD domains in the downstream part of the compressor should be investigated in the future to improve the agreement of the CFD simulations in regions of acoustic resonance. However, the applied CFD model V2 is able to estimate aerodynamic damping sufficiently accurate away from acoustic resonances. An overview of the mean and maximum error of the aerodynamic damping CFD simulations at all 23 operating points is presented for each mode individually in table 11.2. The overall mean error of the CFD prediction based on simulations and measurements at $23 \times 4 = 92$ operating points, including the significant errors

⁵the swirl downstream of the rotor decreases for higher mass flow rates (less flow turning) and hence Ma_θ decreases in eq. (2.35)

⁶ANSYS CFX 2020 R1 with the harmonic balance method, see section 6.2

Table 11.2: Aerodynamic damping of the complete compressor map (23 operating points): Maximum and mean difference between CFD and measurements. $\Delta\zeta_{\text{CFD/EXP}} = (\zeta_{\text{aero,CFD}} - \zeta_{\text{aero,EXP}}) / \zeta_{\text{aero,EXP}}$

	B1ND2BTW	B1ND2FTW	B1ND3BTW	B1ND4BTW	all
$\max(\Delta\zeta_{\text{CFD/EXP}})$	18.5 %	45.7 %	13.4 %	32.2 %	45.7 %
$\text{mean}(\Delta\zeta_{\text{CFD/EXP}})$	10.7 %	20.0 %	8.2 %	11.7 %	12.6 %

for some operating points of B1ND2FTW, is 12.6 %. The maximum error is 45.7 % for B1ND2FTW at the acoustic resonance discussed above.

In order to illustrate the distribution of aerodynamic damping at different operating points, figs. 11.8 and 11.9 show contours of the measured (figures on the left-hand side) and numerically predicted (figures on the right-hand side) aerodynamic damping on the compressor map. The same color scale is applied for CFD and experimental data for each mode. The distribution of the measured aerodynamic damping of B1ND2BTW, see fig. 11.8a, shows a clear increase of aerodynamic damping towards lower pressure ratios, as well as a region of minimum damping in the center of the compressor map at 14000 rpm. However, the general variation in aerodynamic damping is very low with a difference between the minimum and maximum value of 12.6 %. While the CFD simulations are able to predict the general level of aerodynamic damping sufficiently well with a slight under-prediction and a mean error of 10.7 %, cf. table 11.2, the qualitative distribution of the measurements not captured completely.

For B1ND2FTW, the measured distribution of aerodynamic damping for higher rotational speeds is mainly dominated by the downstream acoustic resonance, see fig. 11.8c. Since the CFD simulations do not reproduce the acoustic resonance, the agreement to the measurements is not sufficient. Nevertheless, there is a clear tendency of increasing aerodynamic damping in the experimental results, which is qualitatively also visible in the CFD contour. The spread between minimum and maximum aerodynamic damping significantly increases to 240.8 % and 115.4 % for the measurements and the CFD results, respectively.

The general agreement between CFD and measurements is very good for B1ND3BTW with a mean error of 8.2 %, which is also visible in the compari-

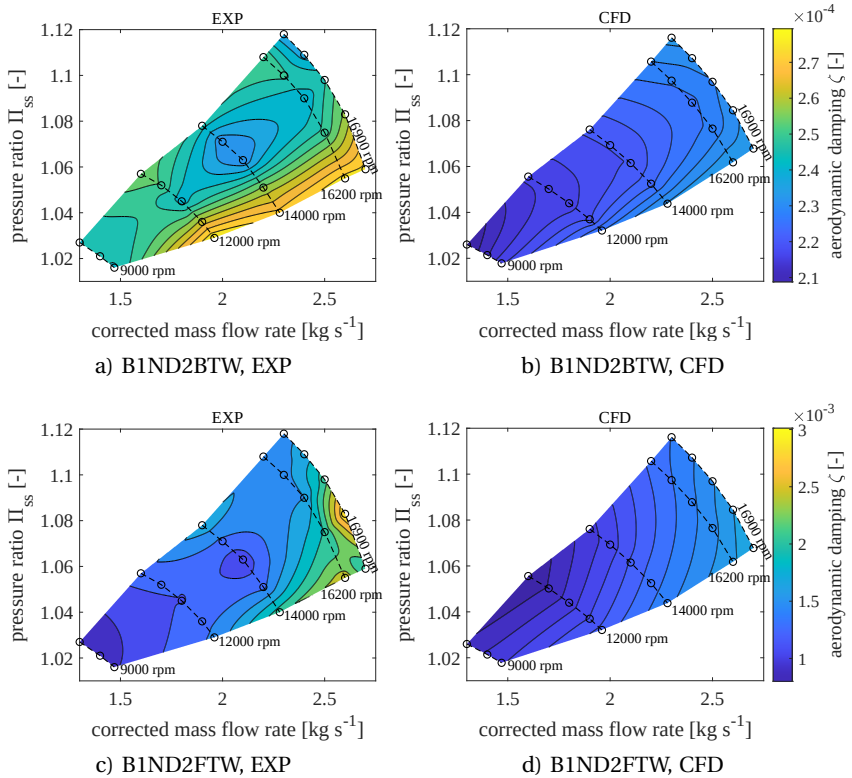


Figure 11.8: Comparison of aerodynamic damping contours on the compressor map for B1ND2BTW and B1ND2FTW. For each mode, the same color scale is used for CFD and experimental data. $p_{t,in} = 500 \text{ mbar}$

son of the experimental and CFD contour, see figs. 11.9a and 11.9b. The CFD results reproduce the clear distribution of increasing aerodynamic damping towards higher mass flow rates with only minor differences in the damping level for 16900 rpm. The measured range in aerodynamic damping over the complete compressor map, which is 93.7 %, is also captured well by CFD with 86.3 %.

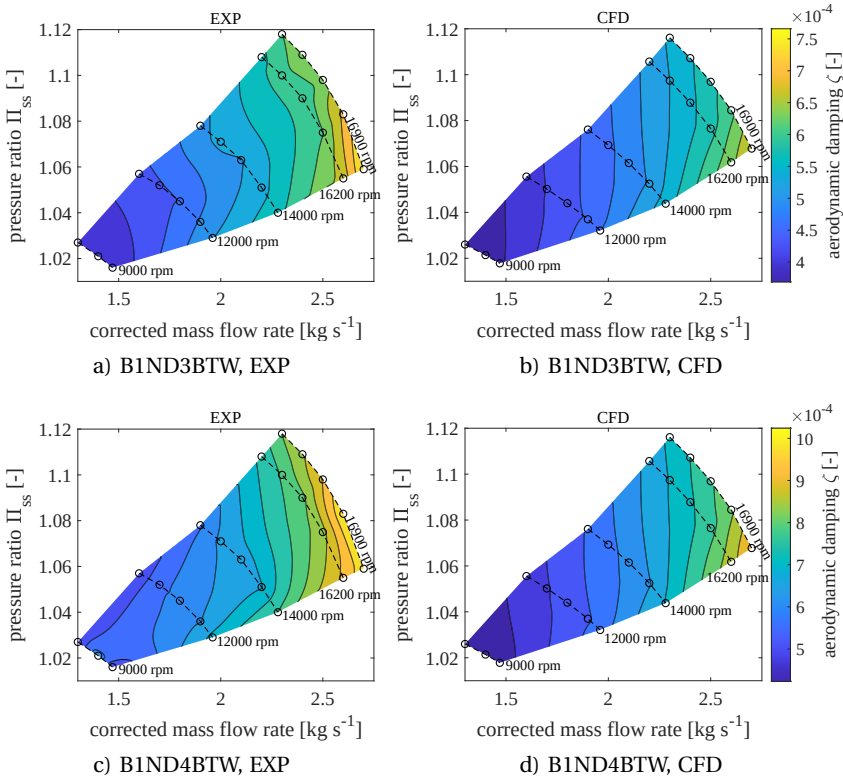


Figure 11.9: Comparison of aerodynamic damping contours on the compressor map for B1ND3BTW and B1ND4BTW. For each mode, the same color scale is used for CFD and experimental data. $p_{t,in} = 500 \text{ mbar}$

For B1ND4BTW, a comparably good result is achieved for the CFD prediction of the aerodynamic damping distribution, cf. figs. 11.10c and 11.10d. Again, the distribution shows an increase in damping with mass flow rate, which is further discussed in the paragraph below. The difference between minimum and maximum damping is slightly increased compared to B1ND3FTW with 101.6 % in the measurements and 124.3 % for the CFD simulations.

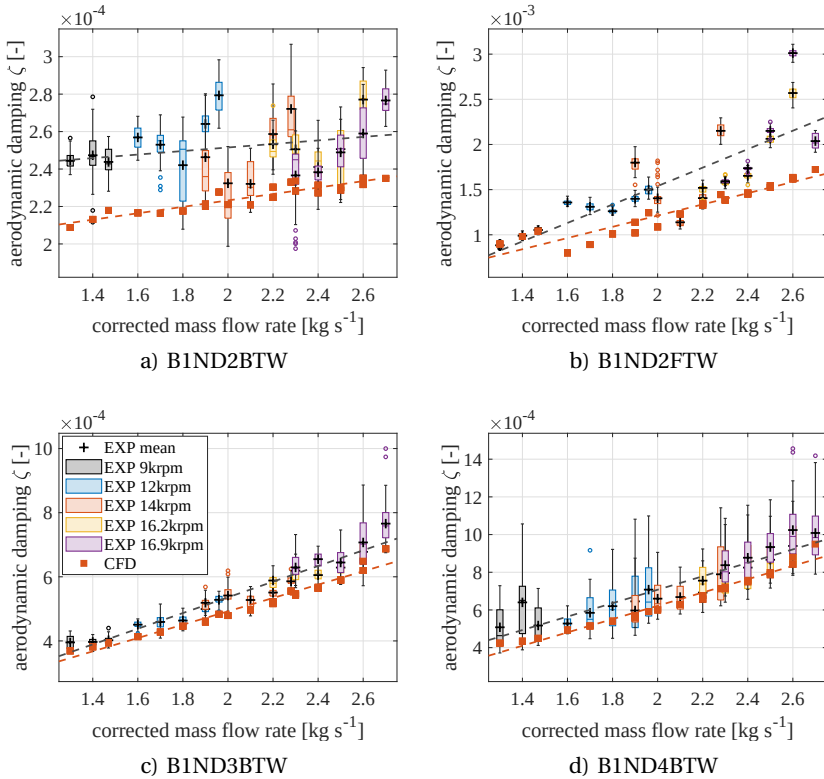


Figure 11.10: Aerodynamic damping vs. corrected mass flow rate

As already discussed in section 2.4.2, there are several publications indicating that aerodynamic damping scales with the mass flow rate to a certain extent. In order to evaluate this for the present investigation, the aerodynamic damping at all operating points with the corresponding corrected mass flow rate is depicted in figs. 11.10a to 11.10d for all four modes and all speed lines. The dashed lines represent linear fits of the CFD and experimental data, respectively. In agreement with the observations regarding the aero-

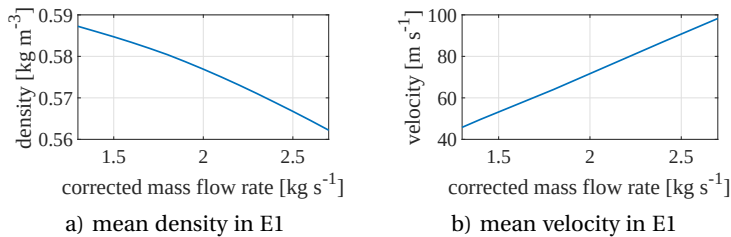


Figure 11.11: Variation in air density and velocity in E1 for different mass flow rates, if mean total pressure at the inlet is kept constant at $p_{t,in} = 500\text{mbar}$

dynamic damping contours discussed above, a linear correlation is visible for B1ND3BTW and B1ND4FTW, where the aerodynamic damping increases with increasing mass flow rate, regardless of the rotational speed. This trend is confirmed in the measurements as well as in the CFD data for both modes, with almost the same gradient for both data sets (EXP and CFD) but clearly different gradients for each mode. For B1ND2FTW, see fig. 11.10b, minor variations in the linear trend are visible for the CFD data and significant variations are present for the experimental results due to the influence of acoustic resonance. However, a linear trend is still detected for the measurements. Only a minor increase in aerodynamic damping is present for B1ND2BTW, see fig. 11.10a, and the aerodynamic damping distribution is dominated by significant variations within a speed line. These variations are less pronounced in the CFD data, where a linear trend is also visible.

Based on the observations presented above, it can be concluded that for all four modes the aerodynamic damping scales linearly with the mass flow rate but with significantly different gradients for each mode. It has to be noted that this linear trend with the mass flow rate is not driven by an increase in air density, which even slightly decreases by 4.3 % between 1.3 and 2.7 kg s^{-1} , see fig. 11.11a, since the total pressure at the inlet is kept constant. It is rather induced by the variation in velocity, which increases linearly with the mass flow rate: $\dot{m} = \rho \cdot c \cdot A$, see fig. 11.11b. The linear dependency on mass flow rate can be considered as the primary influence factor for aerodynamic damping. In addition, secondary factors, such as variations in the incidence angle be-

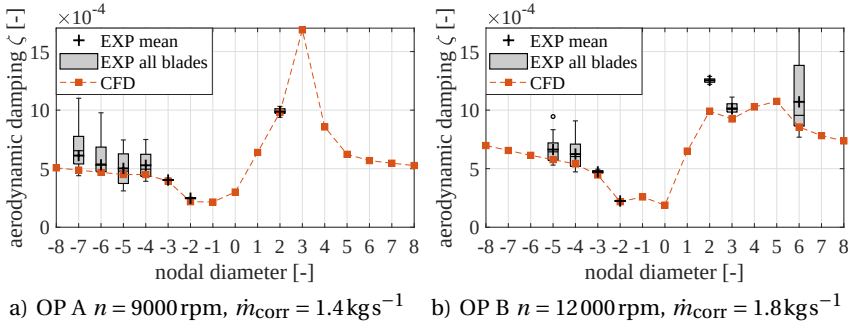


Figure 11.12: Aerodynamic damping for multiple modes of the first bending mode family B1 at two different operating points, $p_{\text{t,in}} = 500 \text{ mbar}$

tween different operating points on a speed line or local flow separations, influence aerodynamic damping. In case the gradient regarding mass flow rate is very small, as for example for B1ND2BTW, the secondary factors become dominant. Similar secondary factors are present for example for B1ND3, but are less pronounced in the damping distribution since the dependency on mass flow rate is stronger for this mode. The results of B1ND2FTW lead to the conclusion that an acoustic resonance can also be considered as a secondary, but very strong, influence superimposed on the general trend of increasing aerodynamic damping with mass flow rate.

11.3 S-Curves

A very common depiction for aerodynamic damping of a blade dominated mode family is the S-curve, where the aerodynamic damping is plotted against the different nodal diameters. While the mode family investigated in the present work is disk dominated for the first few nodal diameters, S-curves nevertheless can be created as it is done in figs. 11.12a and 11.12b. However, it has to be noted that every nodal diameter represents one of the isolated double modes, which differ significantly in their natural frequen-

cies. In fig. 11.12a, the aerodynamic damping is predicted by means of CFD simulations for all possible modes of the B1 mode family in BTW and FTW excitation at operating point A and $p_{t,in} = 500\text{ mbar}$. In addition, aerodynamic damping measurements are conducted for $n_{nd} = -7$ to -2 and $+2$. The comparison of CFD and experimental results reveals a very good agreement. The CFD simulations are able to reproduce the mean of the measured aerodynamic damping sufficiently well with a maximum difference of 20.3 % at B1ND7BTW ($n_{nd} = -7$) and a mean difference of 10.4 %. The typical sinusoidal distribution of the aerodynamic damping is also visible. Furthermore, an acoustic resonance can be seen for $n_{nd} = +3$ with a significant increase in damping. The acoustic resonance is also predicted in fig. 11.6a, where the cut-off frequency of the downstream duct almost coincides with the natural frequency of B1ND3FTW at OP A. This is interesting to note, since the downstream acoustic resonance for B1ND2FTW at 16900 rpm was not reproduced by CFD. Unfortunately, no aerodynamic damping measurements were performed for B1ND3FTW at OP A, which should be done in future investigations in order to validate the acoustic resonance.

A similar comparison is presented in fig. 11.12b at OP B at 12000 rpm. While $n_{nd} = -7$ and -6 are not measured, $n_{nd} = 3$ and $n_{nd} = 6$ are included in the experimental data. Measurements were also done for $n_{nd} = 4$ and 5 , but insufficient displacement amplitudes and significant noise in the BTT data prevented a reliable damping extraction. The mean CFD error at this operating point is similar to OP A with 12.1 %. The maximum difference is detected for B1ND2FTW with 20.9 %.

11.4 Unsteady Casing Pressure at Constant Excitation

In addition to aerodynamic damping measurements, the unsteady pressure sensors installed in the casing between measurement plane E1 and E3, see section 5.3.3, provide insight into the unsteady pressure field induced by blade vibration. For this purpose, constant excitation measurements are performed, where a certain blisk mode is excited with a constant frequency and amplitude. Once a constant blade vibration amplitude is reached, which

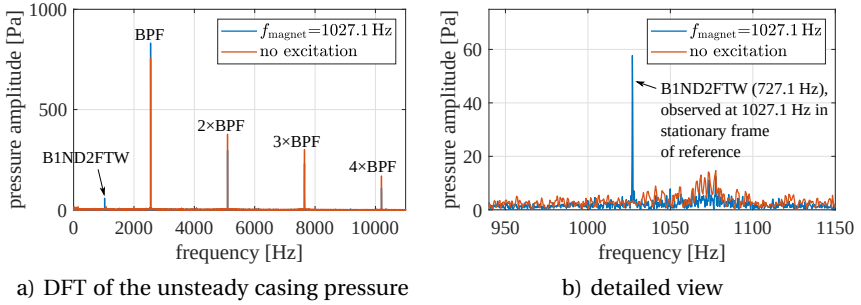


Figure 11.13: Comparison of the unsteady casing pressure frequency content at K9 with and without the excitation of a B1ND2FTW in OP A.

happens with almost no delay after turning on the electromagnetic excitation system, the unsteady pressure is acquired for a measurement time of 5 s with a high sampling rate (400 kHz). An example of such a measurement is depicted in fig. 11.13a, where B1ND2FTW is excited. The pressure signal of one sensor above the rotor (K9, $z_{\text{norm}} = 0.65$) is transformed into the frequency domain by applying a DFT. For comparison, the same measurement is conducted without any external excitation. First, it can be clearly seen that the unsteady pressure is dominated by the blade passing frequency (BPF) and its harmonics. Second, an additional peak appears below the first BPF for the excitation measurement, which can be attributed to the mode vibration. Besides, no other significant changes of the pressure signal in the frequency domain are observed. A detailed view on the mode vibration peak is shown in fig. 11.13b. It is interesting to note that the pressure peak is detected at 1027.1 Hz due to the transformation of the blisk vibration frequency of 727.1 Hz from the rotating into the stationary frame of reference. The peak thus appears at the same frequency as the excitation frequency in the stationary frame of reference, which is prescribed for the electromagnets, see section 5.5.3.

A series of constant excitation measurements for B1ND2BTW at OP A and $p_{\text{t,in}} = 970 \text{ mbar}$ is presented in fig. 11.14a. The mode is excited to four differ-

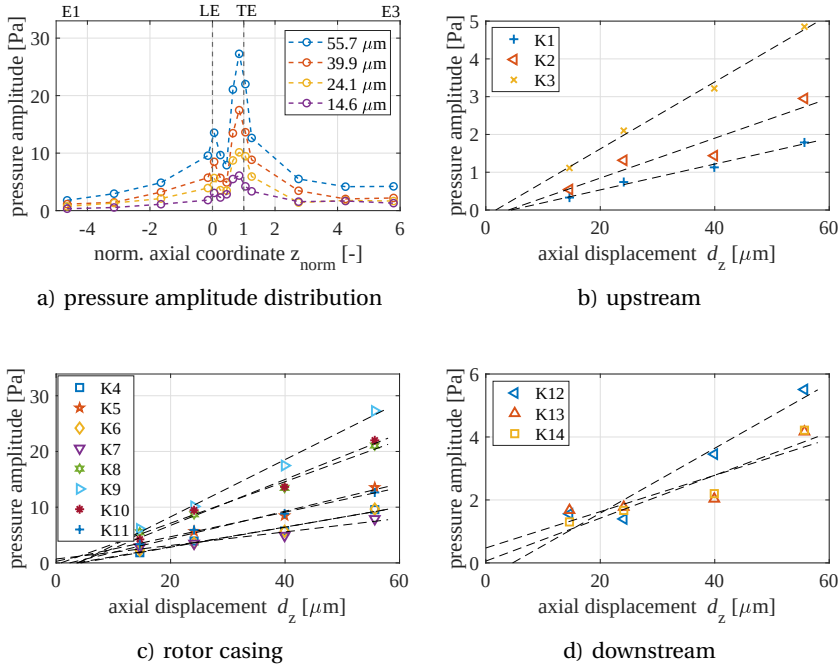


Figure 11.14: Influence of blade displacement amplitude on unsteady casing pressure amplitudes for B1ND2BTW, OP A, $p_{t,in} = 970 \text{ mbar}$

ent blade displacement amplitudes while maintaining the operating point. In general, the highest pressure amplitudes are observed in the region of the vibrating blade between $z_{\text{norm}} = 0$ and 1, as expected, with maximum towards the trailing edge. The pressure amplitudes decay rapidly in upstream and downstream direction. Furthermore, the qualitative distribution of the pressure amplitudes is maintained for different excitation amplitudes, but the level changes. To quantify the dependency of pressure amplitude on displacement amplitude, figs. 11.14b to 11.14d present the pressure data individually for the regions upstream and downstream of the rotor as well for the

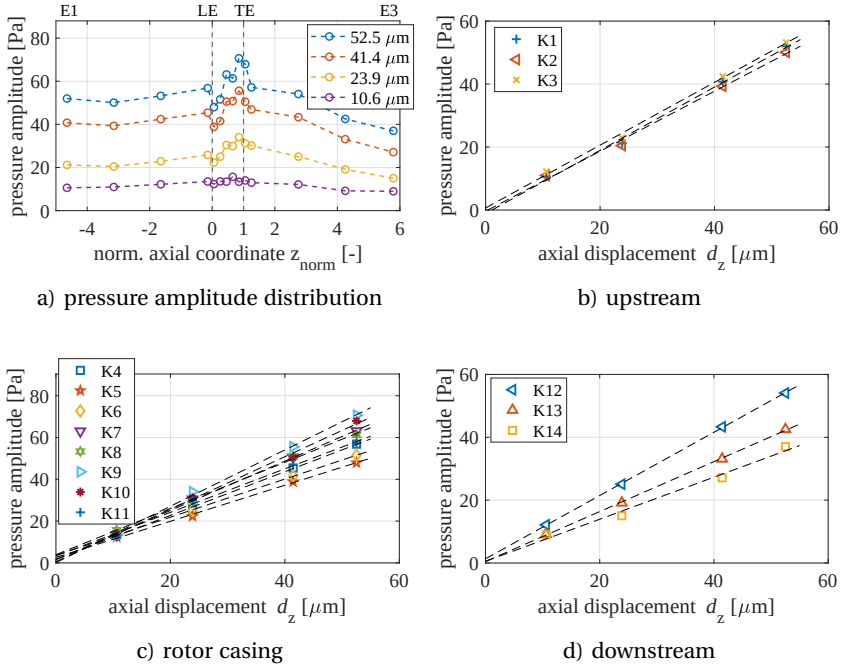


Figure 11.15: Influence of blade displacement amplitude on unsteady casing pressure amplitudes for B1ND2FTW, OP A, $p_{t,in} = 970$ mbar

sensors directly in the rotor casing. For all three regions, a clear linear trend can be observed and the pressure amplitudes increase with increasing displacement. For the upstream sensors, the linear trend can be measured down to pressure amplitudes of approximately 1 Pa. In the region downstream of the rotor, the pressure noise level due to turbulence induced by the rotor blades prevents accurate measurements of amplitudes below 4 Pa. A linear trend can be recognized nevertheless.

In a second series of measurements, B1NDFTW is investigated in the same manner. The results in terms of pressure and displacement amplitudes are

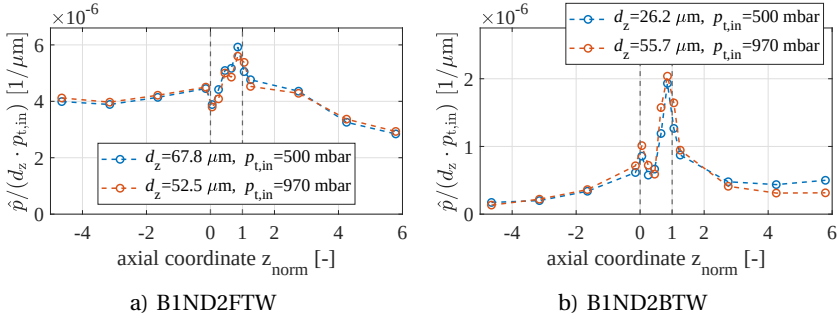


Figure 11.16: Comparison of normalized pressure amplitudes, OP A

shown in figs. 11.14a to 11.14d. First, the measurements confirm the linear dependency of pressure and displacement amplitude. Second, a significantly different distribution of the pressure amplitudes is observed for this mode, see fig. 11.14a. The highest pressure amplitudes are also located in the region of the blade, but the decay in downstream direction is significantly weaker compared to B1ND2BTW. In the upstream region, almost no decay of the pressure amplitudes is observed towards the inlet measurement plane E1. This leads to the assumption that the excited mode is cut-on in upstream and cut-off but close to an acoustic resonance in the downstream region. However, the CFD simulation, see discussion in the paragraph below, and the simplified estimation of the cut-off frequencies, see fig. 11.6a, predict a cut-off condition in both regions. Considering the simplifications in both methods, where a cylindrical duct is modeled and assumed without the consideration of the inlet bulb and the upstream struts or the downstream diffuser and volute, see fig. 6.3b, the difference between the measured and the predicted acoustic conditions can be explained.

The measurements above show a clear correlation of pressure amplitudes and blade displacement. In addition, a correlation of the aerodynamic damping and the static pressure level is confirmed in section 11.1. Both findings are combined in fig. 11.16a, where the distribution of the pressure amplitudes for B1ND2FTW, measured at two different inlet pressure con-

ditions and displacement amplitudes, is depicted in a normalized manner. The normalization is realized with the maximum blade displacement amplitude and the static pressure at the inlet. The agreement of both normalized pressure distributions is very good, which leads to the conclusion that the qualitative characteristic of the unsteady pressure field can be considered independent of the displacement amplitude and pressure level in the investigated ranges. No measurements are conducted at lower pressure levels, but it is assumed that for very low pressure levels, Reynolds number effects significantly influence the flow field, which may influence the unsteady pressure distribution and limit the normalization approach. The measurement at $p_{t,in} = 500 \text{ mbar}$ also confirms the very low decay in upstream and downstream direction and thus the conclusions drawn above for this mode. The possibility to normalize the pressure amplitudes is verified for B1ND2BTW as well, see fig. 11.16b. It is also worth noting that the maximum normalized pressure amplitude for B1ND2FTW is approximately three times higher compared to the BTW excitation of the same double mode, while the aerodynamic damping of B1ND2FTW is approximately four times higher. Thus, a tendency of higher pressure amplitudes for higher aerodynamic damping can be observed, but no correlations should be derived because the pressure measurement is conducted at a few points on the casing and represents an average of the pressure amplitude in circumferential direction without any information of the local amplitude or phase. The aerodynamic damping, however, is determined by the unsteady pressure distribution on the blades, which is not measured in the experiment, and both the amplitude and phase of the local blade pressure are important.

In a last step, the measured pressure amplitudes for constant excitation are compared to results of CFD simulations with the same blade displacement amplitudes.⁷ The numerical results are extracted from the CFD simulations by calculating the average pressure amplitude in circumferential direction at the corresponding unsteady pressure sensor positions.

Figure 11.17a presents a comparison for B1ND2FTW at operating point A. A depiction of the same data for a larger scale in axial direction is depicted

⁷Since a linear correlation of the pressure and displacement amplitudes is shown for the measurements, one could also use an aerodynamic damping CFD simulation with a different prescribed displacement amplitude and scale the results for comparison to the measurements

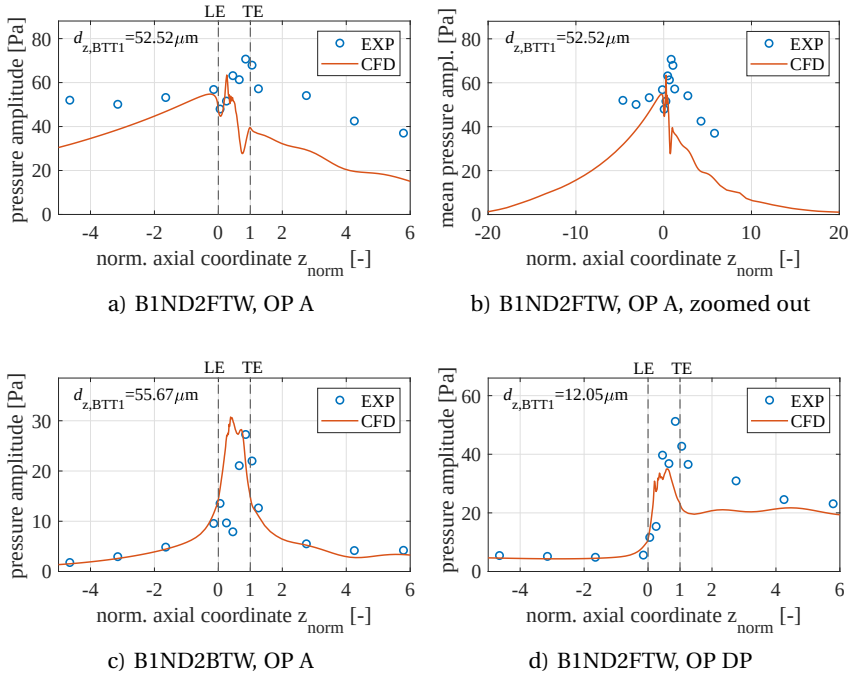


Figure 11.17: Comparison of measured pressure amplitudes to CFD results for different modes and at different operating points, $p_{t,\text{in}} = 970 \text{ mbar}$

in fig. 11.17b. As discussed earlier, the decay is very weak downstream in the experimental data and clearly not visible upstream. The CFD simulation captures the pressure amplitudes sufficiently well between $z_{\text{norm}} = -0.15$ and 0.45 . Further downstream, a shift to lower values by approx. 30 % is visible, even though the agreement in terms of aerodynamic damping is very good ($\Delta\zeta_{\text{CFD/EXP}} = -0.9\%$). However, The CFD results qualitatively capture the decay downstream quite well. In the upstream region, the pressure amplitudes show a decay for the CFD simulation whereas the experimental data seems to show a cut-on condition and thus constant pressure amplitudes. The same

mode is also investigated in the design point, see fig. 11.17d, where a difference of 12.2 % in terms of aerodynamic damping was identified. The pressure amplitudes again agree well in the leading edge region and the CFD underestimates the experimental results between $z_{\text{norm}} = 0.85$ and 4.00. At this operating point, both the CFD as well as the experimental results indicate a cut-on condition upstream and the quantitative agreement is very good. In the downstream region, a cut-on mode is visible for CFD simulation and the experimental results seem to converge to the CFD results towards E3 leading to the assumption that a cut-on condition is also present in the experiment, which is in agreement to the simplified analytic prediction, see fig. 11.6a.

A comparison for B1ND2BTW at OP A is depicted in fig. 11.17c. For this mode, the agreement of the pressure amplitudes is sufficient for all measurement positions except for a limited region downstream of the leading edge, where the CFD simulation predicts large pressure amplitudes. Both, fig. 11.6 and the CFD results indicate cut-off conditions upstream and downstream for B1ND2BTW at this operating point.

11.5 Unsteady Casing Pressure at Sweep Excitation

Unsteady pressure measurements are also conducted for sweep excitation measurements in order to correlate pressure amplitudes at the blade vibration frequency with blade displacement amplitudes. A first example is presented in fig. 11.18a for B1ND2BTW at operating point A, which is excited by means of a frequency sweep between 740 and 745 Hz in 60 s. The figure depicts the mean axial blade displacement on the second axis (black) and the pressure amplitude of the corresponding vibration frequency at K8 ($z_{\text{norm}} = 0.65$) in red on the first axis. The pressure amplitude is determined by a short time Fourier transformation of the pressure signal with a block length of 5 s and an overlap of 90 %. Comparing both response curves, a clear correlation of the blade displacement and the pressure amplitudes can be observed for sweep excitation. This is in agreement to findings of the section above, where the same correlation is identified for constant excitation. Damping is determined from both signals using a SDOF fit and a insignificant difference of 2.7 % is identified. In a second step, damping is calculated for all 14 pressure sensors and the results are depicted on the first axis in

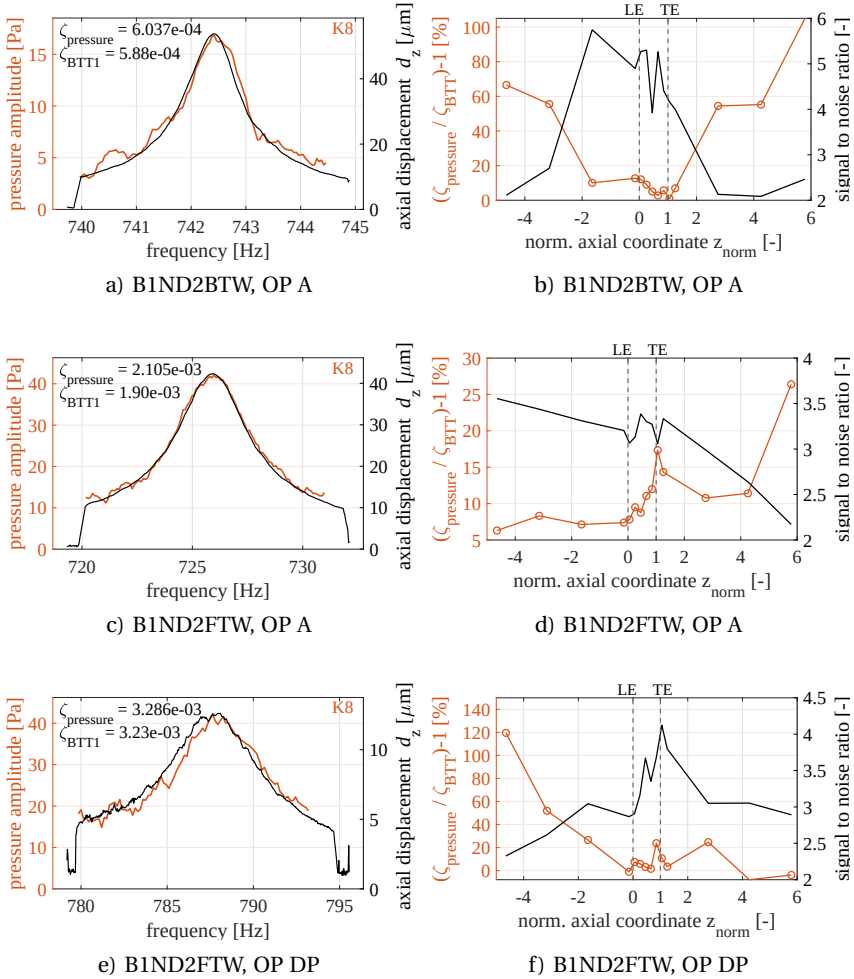


Figure 11.18: Correlation of unsteady pressure amplitudes and blade displacement for sweep excitation experiments

fig. 11.18b in red as a difference between damping from the pressure and the blade displacement response. For all pressure sensors in the rotor casing, the error of damping determined by means of the pressure response is below 20 %. The error increases significantly for sensors further away from the blade in upstream and downstream direction, which can be attributed to very low pressure amplitudes and hence the increasing influence of signal noise. Influence of noise is evaluated by calculating the signal-to-noise ratio (SNR) as the ratio between the maximum pressure amplitude and the mean noise level. The SNR is depicted on the second axis fig. 11.18b and a dependency of the error in damping calculation on the SNR can be seen. The same investigation is conducted for B1ND2FTW in OP A and the design point, see figs. 11.18c to 11.18f, where damping can be calculated based on the pressure data with very low errors as well. In general, the investigation reveals that a signal-to-noise ratio > 3 is required to achieve an error in the damping determination $< 20\%$.

This method works well since the blade vibration frequency is sufficiently far away from multiples of the BPF. In the case of forced response, where the excitation and response frequency both are multiples of the rotational speed, a more complex extraction method can be used, see Leng and Key [142], who use data of three unsteady pressure sensors in circumferential direction and a temporal-spatial analysis method.

11.6 Capability and Limitations of the Excitation System

For the investigations presented in the sections above, 401 excitation experiments were conducted at $p_{t,in} = 500\text{ mbar}$ in total for various modes. This data base is investigated with regards to the achieved blade displacement. The maximum displacement amplitudes for each experiment are depicted in fig. 11.19 with gray circles, where the excited modes are distinguished by the nodal diameter and traveling wave direction. Furthermore, the required excitation frequency for each mode is depicted on the second axis in red. A simplification is made as only the required excitation frequency for 9000 rpm is depicted to discuss general trends, please see fig. 5.15 for the excitation

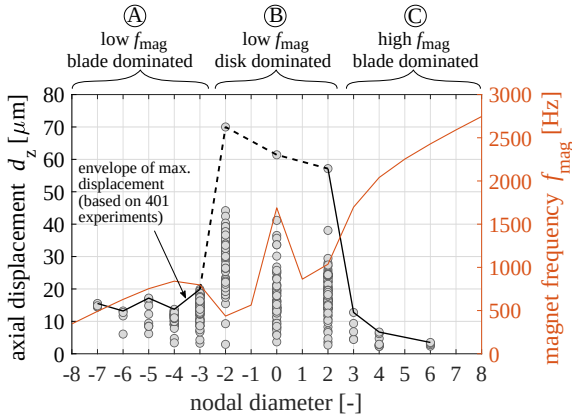


Figure 11.19: Maximum axial blade displacements of all measurements conducted at $p_{t,in} = 500 \text{ mbar}$, excitation frequencies calculated for $n = 9000 \text{ rpm}$

frequencies at higher rotational speeds. An envelope on the maximum displacement amplitudes is drawn with a solid and a dashed black line. The solid lines represent a hard limit of the excitation system, which means that the electromagnetic excitation system was used with the maximum number of magnets and the maximum electric current in the coils, which is limited by the individual, frequency-dependent electrical impedance and thermal limits of the coils and the blisk. The dashed line simply represents the envelope of the maximum displacements in the current work but significantly higher amplitudes would be possible for these modes since the excitation system was not used in its limits.

Three main regions can be identified in 11.19 allowing for general conclusions on the capability and limitations of the excitation system. In region (B), the highest displacement amplitudes can be achieved due to the combination of reasonably low excitation frequencies and disk dominated modes. Since the electromagnets are placed upstream of the blisk hub and thus induce a force in the disk region of the blisk, disk dominated modes are easier to excite compared to blade dominated modes with less displacement in the hub disk. For region (A), the required excitation frequencies are also compa-

rably low due to BTW excitation, which enable high magnet forces due to low losses. However, the excitation of blade dominated modes in the hub disk region is not ideal and requires high magnetic forces, which brings the excitation system to its limits. Nevertheless, displacement amplitudes between 15 and 20 μm are realized with the current setup, which is sufficient for the determination of aerodynamic damping. The lowest displacement amplitudes are achieved in region ③, where the FTW excitation of blade dominated modes requires high magnet frequencies. Eddy current and hysteresis losses significantly reduce the resulting magnetic force leading to very low displacement amplitudes for high nodal diameter modes. For most of the FTW modes with $n_{\text{nd}} > 3$, the excitation force was insufficient for reliable determination of aerodynamic damping.

Since a linear dependency of aerodynamic damping on the pressure level is established in section 11.1, it is assumed that the envelope of the maximum achievable displacement amplitudes in fig. 11.19 can be linearly scaled to other pressure levels. For example, displacement amplitudes between 30 and 40 μm should be possible for region ① at $p_{\text{t,in}} = 250 \text{ mbar}$ without problems.

11.7 Conclusions

The investigations presented in this chapter have three main objectives:

1. provide a large data set of aerodynamic damping measurements and gain physical understanding of how aerodynamic damping changes in the compressor map
2. show the capability and limitations of the electromagnetic excitation system
3. validation of aerodynamic damping CFD simulations

The following conclusions with respect to these objectives can be drawn:

- A clear linear dependency of aerodynamic damping on the static pressure level and hence on the density level is confirmed by means of

measurements for six different modes at two different operating points and four different pressure levels. No significant Reynolds number effects on the aerodynamic damping are detected for the measured pressure range. The linear trend is also confirmed for local casing pressure amplitudes by means of constant excitation experiments.

- Aerodynamic damping is by far the largest contributor to the overall damping. Material damping only represents between 3.3 % and 23.3 % of the overall damping, depending on the mode and operating point.
- A linear correlation of the aerodynamic damping with the mass flow rate is observed for four different modes in first approximation, based on sweep excitation experiments at 23 different operating points on five speed lines. The gradient of the linear correlation is different for each mode and secondary influences on the aerodynamic damping are observed within a speed line superimposing on the linear trend. These findings are in agreement to investigations in open literature, see [97, 156, 184, 201]. However, only axial turbine rotors are investigated in these publications with a limited number of operating points and modes. It is assumed that the measurements in the current work represent the largest set of measurement data published so far, confirming the correlation of aerodynamic damping and mass flow rate for several normal modes of an axial compressor blisk and a wide operating range.
- The amplitudes of the unsteady casing pressure at the mode resonance frequency scale linearly with the blade displacement amplitude. While this linear trend was often shown with CFD in the past, it is assumed that this one of the first times, that the linear correlation is clearly measured by means of constant excitation for an axial compressor blisk in realistic operating conditions. Furthermore, such measurements show the advantage of the presented excitation system, where the magnitude of the excitation force can be adjusted individually and independent of the operating conditions.
- For sweep excitation experiments, it is possible to accurately determine the overall blisk damping from unsteady casing pressure sensors using a short-time Fourier transformation and the same SDOF fit as for BTT data. A signal-to-noise ratio of > 3 is required for errors $< 20\%$.

- An acoustic resonance is observed for B1ND2FTW causing a significant increase in aerodynamic damping, which is in agreement with observations from other, but solely numerical, investigations, for example [47, 74, 156], where acoustic resonance also caused an increase in aerodynamic damping.
- The mean error for the prediction of aerodynamic damping by means of CFD simulations is 12.6 %, based on simulations of the complete compressor map and four different modes ($23 \times 4 = 92$ aerodynamic damping simulations). This agreement is considered to be very good, in particular in comparison to other investigations [67, 105], where errors in the range of an order of magnitude are reported. However, the current CFD model is not able to capture the acoustic resonance for B1ND2FTW, resulting in a maximum error of 45.7 %. It is assumed that the simplifications made in the CFD domain, which is modeled as a strictly axial duct in the upstream and downstream region, are the reason for the limitations of the CFD close to the acoustic resonance. It is therefore indicated for future investigations to use more detailed CFD models, which better reproduce the contour of the compressor away from the axial test section as these details can significantly influence the acoustic resonance conditions. An attempt was made with a CFD setup modeling the inlet bulb and the downstream diffuser, but convergence issues were detected close to the acoustic resonance preventing the use of this setup for productive simulations. Apart from the acoustic resonance, cut-on and cut-off conditions are reproduced sufficiently well by simplified axial CFD model in terms of aerodynamic damping.
- The results confirm that the kinetic energy of both real modes of the double mode creating the traveling wave mode have to be considered in the calculation of the aerodynamic damping with CFD simulations, see appendix A.1.
- The developed test facility including the excitation system shows a very high reliability in terms operating conditions and repeatability of mode excitation. The significant influence of the acoustic resonance for B1ND2FTW at 16900 rpm, for example, was measured on two measurement days and multiple excitation sweeps with insignificant differ-

ences of the results. Over 400 excitation experiments were conducted just for $p_{t,in} = 500$ mbar without problems and electromagnets proved to be a reliable and accurate source of excitation without any intrusion of the fluid flow. However, the maximum amplitude of blade dominated modes is limited using electromagnets in the hub region in particular for forward traveling wave excitation, where very high excitation frequencies are required. As a consequence, the current excitation system is mostly suited for disk dominated modes and backward traveling blade dominated modes.

- In order to achieve larger displacement amplitudes for blade dominated modes, the following improvements or modifications could be implemented to the electromagnet excitation system, which can be derived using eq. (5.3):

$$F_{\text{mag}} = \frac{\mu_0 (N_w I)^2 A}{2 l_{\text{gap}}^2} \quad (5.3)$$

- Larger magnet cores, since the magnetic force scales linearly with the magnet core area A .
- Reduction of the air gap between the magnets and the disk l_{gap} while maintaining a safety margin.⁸
- More windings and increased electric current in the coils. More windings increase the electric impedance, which requires a higher voltage output from the amplifier. Higher electric current also necessitate higher supply voltage. This can be realized by either using an amplifier with a higher voltage output or by introducing capacitors to the electric circuit to compensate the inductive reactance and hence decrease the required supply voltage.

In addition, the electromagnets could be positioned in regions where higher mode displacements are present. For blade dominated modes, an (additional) excitation system in the rotor casing could be effective.

⁸ $l_{\text{gap}} = 1.6$ mm for the present setup

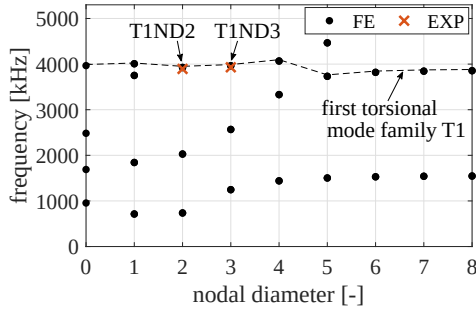
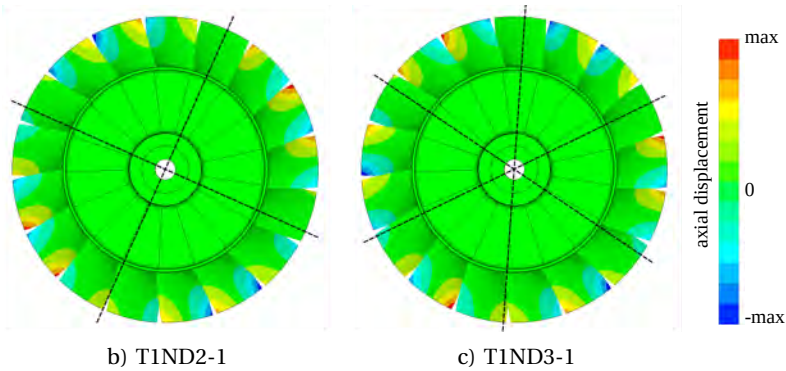
12 Aerodynamic Damping of Torsional Modes

Since all investigations in the previous chapters focused on various modes of the first bending mode family, the capability to use the developed excitation system for the investigation of higher frequency modes as well is presented in the following. For this purpose, the aerodynamic damping of two modes of the first torsional mode family (T1) is determined by means of frequency sweep excitation measurements and the results are compared to CFD simulation results. The experiments are conducted at operating point A ($n = 9000 \text{ rpm}$, $\dot{m}_{\text{corr}} = 1.4 \text{ kg s}^{-1}$) for T1ND2 and T1ND3, which are both excited as backward traveling wave modes.

In a first step, the resonance frequency of both modes at 9000 rpm is determined in vacuum and compared to the prediction of a pre-stressed FE modal analysis. The results are summarized in fig. 12.1a, where a dispersion diagram is depicted. In contrast to the first bending mode family, the individual natural frequencies of the torsional mode family are almost independent of the number of nodal diameter and concentrate at approximately 4000 Hz. The FE results agree well to the measurements for both modes with a minor over-prediction of the natural frequencies of 1.4 % and 1.5 % for T1ND2 and T1ND3, respectively. The reduced frequencies are 9.0 for T1ND2 and 9.1 for T1ND3. Both modes are cut-on in the upstream and downstream duct. The mode shapes are depicted in figs. 12.1b and 12.1c.

Similar to section 11.1, both modes are investigated in terms of overall damping at different inlet pressure levels. The results are presented in figs. 12.1a and 12.1b. For T1ND2BTW, a clear linear trend of the overall damping is observed. The extrapolation method over-predicts the material damping with a minor difference. The aerodynamic damping CFD simulations are able to predict the measured aerodynamic damping accurately with a mean error of 5.6 %.

A similar result is achieved for T1ND3BTW, see fig. 12.1b, where a mean CFD error of 7.8 % is also considered to be sufficiently low. Furthermore, a

a) dispersion diagram at $n = 9000$ rpm

b) T1ND2-1

c) T1ND3-1

Figure 12.1: Dispersion diagram and mode shapes for two torsional modes. The blisk in b) and c) is colored with the mode displacement in axial direction (red = maximum displacement in positive axial direction, blue = maximum displacement in negative axial direction)

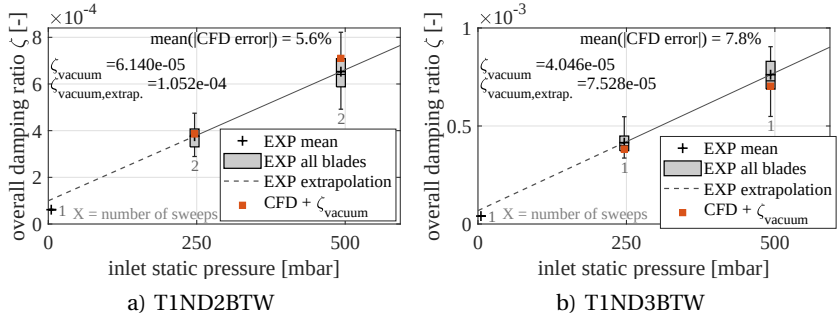


Figure 12.2: Damping measurements vs. CFD at different inlet pressure levels, OP A ($n = 9000\text{rpm}$, $\dot{m}_{\text{corr}} = 1.4\text{kg s}^{-1}$)

linear correlation with the static pressure is also visible for this mode and the extrapolation method is slightly more accurate compared to T1ND2BTW. For both modes, the material damping is relatively close to 5×10^{-5} , which agrees well with the mean material damping of the first bending mode family B1, see fig. 10.5. However, B1ND2 was an exception with higher material damping, which is not the case for T1ND2.

By subtracting the material damping from the overall damping, the mean measured aerodynamic damping is 5.93×10^{-4} and 7.25×10^{-4} , for T1ND2BTW and T1ND3BTW, respectively, at $p_{t,\text{in}} = 500\text{mbar}$. It can be concluded, that the aerodynamic damping of both modes is approximately twice as high as the corresponding modes of the first mode family B1ND2BTW and B1ND3BTW, see fig. 11.12a.

The frequency response curves for T1ND2BTW at all three pressure levels are depicted in figs. 12.3a to 12.3c. The excitation system is used in its limits in terms of coil current to excite this mode at $p_{t,\text{in}} = 250\text{mbar}$ and 500mbar . It becomes apparent that only very low maximum displacement amplitudes can be achieved for this mode with approximately $2\mu\text{m}$ in average for $p_{t,\text{in}} = 500\text{mbar}$. As a consequence, the influence of noise is significant. Nevertheless, the mean blade response clearly resembles a tuned SDOF-like blade response and aerodynamic damping can be extracted with

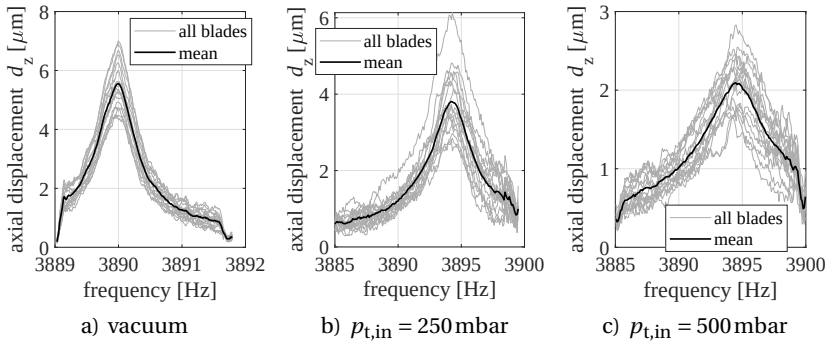


Figure 12.3: T1ND2BTW: Axial blade displacement at BTT1 for different inlet pressure levels

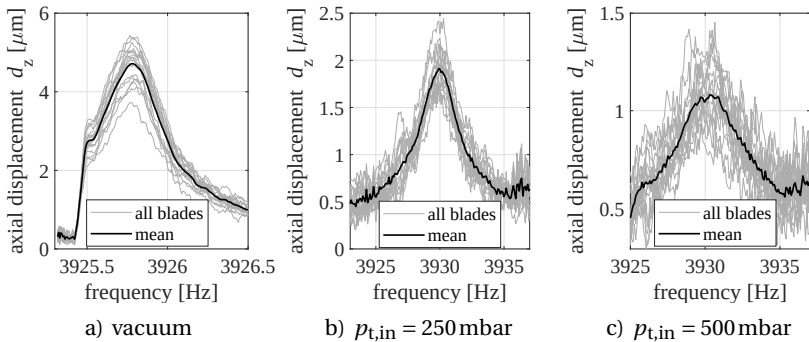


Figure 12.4: T1ND3BTW: Axial blade displacement at BTT1 for different inlet pressure levels

sufficient accuracy for all blades.

For T1ND3BTW, the excitation system is also used with all four electromagnets and the maximum output voltage of the amplifier. It can be seen in figs. 12.4a to 12.4c, that the maximum displacement amplitude reduces by a factor of 2 compared to T1ND2BTW. While the required excitation frequency decreases for BTW excitation of T1ND3BTW, less mode displacement is observed in the hub disk region for this mode, which required significantly higher excitation forces to ultimately reach comparable displacement amplitudes to T1ND2BTW. Regardless of the lower amplitudes, the excitation force is sufficient to achieve a response curve for damping determination. It should be mentioned that the excitation of T1ND3BTW in vacuum was started somewhat “late”, meaning that the frequency sweep was started close to the resonance frequency at 3925.5 Hz, see fig. 12.4a. Nevertheless, the response curve is sufficient to extract the correct damping if the boundaries of the SDOF-fit are set appropriately.

The measurements impressively show that even a displacement amplitude of $1\text{ }\mu\text{m}$ can be sufficient to accurately determine aerodynamic damping with a mean difference to CFD of 7.8%. Of course this requires a very accurate blade tip timing system with a low noise level and a reliable excitation system, as used for the present investigation. However, the measurements also clearly show the limits of the current excitation system and it is assumed that modes with higher numbers of nodal diameter cannot be excited sufficiently well. While forward traveling wave modes were not investigated, it is also assumed that the frequency requirement for FTW excitation is too high for mode family T1.

It can be concluded that it is possible to excite and successfully measure aerodynamic damping for modes of the first torsional mode family using an electromagnetic excitation system in the hub. Furthermore, the aerodynamic damping for modes with cut-on conditions both upstream and downstream and with reduced frequencies of approximately 9 can be predicted with the current CFD setup with errors $\leq 7.8\%$.

13 Summary and Outlook

The accurate prediction of aerodynamic damping is essential in the aeromechanical design of turbomachinery. Nowadays, CFD simulations are increasingly used to determine aerodynamic damping in early stages of the design process, but often with high uncertainties. Unfortunately, there is very few experimental data available in open literature to validate and improve numerical methods with regards to the prediction of aerodynamic damping. In the present thesis, a new aeromechanical test facility is introduced, which allows for the controlled and non-intrusive excitation and vibration measurement of axial compressor rotors at relevant operating conditions and varying inlet conditions. The test facility is used to investigate an axial compressor blisk in detail and the experimental results are compared to accompanying CFD and FE simulations.

The new test facility is called **ACTIVE** (Axial Compressor Test rig for the Investigation of blisk Vibrations with an active Excitation system) and is located at the Institute of Thermal Turbomachinery and Machinery Laboratory (ITSM) of the University of Stuttgart. Since the test facility is built as a closed loop, the total pressure at the inlet of the compressor section can be controlled and experiments at pressure levels below and above atmospheric pressure as well as at vacuum can be conducted. An electromagnetic excitation system is developed and installed in the hub upstream of the rotor to enable the excitation of synchronous and non-synchronous vibrations without disturbing the flow field. Blade vibrations are measured by means of an optical blade tip-timing system (BTT) and the test facility is equipped with multiple miniature fast-response pressure sensors as well as traverse systems to accurately measure the flow field. Aerodynamic damping is determined by measuring damping at vacuum and subsequently subtracting the result from damping measurements of the blisk operating in air. The test section is designed as an isolated rotor case without a stator row close to the rotor yielding a clean test case which is comparable to the standard isolated rotor setup in aerodynamic damping CFD simulations. However, the compressor

section comprises a modular design, where upstream and downstream stator rows can be implemented easily.

In preparation of the vibration measurements, the BTT system is validated using an unique experimental setup, where a scanning laser Doppler vibrometer (SLDV) is used to simultaneously measure the blade vibration in the rotating frame of reference with the help of an optical derotator synchronized to the shaft speed. The measurements are performed at vacuum through an optical access upstream of the compressor section and reveal a mean error of the BTT system below 5 % in terms of peak blade displacement. It is also shown that the BTT system is able to reliably measure blade displacement amplitudes as small as 2 μm .

The normal modes of the axial compressor blisk, which was designed as a test object within the present work, are determined by means of experimental modal analyses (EMA). First, measurements are performed at near free-free conditions and compared to a FE modal analysis. Subsequently, the influence of the shaft clamping is determined using a second EMA, where the blisk is mounted in the compressor. No significant mistuning can be detected for the first bending mode family (B1) except for the two highest nodal diameters (seven and eight), and the clamping condition only influences modes with zero and one nodal diameter. The tuned FE modal analysis is able to reproduce normal modes and natural frequencies sufficiently well. Campbell diagram measurements using the BTT system furthermore reveal that a significant Coriolis force effect is present for some modes of B1 causing a split of the backward and forward traveling wave modes of up to 30 Hz at the design rotational speed of 16200 rpm. It is shown that FE simulations with the consideration of the Coriolis force are able to reproduce the frequency split very accurately.

A damping measurement study shows that frequency sweep excitation is more accurate for aerodynamic damping measurements with a BTT system for the present case. While free decay measurements are feasible at vacuum, the contribution of aerodynamic damping significantly reduces the observation time of the free decay and thus introduces large errors to the determination of damping. A sensitivity study is conducted regarding the frequency sweep rate and experimentally confirms the findings of Hackenberg & Hartung [89, 90], where a reduced sweep velocity of $\kappa \leq 1$ is defined to obtain a

steady blade response.

The overall uncertainty in the experimental determination of damping can be estimated with 10 %. This uncertainty is determined by means of reproduction measurements for multiple modes and at multiple operating points as well as for different displacement amplitudes.

Due to the electromagnetic excitation, detailed damping measurements are conducted at vacuum allowing the determination of the material damping of the blisk (structural damping is assumed to be negligible for the investigated blisk). Based on damping measurements at various rotational speeds, the material damping (blisk material is stainless steel 17-4 PH) can be estimated in good approximation with 0.5×10^{-4} (critical damping ratio ζ) for all modes of the first bending mode family with more than two nodal diameters. This value is also confirmed for modes of the first torsional mode family. No significant influence of the rotational speed was detected.

The main focus of the aerodynamic damping measurements is put on the first bending mode family (B1) with reduced frequencies between 0.96 and 3.66. Measurements at different pressure levels confirm a clear linear correlation of aerodynamic damping with fluid density. In addition, aerodynamic damping measurements for four modes of B1 are conducted for the complete compressor map revealing a linear trend of aerodynamic damping with the mass flow rate. The measurements furthermore show that aerodynamic damping is the dominating damping mechanism with a contribution between 76.7 % and 96.7 % to the overall damping at atmospheric inlet conditions. Traveling wave mode (TWM) CFD simulations using the harmonic balance method (ANSYS CFX 2020 R1) are applied to determine aerodynamic damping numerically. The comparison to measurement results shows a good agreement with a mean error of 12.6 % (based on 275 excitation measurements). The maximum error of 45.7 % is observed for the two nodal diameter mode at forward traveling wave excitation. An acoustic resonance is detected in the downstream duct of the compressor for this mode, which is not captured sufficiently by the simplified CFD model. However, the aerodynamic damping CFD setup is considered validated due to the low mean error with the exception of modes close to acoustic resonance.

The evaluation of the unsteady casing pressure measurements shows a clear linear trend of the pressure amplitudes with the blade displacement ampli-

tudes as well as with the pressure level. Moreover, it is shown for sweep excitation measurements that the unsteady casing pressure at the vibration frequency can be correlated to the displacement amplitude and thus be used to determine blisk damping with high accuracy.

Aerodynamic damping measurements are also conducted for two modes of the first torsional mode family (T1) with a reduced frequency of 9 and 9.1. The CFD simulations show errors below 10 % for these modes and confirm the capability of the CFD model to accurately predict aerodynamic damping.

It is assumed that the test facility built within the scope of this thesis represents one of the first test facilities for axial compressor blisks, where aerodynamic damping can be measured for various modes at relevant operating conditions and independent of the rotational speed or operating point. High accuracy of the aerodynamic damping measurements is achieved by an accurate and validated BTT system and a reliable excitation system along with the ability of the test facility to determine material and structural damping at vacuum without any changes to the test configuration.

Based on the findings of the present thesis, the following suggestions are made for possible future work:

- The simplified CFD model shows its limitations in the prediction of aerodynamic damping for B1ND2FTW in the region of acoustic resonance. As a consequence, more detailed CFD domains and different CFD setups should be investigated. In particular the downstream volute and the region of the upstream inlet bulb should be considered in future CFD models.
- Magnets or electromagnets in the casing above the rotor could improve the excitation of blade dominated modes. Furthermore, the simultaneous excitation of two modes would be possible with a second excitation system. Multiple modes could also be excited simultaneously by using the current excitation system but with different frequencies for each electromagnet.
- A stator row downstream and/or upstream of the rotor could be implemented. Since the electromagnetic excitation does not rely on rotor-stator action, the same blisk mode can be excited with and without

a stator row allowing for the isolated investigation of the influence of stator rows on the aerodynamic damping. Instead of additional stator rows, inlet screens to vary the inlet turbulence level or to generate asymmetric inlet total pressure contours can also be installed easily.

- The current setup only allows for the measurement of circumferentially-averaged pressure amplitudes of the transient pressure field. Additional fast-response pressure transducers in circumferential direction should be installed for future measurements to gain information of the pressure amplitude and phase of the rotating, transient pressure field and thus to provide additional validation data.

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A Appendix to Chapter 2

A.1 Kinetic Energy of Double Modes

Carta derived an expression for the logarithmic decrement of the aerodynamic damping based on the work per cycle and the average kinetic energy, see [31] or [189]:

$$\delta_{\text{aero}} = \frac{-W_{\text{cycle}}}{4W_{\text{KE,ave}}} \quad (\text{A.1})$$

which can be expressed as a critical damping ratio using eq. (2.25)

$$\zeta_{\text{cycle}} = \frac{-W_{\text{cycle}}}{8\pi W_{\text{KE,ave}}} \quad (\text{A.2})$$

For a single mode, the average kinetic energy can be expressed with the maximum kinetic energy within one vibration cycle:

$$W_{\text{KE,ave}} = \frac{1}{2} W_{\text{KE,max}} \quad (\text{A.3})$$

see also fig. A.1a, where the kinetic energy is depicted for a single standing wave mode of a blisk along with the average value for one vibration period T . The maximum kinetic energy for a single mode can be calculated with the mass matrix $\underline{\underline{M}}$, the vibration frequency ω and the displacement vector $\underline{\phi}$ [119]:

$$W_{\text{KE,max}} = \frac{1}{2} \omega^2 \underline{\phi}^T \underline{\underline{M}} \underline{\phi} \quad (\text{A.4})$$

In order to define a specific maximum blade displacement in the mesh moving CFD simulation, the mode displacement is scaled with a factor S_{CFD} :

$$W_{\text{KE,max}} = \frac{1}{2} \omega^2 (\underline{\phi}^T S_{\text{CFD}}) \underline{\underline{M}} (\underline{\phi} S_{\text{CFD}}) = \frac{1}{2} \omega^2 S_{\text{CFD}}^2 \underline{\phi}^T \underline{\underline{M}} \underline{\phi} \quad (\text{A.5})$$

Using mass-normalized eigenvectors, c.f. eq. (2.6), eq. (A.4) simplifies to

$$W_{\text{KE,max}} = \frac{1}{2} \omega^2 S_{\text{CFD}}^2 \quad (\text{A.6})$$

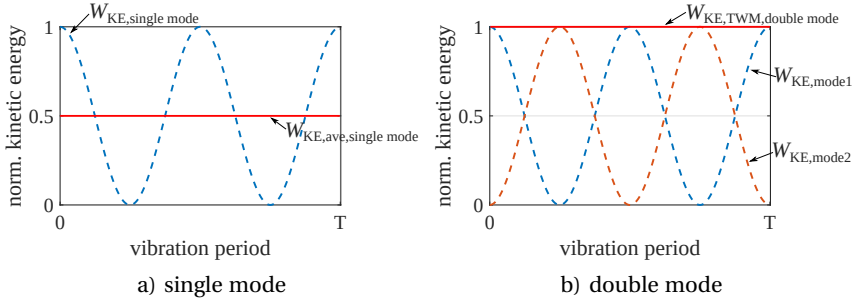


Figure A.1: Normalized depiction of the instantaneous and average kinetic energy of a single standing wave mode and a traveling wave mode of a blisk.

and the aerodynamic damping ratio for a single mode can be calculated with $\omega = 2\pi f$ as:

$$\zeta_{\text{aero}} = \frac{-W_{\text{cycle}}}{8 \pi^3 f^2 S_{\text{CFD}}^2} \quad (\text{A.7})$$

If a traveling wave mode is prescribed in the CFD simulation by superimposing two orthogonal, mass-normalized real modes of a double mode with the same maximum amplitude according to eq. (2.12) and eq. (2.13), the overall kinetic energy of the blisk vibration is constant, see fig. A.1b. The instantaneous kinetic energy of the blisk is a summation of the instantaneous kinetic energy of both modes

$$W_{\text{KE, TWM}} = W_{\text{KE, mode1}} + W_{\text{KE, mode2}} \quad (\text{A.8})$$

and the average kinetic energy is equal to the maximum kinetic energy. Hence, eq. (A.3) becomes:

$$W_{\text{KE, TWM, ave}} = W_{\text{KE, TWM, max}} = \frac{1}{2}(W_{\text{KE, mode1, max}} + W_{\text{KE, mode2, max}}) \quad (\text{A.9})$$

$$\begin{aligned}
 W_{\text{KE,TWM,max}} &= \frac{1}{2} \left(\frac{1}{2} \omega^2 S_{\text{CFD,m1}}^2 \underline{\phi_1}^\top \underline{M} \underline{\phi_1} + \frac{1}{2} \omega^2 S_{\text{CFD,m2}}^2 \underline{\phi_2}^\top \underline{M} \underline{\phi_2} \right) \\
 &= \frac{1}{2} \left(\frac{1}{2} \omega^2 S_{\text{CFD,m1}}^2 + \frac{1}{2} \omega^2 S_{\text{CFD,m2}}^2 \right) \quad (\text{A.10})
 \end{aligned}$$

Using eq. (A.9) and eq. (A.10) in eq. (A.2), the aerodynamic damping ratio can be calculated with

$$\zeta_{\text{aero,TWM}} = \frac{-W_{\text{cycle}}}{(8 \pi^3 f^2 S_{\text{CFD,m1}}^2) + (8 \pi^3 f^2 S_{\text{CFD,m2}}^2)} \approx \frac{-W_{\text{cycle}}}{2 \cdot (8 \pi^3 f^2 S_{\text{CFD}}^2)} \quad (\text{A.11})$$

Since both mass-normalized orthogonal modes share the same maximum length¹, both scaling factors $S_{\text{CFD,m1}}$ and $S_{\text{CFD,m2}}$ are equal and can be simplified to S_{CFD} .

¹except for very small numerical differences, if they are obtained by a FE modal analysis

B Appendix to Chapter 5

B.1 Blade Tip Timing Measurement Conversion

Conversion of displacement measured by the blade tip timing system into axial and circumferential blade displacement.

α = local mode vector angle

β = local blade angle

$$-\tan \alpha = \frac{d_\theta}{d_z} \quad (\text{B.1})$$

$$\tan \beta = \frac{d_{\text{BTT}} - d_\theta}{d_z} \quad (\text{B.2})$$

$$\frac{d_{\text{BTT}} - d_\theta}{\tan \beta} = \frac{d_{\text{BTT}}}{-\tan \alpha} \quad (\text{B.3})$$

$$d_\theta = -d_{\text{BTT}} \frac{\tan \alpha}{\tan \beta - \tan \alpha} \quad (\text{B.4})$$

$$d_z = \frac{d_\theta}{\tan \alpha} \quad (\text{B.5})$$

$$d_z = d_{\text{BTT}} \frac{1}{\tan \beta - \tan \alpha} \quad (\text{B.6})$$

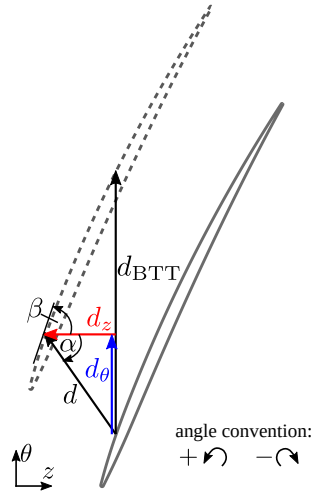


Figure B.1: Blade displacement and measured displacement

Table B.1: BTT conversion factors for first bending mode family B1

	BTT1		BTT2	
n_{nd}	d_z/d_{BTT1}	d_θ/d_{BTT1}	d_z/d_{BTT2}	d_θ/d_{BTT2}
0	0.394	0.003	0.562	0.007
1	0.398	0.014	0.543	0.041
2	0.404	0.028	0.530	0.064
3	0.394	0.003	0.544	0.039
4	0.388	0.011	0.559	0.014
5	0.386	0.017	0.566	0.001
6	0.385	0.020	0.570	0.007
7	0.385	0.021	0.572	0.010
8	0.384	0.021	0.573	0.013

B.2 BTT Sensor Positions

Table B.2: BTT sensor positions

BTT plane	sensor number	position
1	1	56.5°
1	2	129.7°
1	3	180.3°
1	4	256.5°
1	5	298°
1	6	319.7°
2	7	42°
2	8	115.2°
2	9	165.8°
2	10	242°
2	11	283.5°

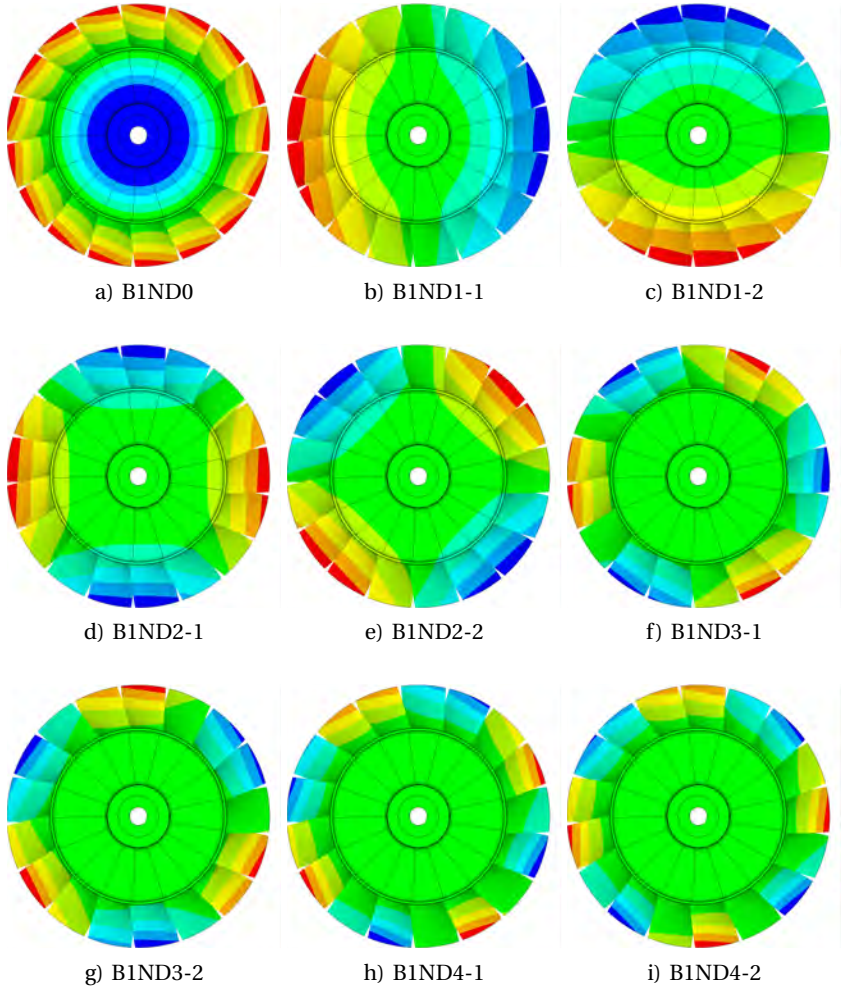


Figure B.2: First bending mode family B1, nodal diameter (ND) 0 to 4. Double modes are marked with “-1” and “-2”. Blisk is colored with the mode displacement in axial direction (red = maximum displacement in positive axial direction, blue = maximum displacement in negative axial direction)

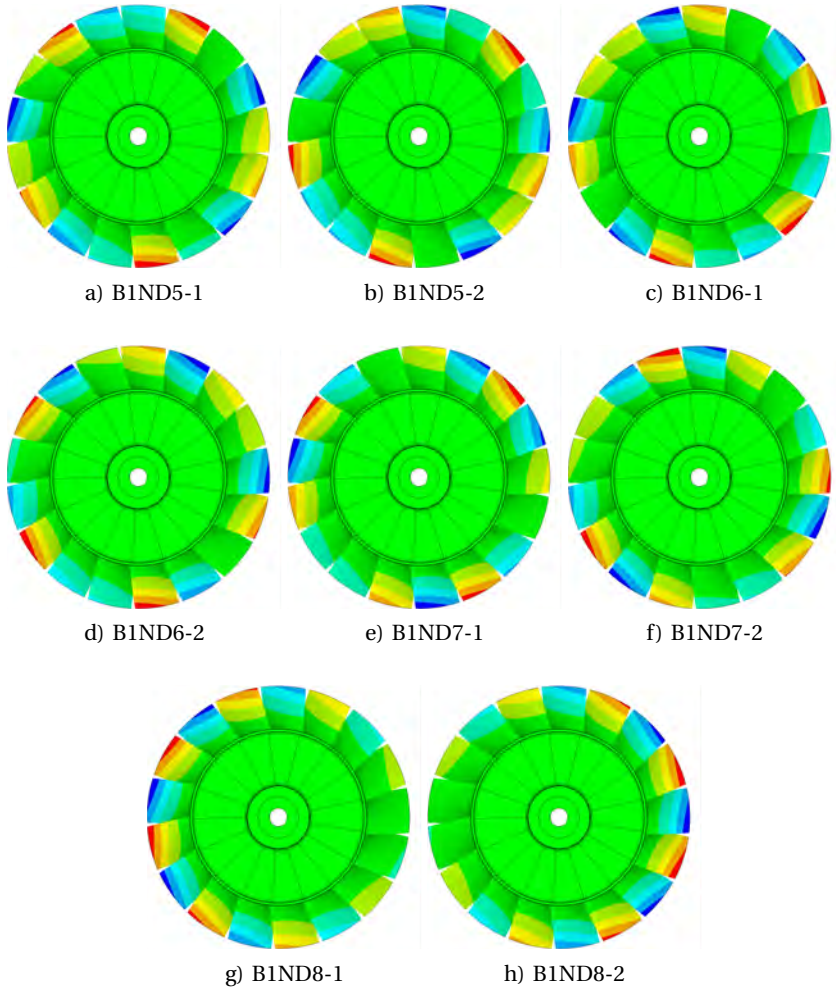


Figure B.3: First bending mode family B1, nodal diameter (ND) 5 to 8. Double modes are marked with “-1” and “-2”. Blisk is colored with the mode displacement in axial direction (red = maximum displacement in positive axial direction, blue = maximum displacement in negative axial direction)

C Appendix to Chapter 6

C.1 Superposition of a Double Mode in CFD

In order to generate a traveling wave mode in CFD simulations, the blade displacement is defined with user expressions and the superposition of the two real modes of a double mode:

$$\underline{\tilde{\phi}}_{12} = \underline{\phi}_1 + i\underline{\phi}_2 \quad (\text{C.1})$$

This complex mode is then rotated with the respective natural frequency ω in time and the real part yields the instantaneous mode displacement, which is prescribed at the blade boundary:

$$\underline{d}_{\text{CFD}}(t) = \Re(\underline{\tilde{\phi}}_{12} \cdot e^{i\omega t}) \quad (\text{C.2})$$

$$= \Re((\underline{\phi}_1 + i\underline{\phi}_2) \cdot e^{i\omega t}) \quad (\text{C.3})$$

Using Euler's identity, eq. (C.3) becomes:

$$\underline{d}_{\text{CFD}}(t) = \Re\left(\underline{\tilde{\phi}}_{12} \cos(\omega t) + i\underline{\tilde{\phi}}_{12} \sin(\omega t)\right) \quad (\text{C.4})$$

$$= \Re\left(\underline{\phi}_1 \cos(\omega t) + i\underline{\phi}_2 \cos(\omega t) + i\underline{\phi}_1 \sin(\omega t) - \underline{\phi}_2 \sin(\omega t)\right) \quad (\text{C.5})$$

$$= \underline{\phi}_1 \cos(\omega t) - \underline{\phi}_2 \sin(\omega t) \quad (\text{C.6})$$

The direction of the traveling wave mode depends on the initial orientation of $\underline{\phi}_1$ and $\underline{\phi}_2$, which is a result of the FE solver and can change for different double modes. As a consequence, the traveling wave direction has to be checked for every double mode prior to the CFD simulations and it can be changed by multiplying $\underline{\phi}_2$ with -1 .

D Appendix to Chapter 8

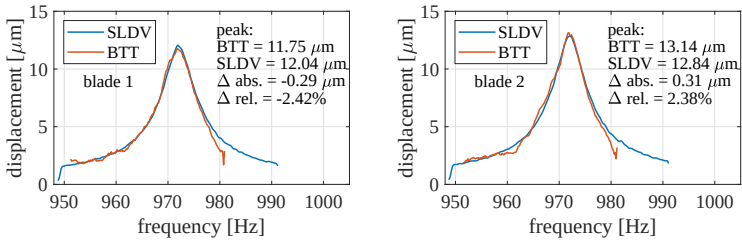


Figure D.1: BTT vs. SLDV: B1ND0 at $n = 6000$ rpm

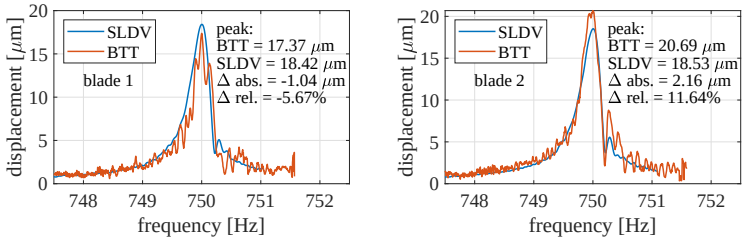


Figure D.2: BTT vs. SLDV: B1ND2BTW at $n = 10000$ rpm

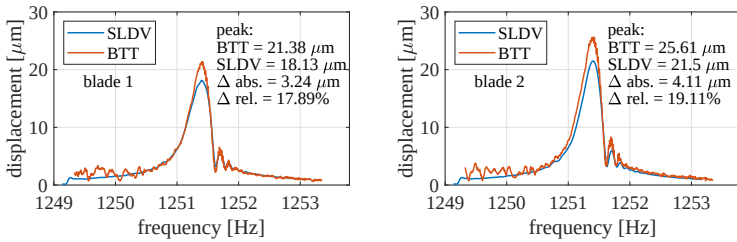


Figure D.3: BTT vs. SLDV: B1ND3BTW at $n = 10000$ rpm

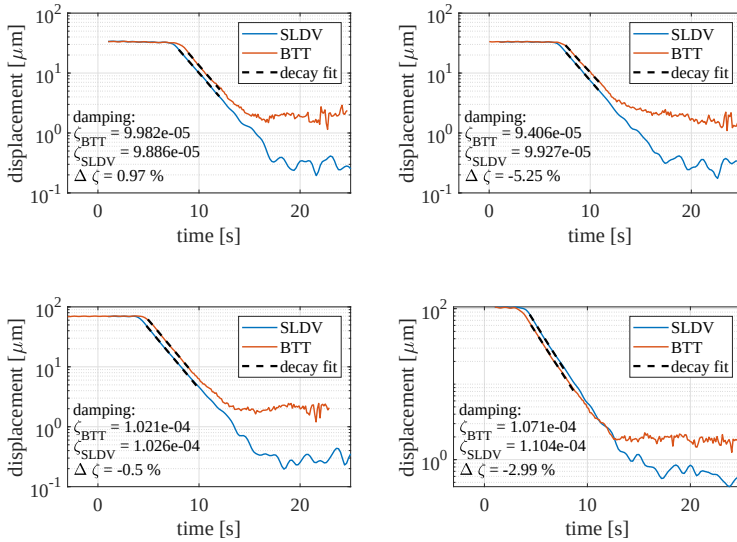


Figure D.4: Free decay measurements of B1ND2BTW at $n = 6000$ rpm with different initial amplitudes, blade 2. BTT data shifted by 1 s for better visibility.

E Appendix to Chapter 9

E.1 Reynolds Number

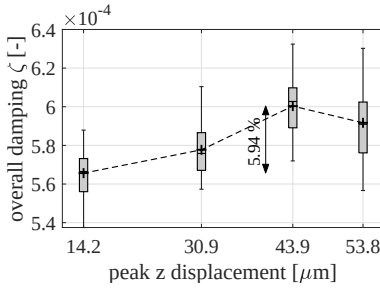
The Reynolds number is calculated based on CFD simulations with the average relative velocity upstream of the blade w , the blade chord length l_{chord} and the kinematic viscosity ν :

$$\text{Re} = \frac{w l_{\text{chord}}}{\nu} \quad (\text{E.1})$$

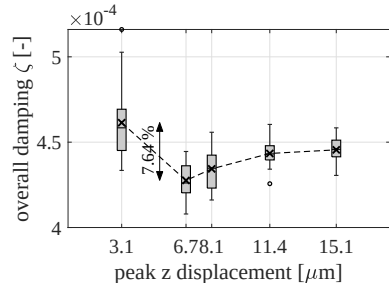
Table E.1: Reynolds number for different operating points and inlet conditions,
 $T_{\text{t,in}} = 294 \text{ K}$

OP	n [rpm]	$\dot{m}_{\text{corr}}$ [kg s^{-1}]	$p_{\text{t,in}} = 970 \text{ mbar}$	$p_{\text{t,in}} = 500 \text{ mbar}$	$p_{\text{t,in}} = 250 \text{ mbar}$
NC	16200	2.6	5.20×10^5	2.68×10^5	1.34×10^5
DP	16200	2.4	5.13×10^5	2.65×10^5	1.32×10^5
NS	16200	2.2	5.07×10^5	2.62×10^5	1.30×10^5
C	14000	2.0	4.45×10^5	2.30×10^5	1.15×10^5
B	12000	1.8	3.86×10^5	1.99×10^5	1.00×10^5
A	9000	1.4	2.94×10^5	1.51×10^5	0.76×10^5

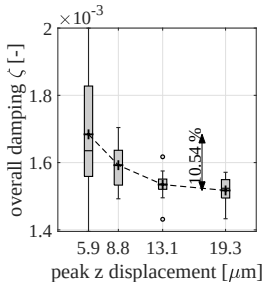
F Appendix to Chapter 10 and 11



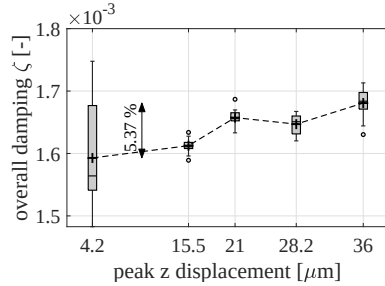
a) B1ND2BTW, OP A, $p_{t,\text{in}} = 970 \text{ mbar}$



b) B1ND3BTW, OP A, $p_{t,\text{in}} = 500 \text{ mbar}$



c) B1ND2FTW, OP C,
 $p_{t,\text{in}} = 970 \text{ mbar}$



d) B1ND3BTW, OP NS, $p_{t,\text{in}} = 970 \text{ mbar}$

Figure F.1: Influence of displacement amplitude on damping measurement, additional measurements see fig. 10.2

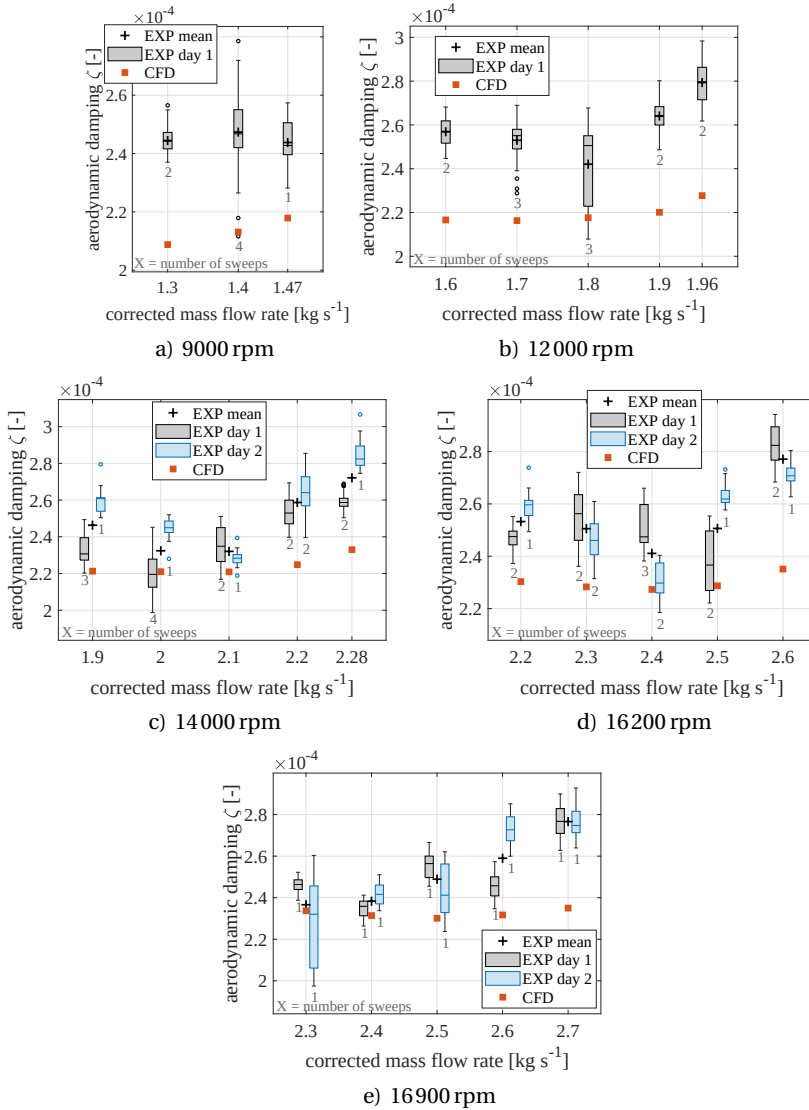


Figure F2: B1ND2BTW, $p_{t,\text{in}} = 500 \text{ mbar}$, comparison of measurements and CFD in terms of aerodynamic damping

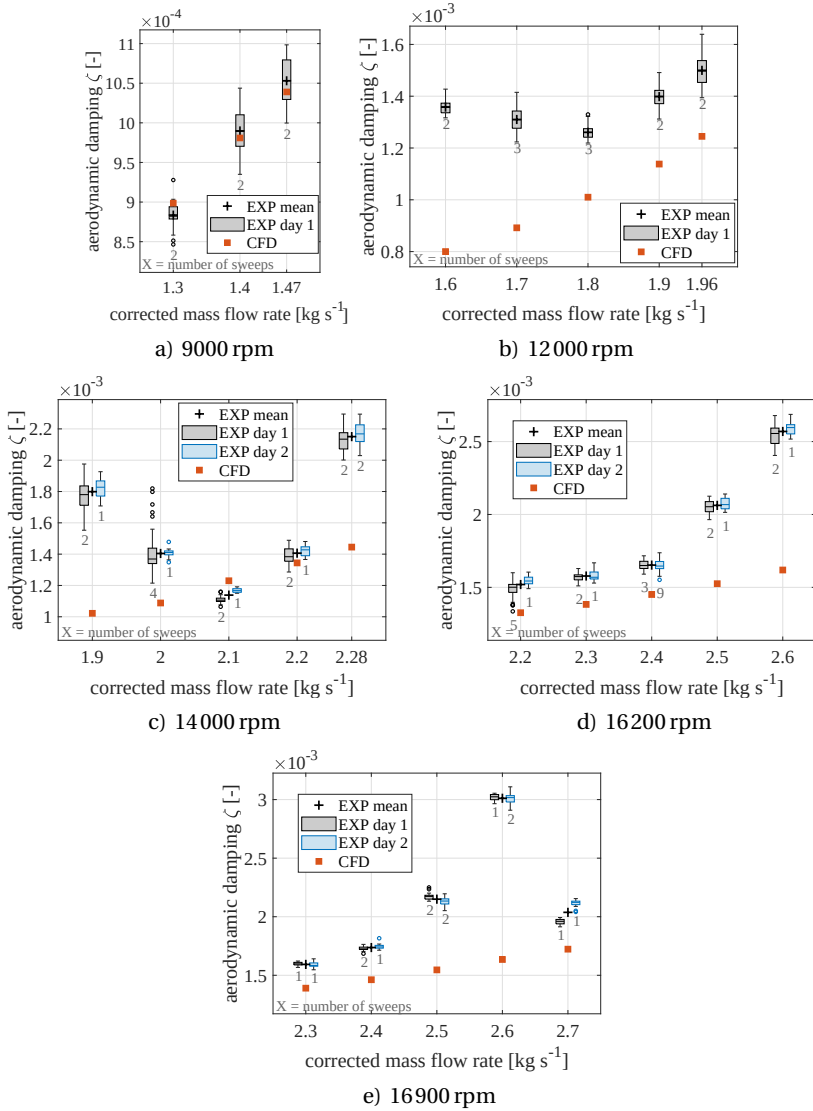


Figure E3: B1ND2FTW, $p_{t,\text{in}} = 500\text{mbar}$, comparison of measurements and CFD in terms of aerodynamic damping

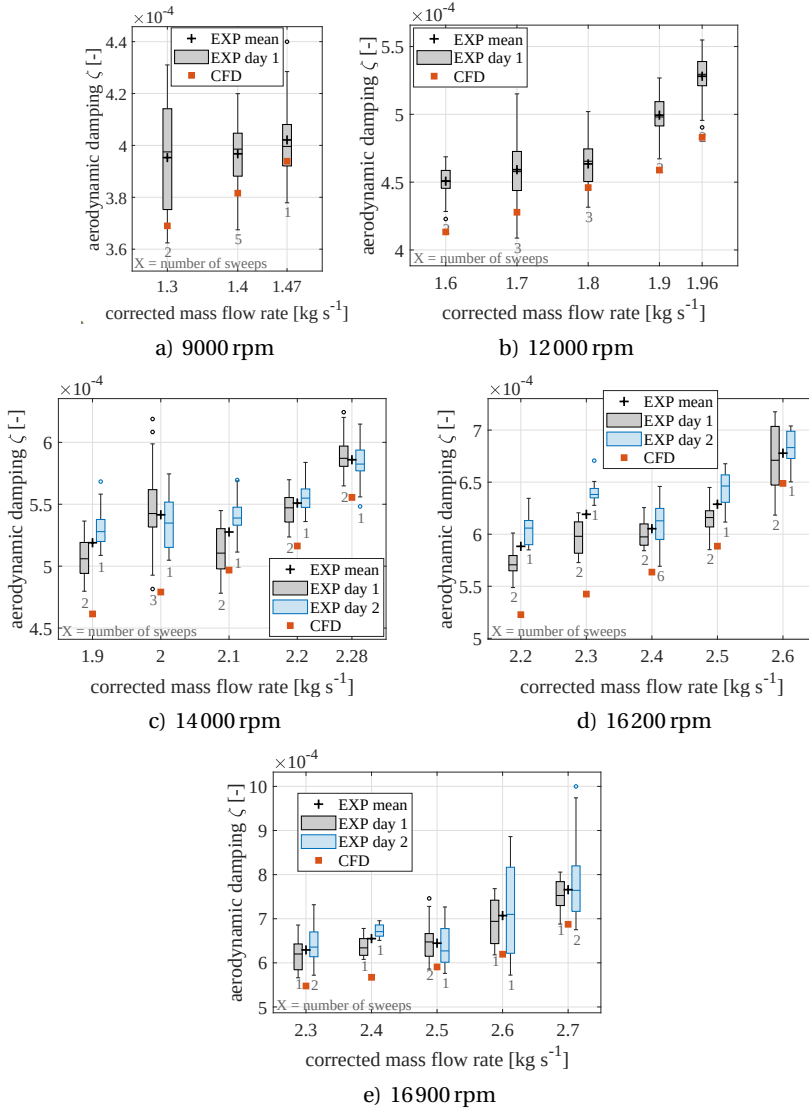


Figure F4: B1ND3BTW, $p_{t,in} = 500$ mbar, comparison of measurements and CFD in terms of aerodynamic damping

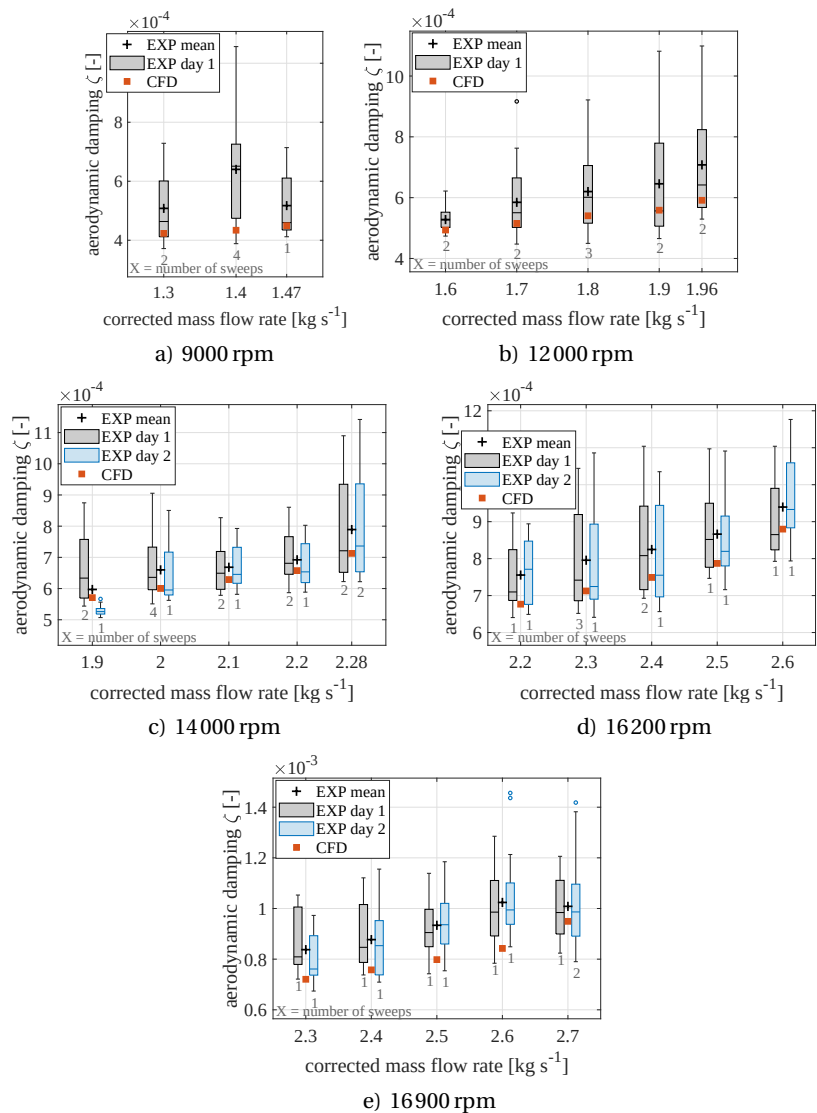


Figure F5: B1ND4BTW, $p_{t,\text{in}} = 500\text{mbar}$, comparison of measurements and CFD in terms of aerodynamic damping