DOCTORAL DISSERTATION

Investigations on Critical Low-Load and Transient Operation of a Prototype Francis Turbine

carried out for the purpose of obtaining the degree of
Doctor technicae (Dr.techn.),
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by

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"Once more unto the breach,
dear friends, once more"

— William Shakespeare, *Henry V, Act III, 1598*

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Erklärung

Diese Arbeit wurde von der Österreichischen Forschungsförderungsgesellschaft (FFG) im Rahmen des BRIDGE-Projekts *MDREST* (Projektnummer 11364358) unterstützt.

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Danksagung

Wie einst Marcus Tullius Cicero sagte "*Keine Schuld ist dringender, als die, Dank zu sagen*". Ich hätte diesen Weg nie erfolgreich beschreiten können ohne die Unterstützung zahlreicher Menschen.

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Abstract

Nowadays, our answer to the challenges of climate change is one of the most pressing concerns facing the public and governments worldwide. In order to reduce the carbon dioxide (CO_2) emissions, the integration of new renewables in the electricity system is a key element.
The use of these volatile sources of energy leads to higher requirements of grid stabilization services. Based on their advantages and possibilities, hydraulic machines are an ideal choice to provide the required balancing energy to the electrical grid. However, Francis turbines are subjected to the dynamic flow phenomena, resulting in unsteady pressure fluctuations in off-design conditions, which may lead to higher risk of fatigue damages and a reduced lifetime. In order to reduce the operating costs and prevent expensive failure events, detailed information of the flow behavior, ensuring a reliable lifetime assessment is required.
The present work targets the further development of numerical fatigue analysis, as site measurements are not feasible and too expensive in most cases. Moreover, to validate the simulations, detailed site measurements on a prototype Francis turbine, including strain gauges on the trailing edge, were performed. Thereby, information about the behavior of the machine set during most of the possible operating regions were gathered. Later on, critical operating regions were examined and a lifetime assignment based on the experimental data is presented.
Furthermore, steady and unsteady computational fluid dynamic (CFD) analyses with the commercial code ANSYS-CFX and finite element method (FEM) computations were performed using ANSYS-Mechanical. The most critical low-load operating point was analyzed by means of hybrid, RANS-LES turbulence models. Thereby, a tube vortex structure leading to high pressure pulsations was revealed. Moreover, it was also possible to detect vortex shedding in this operating region by the use of higher resolution grids and high performance computing (HPC). Consequently, a transient CFD approach was developed and tested on load rejection and two start up cases. The applicability and accuracy of the numerical approach is validated by the measurement data.

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Kurzfassung

In den nächsten Jahren ist es eine der größten Herausforderungen der Menschheit, Antworten auf die Auswirkungen des Klimawandels zu finden. Um den CO_2-Ausstoß entscheidend zu vermindern, ist eine Integration von neuen erneuerbaren Energieträgern in das bestehende Stromnetz von Nöten. Das Problem dabei ist, dass Energiequellen wie Wind und Sonne vom Wetter abhängen und damit eine volatile Energiemenge in das bestehende Stromnetz einspeisen. Dies führt wiederum zu einem erhöhten Regelaufwand um das Stromnetz innerhalb seiner Frequenzlimits stabil zu halten. Basierend auf den Vorteilen und Möglichkeiten von hydraulischen Maschinen können diverse Netzdienstleistungen zeitunabhängig zu Verfügung gestellt werden. Bezogen auf Francis Turbinen führt dies allerdings zu einer erhöhten Belastung infolge von Druckpulsationen, welche im Teillastbetrieb und bei transienten Vorgängen vermehrt auftreten.
Um die Betriebszeit zu verlängern und damit auch die Instandhaltungskosten zu senken sowie die Schadensrisiken besser einschätzen zu können, wird eine Lebensdauer Analyse benötigt.
Die gegenständliche Arbeit zielt auf eine Weiterentwicklung der numerischen Lebensdaueranalyse ab, da detaillierte Prototyp Messungen nicht für alle Maschinensätze durchführbar und meist mit erhöhten Kosten verbunden sind. Um die numerische Methodik zu validieren wurden umfangreiche Druck- und Schwingungsmessungen an eine Prototypanlage durchgeführt. Darüber hinaus wurden die Laufradspannungen mittels Dehnmessstreifen bestimmt, um eine umfangreiche Beurteilung des Verhaltens in tiefer Teillast zu ermöglichen. Weiters sind stationäre und instationäre CFD Analysen mittels der Software ANSYS-CFX und FEM Simulationen mit ANSYS-Mechanical durchgeführt worden. Die Strömung im kritischen stationären Lastpunkt wurde unter der Verwendung von hybriden RANS-LES Turbulenzmodellen analysiert. Die Untersuchungen ergaben, dass ein Teillastwirbelzopf der Grund für die erhöhten Druckpulsationen und damit Schädigungsraten ist. Außerdem war es möglich durch Anwendung von rechenintensiven CFD-Modellen und High Performance Computing (HPC), Vortex Shedding ausgehend von der Schaufel Hinterkante, welches zu einer Systemanregung führte, zu detektieren.
Zusätzlich zu den instationären Betrachtungen wurde auch ein transienter CFD-Ansatz, welcher am Beispiel des Lastabwurf Szenarios und des Startprozesses getestet und validiert wurde, entwickelt. Die Anwendung und Genauigkeit des numerischen Ansatzes wurde in allen betrachteten Fällen durch die Messdaten validiert.

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Nomenclature

Acronyms

Symbol	Description
BEP	Best-efficiency-point
CFD	Computational fluid dynamics
CEL	CFX expression language
DES	Detached eddy simulation
FEM	Finite element method
GSG	GreenStorageGrid
DES	Detached eddy simulation
HPC	high performance computing
LES	Large eddy simulation
$RANS$	Reynolds averaged Navier-Stokes equation
RSI	Rotor-stator interaction
SAS	Scale adaptive stimulation
$SBES$	Stress blended eddy stimulation
SNL	Speed-no-load
$URANS$	Unsteady RANS
$WMLES$	Wall-Modelled LES

Greek characters

Symbol	Description	Unit
α_0	Guide vane opening angle	rad
α_0'	Guide vane displacement angle	rad
Γ	Vortex strength	m^2/s
ν	Kinematic viscosity	m^2/s
ν_t	Turbulent eddy viscosity	m^2/s
ρ	Density	kg/m^3

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Latin characters

Symbol	Description	Unit
a	Vortex core radius	m
a_0	Guide vane opening	m
c	Absolute velocity component	m/s
c_m	Meridional component of c	m/s
c_u	Circumferential component of c	m/s
D_{ref}	Reference diameter of the runner	m
E	Specific energy	m^2/s^2
f_0	Rotational frequency of the runner	Hz
f_{BP}	Blade passing frequency	Hz
f_{GVP}	Guide vane passing frequency	Hz
f_s	SBES shielding function	$-$
$\boldsymbol{L}$	Position vector	m
$\boldsymbol{M_{rot}}$	Rotation matrix	rad
n	Rotational speed	$1/min$
n_q	Specific speed	$1/min$
n_{ED}	Speed factor	$-$
p	Static pressure	Pa
p_∞	Reference static pressure	Pa
p_{min}	Minimum static pressure	Pa
p_v	Vapor pressure	Pa
Q	Discharge	m^3/s
Q_{ED}	Discharge factor	$-$
r	Radius	m
u	Circumferential velocity	m/s
$\boldsymbol{u}$	Velocity field in 3D	m/s
w	Relative velocity	m/s
$\boldsymbol{x}$	Position vector in 3D	m
y^+	Non-dimensional wall distance	$-$
Z_{GV}	Number of guide vanes	$-$
Z_{RN}	Number of runner blades	$-$

Additional subscripts

Symbol	Description
1	Runner inlet
2	Runner outlet
i	First index of the Einstein summation convention
j	Second index of the Einstein summation convention

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1 Introduction

1.1 Research context

Today, the worlds primary energy usage is still based on 81% fossil fuels and 5% nuclear power. With biofuels and waste not taken into account, only 3% of the worlds primary energy is corresponding to renewable sources [1]. Considering the negative side effects such as climate change, air pollution and radioactive waste, it is a worldwide concern to facilitate renewable energies. Considering the negative side effects such as climate change, air pollution and radioactive waste, it is a worldwide concern to facilitate renewable energies. To cover the increasing electricity demand new renewable sources of energies, such as solar and wind, are highly supported and promoted. Therefore, the expansion rate of these new renewables is increasing continuously, as it can be seen in Figure 1.1.

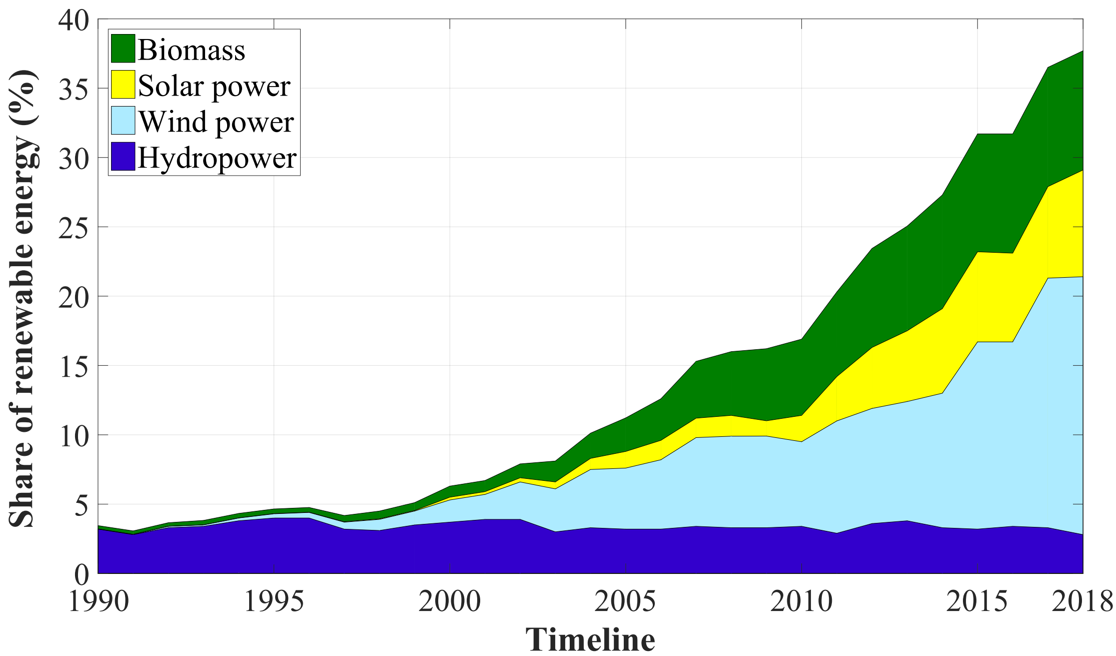

Figure 1.1: Share of renewables in Germany [2]

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However, one of the greatest downsides of the generation of electrical energy by solar and wind is their dependency on the weather and therefore unreliability. As a result, this leads to challenges of balancing electricity generation and consumption, which is essential for a stable power grid [3]. The allowed frequency deviations within the continental European power system are really small with a quasi-steady-state frequency limit of $\pm 200 mHz$ and maximum dynamic limit of $800 mHz$ [4] [5]. Therefore, besides long-term contracts and day-ahead as well as intra-day transactions, there is also the balancing energy market, which is concerned with that issue. Figure 1.2 shows the classification of balancing energy based on the response time. For the inertia there is currently no market, but may be in the future [4].

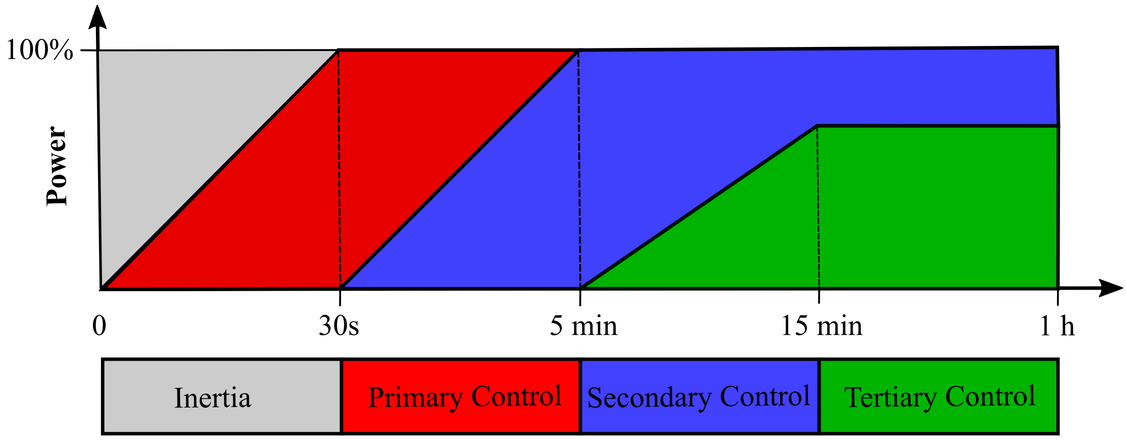

Figure 1.2: Specifications of balancing energy

Figure 1.3 (a) shows the monthly averaged prices of the EPEX-Spot market compared to the positive secondary control energy. One can see that in times of low spot market prices it is still economical beneficial to participate on the balancing energy market for operators of for example gas or hydro-power plants. Besides the existing market for primary, secondary and tertiary control power there are also the costs of congestion management better known as Redispatch costs. These costs are corresponding to a request issue by the transmission system operator to power plants, to adjust the real power output. This method to avoid congestion can be applied within or between control areas. Due to Germany's "Energiewende" and the trend of facilitating new renewables in Europe these costs have rised significantly, as is it can be seen in Figure 1.3 (b).

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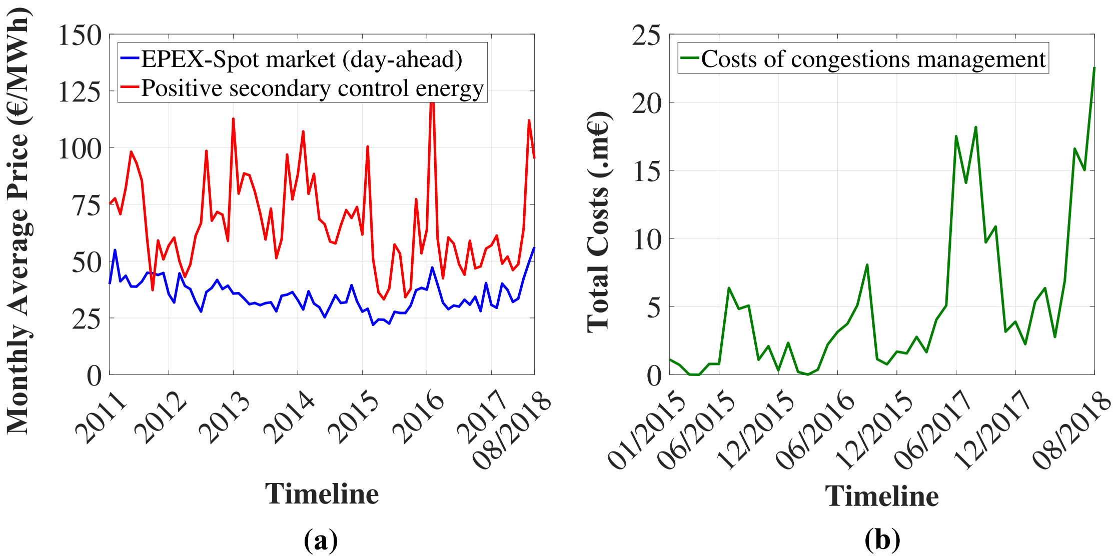

Figure 1.3: Prices on the spot market [6] as well as balancing energy [7] and Redispatch costs of the transmission grid operator APG [8]

Conventional Hydro-power, a technology which has been used for more than a century to generate electricity is nowadays sometimes left "out in the cold" by higher level political discussions. The geographical restrictions, the widely spread opinion that the potential is fully expanded and associated environmental impacts, were leading to a downward trend for this technology. However, with high efficiency, high reliability and a good regulating capacity hydro-power is superior for producing sustainable balancing power. Due to fast switching times of modern hydraulic power plants, they are capable of delivering tertiary and secondary as well as even primary control energy. Admittedly, older hydraulic turbines are designed to operate at their best-efficiency-point (BEP) and are not used to deal with high numbers of transient operations [9].

1.2 Francis turbines

The Francis turbine (see Figure 1.4) is the most common hydro-power turbine type, which is used for a head range between $50m$ up to approximately $700m$ and a power range of more than $800MW$ corresponding to a specific speed n_q in a range of $20min^1 \leq n_q \leq 120min^1$ [10]. The runner geometries vary from a more radial shape with narrow blade channels (lower n_q) to nearly axial shapes with large more complex blade channels (higher n_q). The turbine itself consists of spiral casing (1), stay vanes (2), guide vanes (3), runner (4), draft tube (5) and shaft (6). Based on the headwater pressure, an angular momentum is created by the spiral case, stay vanes and guide vanes and afterwards recovered by the runner. The stay vanes are responsible for mechanical strength of the spiral casing and equal distribution of water to the guide vanes. The adjustable guide vanes are the key part to control the flow rate and therefore the angular momentum recovered by the runner blades. The produced runner momentum is then converted into electrical power by a generator, which is connected to the turbine by the shaft. After leaving

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the turbine the water enters the elbow draft tube, where the cone decelerates the fluid to minimize the loss of kinetic energy.

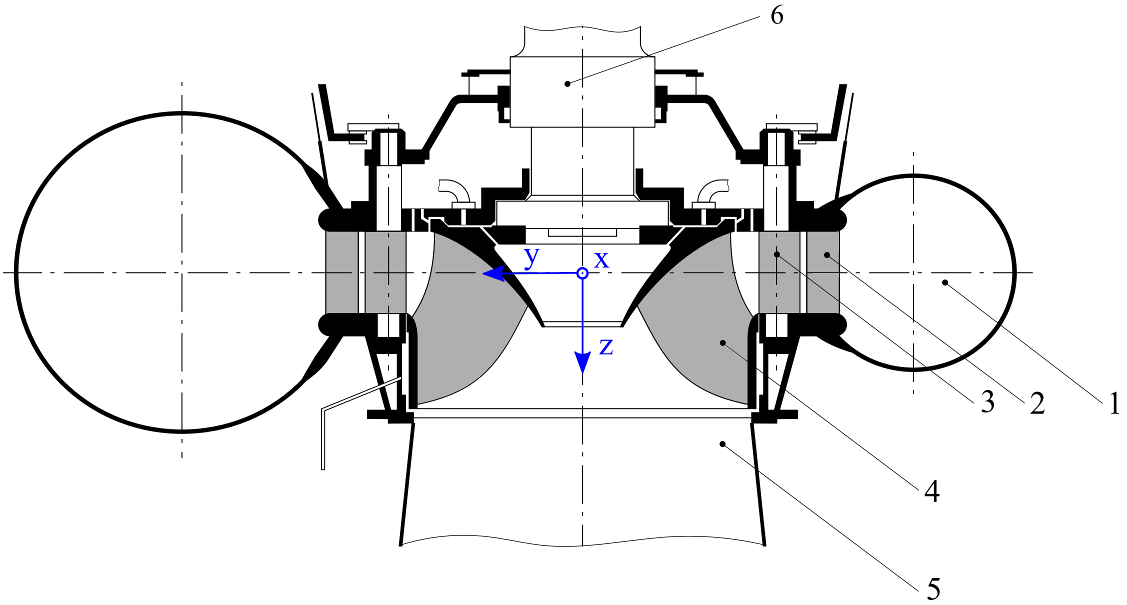

Figure 1.4: Meridional sketch of a typical Francis turbine

1.3 Operation regions

The operating conditions of Francis turbines are determined by discharge and rotational speed, which are defined according to IEC60193 [11] as non-dimensional parameters.

$$Q_{ED} = \frac{Q}{{D_{ref}}^2\sqrt{E}} \qquad n_{ED} = \frac{nD_{ref}}{\sqrt{E}} \tag{1.1}$$

In Equation 1.1, Q represents the discharge, D_{ref} the reference diameter of the runner, n the runner rotational speed, and E the specific energy. The variation of these parameters causes a change of runner inlet and outlet velocity triangles and therefore of the flow behavior (see Figure 1.5). The appearance of a circumferential component of the absolute velocity c_{u2} at the runner outlet leads to a secondary flow region. Based on the tangential velocity profile a Rankine Vortex distribution [12], as a combination of a free vortex and solid body rotation, is assumed [13]. The superposition of those models prevents the velocity from becoming infinite at the center of rotation. In its core, the vortex is characterized by constant vorticity and dominant viscous effects. However, in the outer region the flow is considered as inviscid and the model allows the velocity to decay at large distances. Based on the angular velocity $\omega = \Gamma/2\pi a^2$, the circumferential component of the absolute velocity of the runner outlet, is given by

$$c_{u2}(r) = \begin{cases} \dfrac{\Gamma}{2\pi a^2} r & r \leq a \\ \\ \dfrac{\Gamma}{2\pi r} & r > a \end{cases} \tag{1.2}$$

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By definition, r is the distance from the vortex axis and Γ vortex strength. The vortex core radius a is defined as the distance, where the circumferential velocity reaches its maximum. Figure 1.6 (a) shows the velocity distribution of the Rankine vortex.

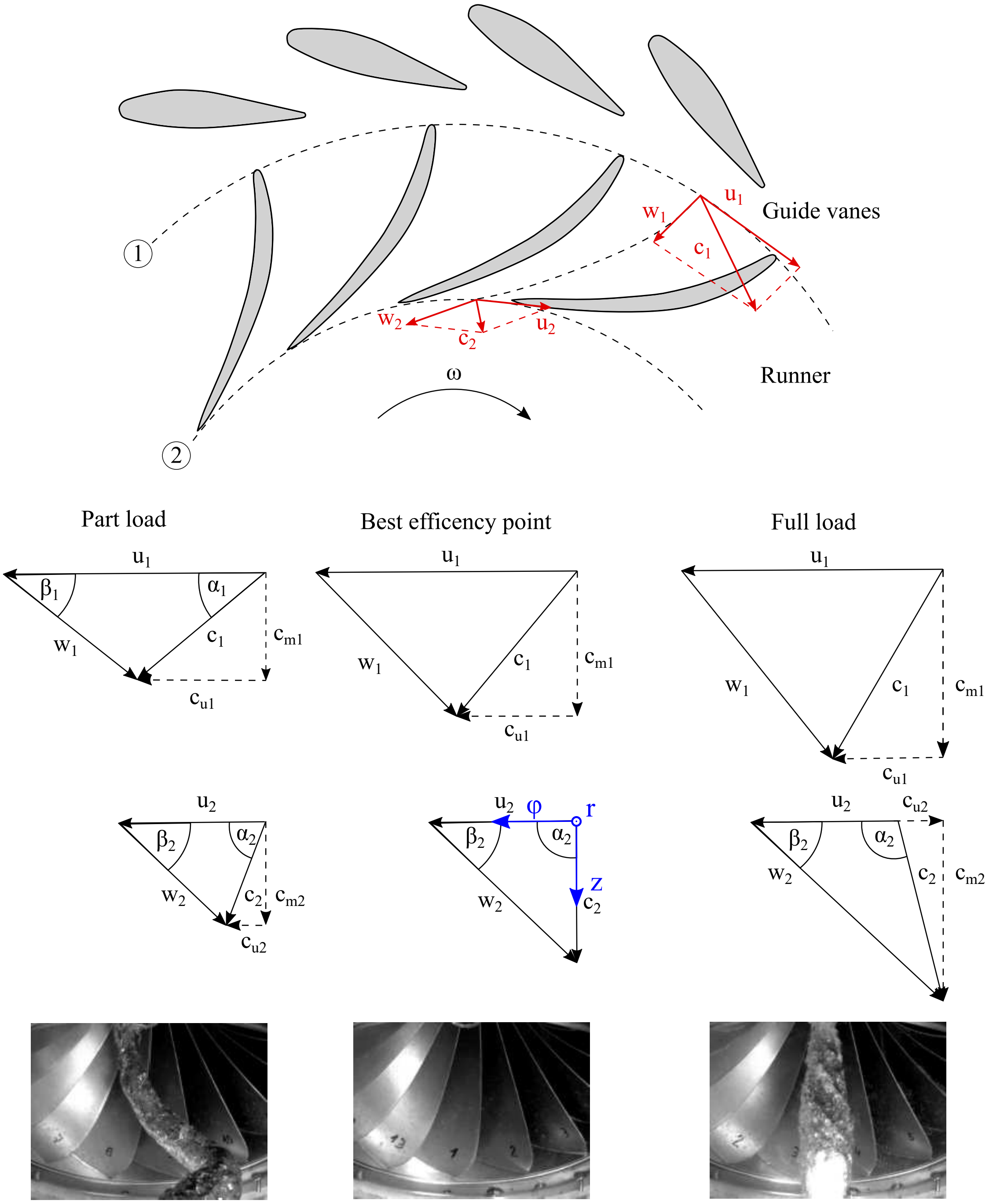

Figure 1.5: Velocity triangle and resulting flow patterns (adapted from [14]) at the runner outlet for operating at part load, the best efficiency point and full load

The radial equilibrium derived from the Navier-Stokes equations in cylindrical coordinates (Figure 1.5)

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$$\frac{\partial p}{\partial r} = \rho \frac{c_{u2}(r)^2}{r} \tag{1.3}$$

allows the computation of the radial pressure distribution $\partial p/\partial r$. Consequently, the radial pressure distribution

$$\int_0^a dp + \int_a^\infty dp = \rho \left(\frac{\Gamma}{2\pi}\right)^2 \int_0^a \frac{r}{a^4} dr + \int_a^\infty \frac{1}{r^3} dr \tag{1.4}$$

can be expressed by superposition. Therefore, $p(0)$ is the pressure minimum and can be expressed as

$$p_{min} = p_\infty - \rho \left(\frac{\Gamma}{2\pi a}\right)^2 \tag{1.5}$$

with p_∞ being the reference pressure, which in this case corresponds to the static wall pressure. The pressure distribution of the Rankine vortex is shown in Figure 1.6 (b). When the the pressure minimum in the vortex center drops below the vapor pressure p_v, cavitation occurs.

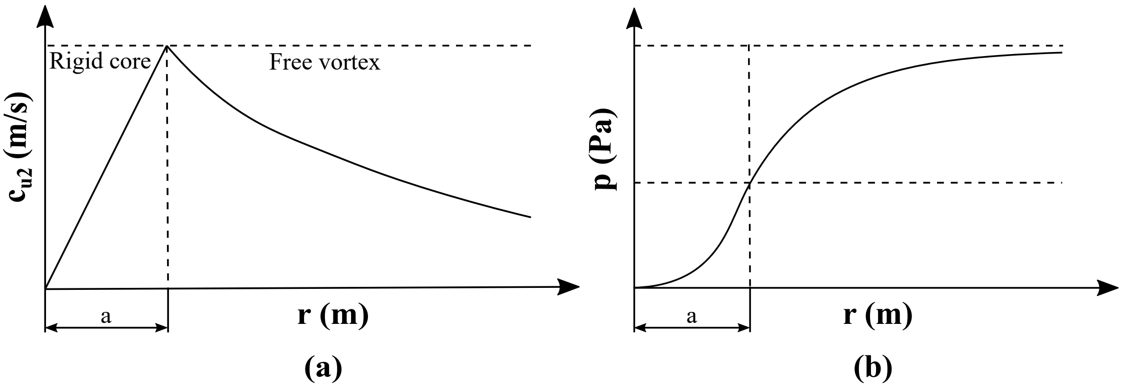

Figure 1.6: Schematic illustration of **(a)** the circumferential velocity c_{u2} and **(b)** the pressure distribution of the Rankine vortex

1.3.1 Part-load

At the part load region the c_{u2}, which points in the direction of the circumferential velocity u_2 leads to the appearance of a rotational secondary flow region in the draft tube. This phenomenon was first identified by Rheingans who determined the characteristic frequency of this effect normalized by the rotational frequency between $0.25 \cdot f_0$ and $0.35 \cdot f_0$ [15]. Later on, the part load vortex was the subject of many research projects, regarding hydraulic turbines, and the frequency range was extended to $0.2 \cdot f_0$ and $0.4 \cdot f_0$ [16]. The appearance of this rotating vortex structure leads to pressure pulsations, which are often noticed at the whole machine set. In 2010, Brekke [17] confirmed that the pressure amplitudes are being higher with higher n_q values.

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1.3.2 Best efficiency point

At the BEP the swirl at the turbine outlet is close to zero with a homogeneous velocity profile at the draft tube inlet. Under these conditions, there is hardly any secondary flow region occurring in the runner or draft tube. However, rotor-stator interaction (RSI) between guide vanes and runner as well as vice versa is still present. Especially, in case of high head Francis turbines with small gaps between guide vanes and runner, the harmonic excitation can induce high pressure pulsations. The kinetics of this phenomenon are explained by Tanaka [18], who discussed the interference of the blade cascade between runner and guide vanes. In the rotating system, the guide vane passing frequency, depending on the rotational frequency f_0, is determined by

$$f_{GV} = Z_{GV} \cdot f_0 \tag{1.6}$$

with Z_{GV} the number of guide vanes. Moreover, a blade passing frequency

$$f_{BP} = Z_{RN} \cdot f_0 \tag{1.7}$$

using Z_{RN} the number of runner blades, can be calculated in the stationary system. As this phenomenon mostly effects the runner inlet, the dynamic pressure oscillations at the outlet are relatively low [16].

1.3.3 Full Load

Above the BEP the full load or overload region is located, which is characterized by a c_{u2} pointing in the reversed direction of u_2 leading to the set-in of a counter rotating torch vortex (see Figure 1.5). The vortex oscillating frequency is found between $0.1 \cdot f_0$ and $0.25 \cdot f_0$ and the discharge is approximately 30% above the BEP [19] [14]. As reported by Flemming et. al [20] the synchronous pressure pulsations might propagate and interact with the hydraulic system. Moreover, as the guide vane blades are close to their maximum opening, the gap between guide vane and runner blades reaches its minimum, resulting in significant RSI effects.

1.3.4 Low load

Many publications dealing with the part load region suggest a separation between higher part load or simply called part load and deep part load, which is sometimes called low-load operation. Thereby the discharge value is further decreased and the swirling flow is expected to intensify due to the decrease of the meridional velocity component c_{m2}, while the circumferential velocity c_{u2} is increasing. This corresponds to the observations that besides the draft tube vortex, there are channel vortices (Figure 1.7) caused by a raising incident angle at the runner inlet and the resulting separation in the runner. As the guide vane opening a_0 becomes smaller, the flow tends to be more chaotic leading to back-flow regions and different vortices close or within the runner channels. Therefore, it is challenging to clearly identify their characteristic frequency. Yamamoto [21] has determined a frequency

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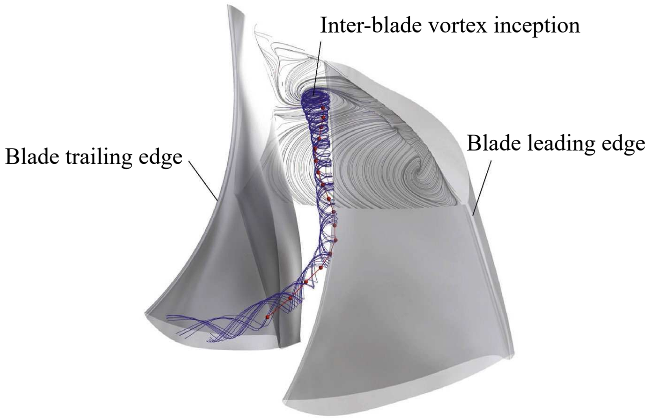
around $0.5 \cdot f_0$ by means of on-board pressure measurements. However, the numerical investigations of Conrad [22] showing a frequency range of $0.2 - 2 \cdot f_0$, which has to be considered if one aims to investigate this phenomenon. Yexiang et. al [23] mentioned the dependency on the actual operation conditions and defines a higher limit of $3 \cdot f_0$ based on the investigations on a model machine. Therefore, the difficulties to determine an actual frequency or even a frequency range are undeniable.

(a)



(b)

Figure 1.7: **(a)** Interblade cavitation vortices at low-load [24] and **(b)** structure highlighted by the flow streamline [25]

Besides draft tube swirl and inter-blade vortices there is also bubble cavitation. The phase change occurs when the local pressure of the water is below the vapor pressure depending on height of the tail-water level to the turbine setting level. As a result traveling bubbles (see Figure 1.8 (a)) or attached bubbles (see Figure 1.8 (b)) can be developed at off-design regions.
Another flow phenomenon which is typically occurring in Francis turbines at far off-design operating conditions, is the vortex shedding. These also called Kármán vortices are two counter-rotating rows of swirls, which alternately fall off at the tail of the nonlinear object when fluid flows past [26]. Thereby, each vortex is initiated at the separation point between the attached flow and the stalled downstream region. In hydraulic machines, this phenomenon can occur at the stay vanes, guide vanes or sometimes at the runner blades. In most cases, the effect can be controlled or at least weakened by modifying the trailing edge shape [27].

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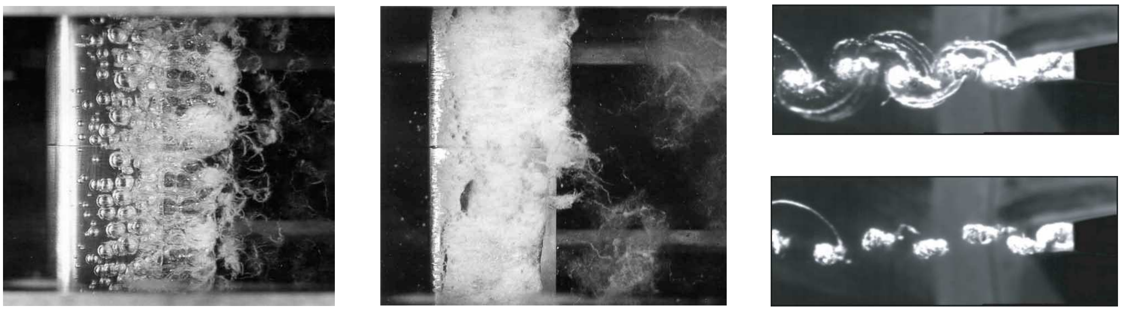

(a) (b) (c)

Figure 1.8: **(a)** Traveling bubble cavitation and **(b)** sheet cavitation on the suction surface of a NACA 4412 hydrofoil [28] as well as **(c)** cavitation vortex street resulting of a blunt trailing edge NACA 0009 hydrofoil in free stream [27]

1.3.5 Speed-no-load

The region where the turbine operates at synchronous speed without electricity production is called Speed-no-load (SNL). At this state the guide vanes are almost closed and the flow behavior is similar to low-load operation and more or less stochastic [29]. The turbine requires just enough torque to equilibrate the mechanical losses. The machine set is sometimes in standby mode for several hours, with the intention to provide immediate power reserve to the electrical grid. Based on the turbine design, high pressure pulsation can occur during this operation [29].

1.3.6 Synchronous condenser mode

The condenser-mode-operation (COM) allows the hydro-power plant to either generate or absorb reactive power. The hydraulic machine is connected to the electrical grid, while the turbine is rotating in dewatered condition. This state of operation is maintained by pressurized air injection in the draft tube to keep the water level below the runner. The operation in this dewatered condition is known to be challenging regarding torque and related power swings as well as complex 2-phase flow conditions like free-surface sloshing motion in the draft tube. The sloshing motion is qualitatively described as a rotating wave with a frequency mainly depending on the rotational speed [30]. Moreover, due to the interaction of the cooling water of the labyrinth seal and the water droplets based on the oscillation of the air-water interface entrained by the runner, a rotating air-water ring in the vaneless space is developed [31].

1.3.7 Transient operation

Besides operation in stationary load points (n_{ED} and Q_{ED} remain constant) there are also transient events like

- Startup
- Shutdown

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- Load rejection

which are assigned to off-design operation. They are a key element to continuously maintain a certain level of voltage and frequency of the electrical grid. In However, during transient operation, the runner speed, head, discharge and torque can vary with time. In general, these events are the most severe operating conditions of a Francis turbine. When the machine starts, the guide vane is opening until the point where the torque is enough to accelerate the shaft to the synchronous speed and to overcome the mechanical losses. At this point in time, the machine can be synchronized with the electrical circuit and the turbine is operating at speed-no-load without electrical power-output. The startup procedure and the control parameters are described in the IEC 61362 [32]. The IEC recommends a synchronization speed within a 99.5% – 101% range of the nominal speed (Figure 1.9).

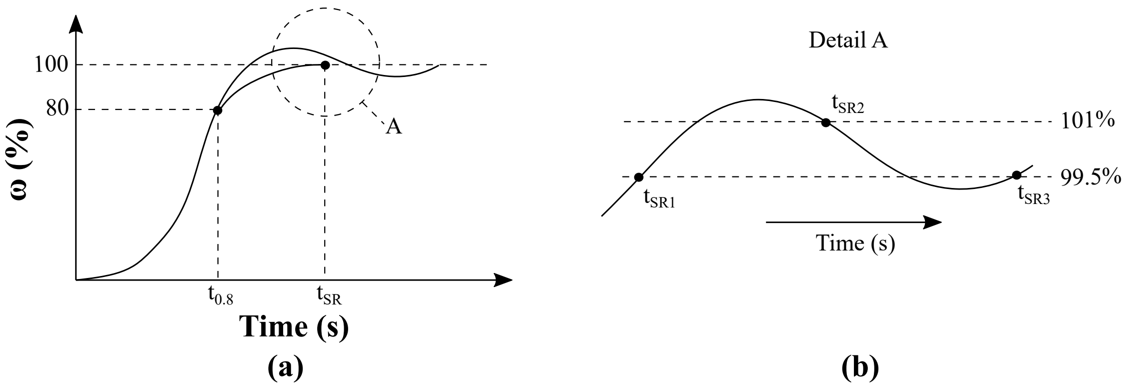

Figure 1.9: **(a)** General overview of the Startup process and **(b)** detailed view of the syncronisation boundaries as defined in the IEC 61362 [32]

The opposite of the startup procedure is obviously the shutdown of the power plant. Thereby, the load is smoothly removed by closing the guide vanes until the turbine can be disconnected from the electrical grid. Than the mechanical brakes of the generator decelerate the rotating parts of the machine. An exceptional case is the load rejection, where the turbine is suddenly disconnected from the electrical grid. Therefore, the turbine torque is no longer equilibrated by the generator and the runner starts to accelerate rapidly. Without an emergency closing procedure, the machine would be accelerated to approximately 1.5 up to 3 times the nominal rotational speed, depending on the design. However, normally the guide vanes and the throttle valve are rapidly closing so that only values between 1.15 up to 1.4 are reached. The closing speed is limited by pressure oscillations, induced by the resulting water hammer in the hydraulic circuit. Again, the runner has to deal with a rapidly decreasing inflow angle, which causes massive flow separation. During this transient process, flow blockage can lead to an increase in the incidence angle for the follower blade and a decrease in the incidence angle for the forerunner blade. This so called rotating stall effect is rarely occurring in Francis turbines, but often observed in a pump-turbine [33] [34].

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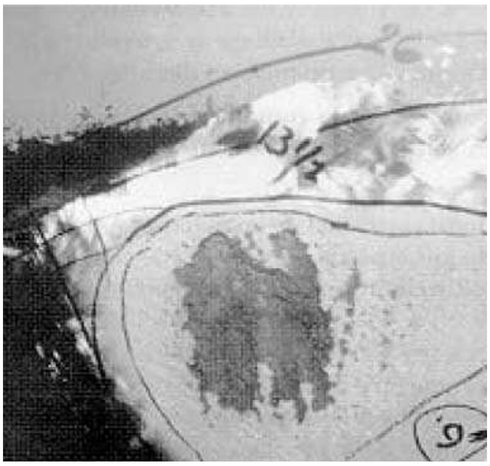
1.4 Fatigue strength and risk of off-design operation

The occurrence of unsteady high fluctuating pressure fields in hydraulic machines can lead to high dynamic mechanical stresses and excitation of the entire machine set. This leads to a safety risk due to high vibrations and material fatigue, which cause structural cracks. Over time several cases corresponding to different flow phenomena have been reported. One of the first accidents was described by Goldwag and Berry [35] in 1968. The paper revealed that vortex-shedding led to numerous and serious fatigue cracks of the stay vanes. Furthermore, Bhave et. al [36] reported blade failures of Francis turbines in 1987 while later on Coutu et al. [37] showed the risk of long-term operation under low-load conditions. The famous Sayano–Shushenskaya power station accident occurred on the 17th August 2009 in Siberia (Russia). An older Francis turbine was used for demanding plant operation and suffered under continuously increasing vibrations [38]. Consequently, the weakest link, in this case the head cover bolts, broke [17]. Although poor management and service may be responsible for failure, this case punctuates the risk of low-load operation. Figure 1.10 shows the effects of operation at off design conditions on a prototype Francis turbine runner.

(a)



(b)

Figure 1.10: **(a)** Typical fatigue crack at the runner blade trailing edge near to the shroud [14] and **(b)** erosion at the wall of the blade suction side due to inlet edge cavitation [39]

1.5 State-of-the-art

1.5.1 Prototype site measurements

Nowadays, it is well-known that e.g. different Froude-numbers, deviations regarding the runner side clearance and the impact of the surface roughness are causing differences between prototypes and their models [40]. Therefore, detailed prototype measurements are a key element to precisely evaluate the impact of the flexible

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operation on a hydro-power plant operating at off-design conditions [41]. In 1966, Oftebro et. al [42] published investigations on pressure oscillations corresponding to the interaction of the runner and stationary parts of Francis turbines. Later on, Dorfler et. al [43] presented detailed prototype site measurements by the use of wall pressure transducers in the draft tube and vibration sensors. Based on the changed operation requirements of Francis turbines, dynamical stress predictions are significantly more important than in the past. Therefore, Fischer et al. [44], Bjørndal et al. [45] and Gagné [46] were using strain-gauges to indirectly measure the mechanical stresses on a prototype runner. Since the hydraulic and mechanical designs of prototype machines differ significantly, it is difficult to extrapolate the measurement data for similar machines.
Although in some cases the draft tube vortex may not have a significant effect on the runner, it can also affect or damage other components such as the draft tube. As a result, further prototype site measurements such as those published by Gagnon et. al [47], Seidel et. al [48] and Huang et. al [49] were necessary to develop a better understanding of dynamical runner stresses in order to access fatigue life. However, in 1982 Pople [50] observed more than 70 sources for uncertainties, regarding strain gauge measurements. Moreover, Montero et. al [51] pointed out that the operating temperature, the measurement device including the non-linear Wheatstone bridge and the proper positioning of the strain gauge are the most important parameters. This was picked up again by Arpin-Pont et al. [52], who was concerned with the problems of a misalignment error regarding strain gauge measurements at Francis turbines. Recent research has aimed to improve some of these parameters, as for example Lofflad et. al [53], who published a more complex approach using telemetry for data acquisition and special protraction layers for the strain gauges.

1.5.2 CFD in hydraulic machines

Due the complexity of 3D flow in hydraulic turbines, the first publications on numerical investigations were published in the later half of the 1980's. Over the past decades, the increase in computational capacity and advancements in numerical techniques have allowed to simulate hydraulic turbines at different operating conditions. Nevertheless, the simulation at off-design conditions and above all transient events are challenging due to the complex flow physics and high simulation times.
Since it is well known that unsteady Reynolds-Averaged Navier-Stokes equation models (URANS) are not adequate for solving all dynamic fluid flow problems, the complex way of using a higher resolving large eddy simulation (LES) model have became more and more established. For instance, Spalart [54] mentioned that URANS models clearly fail to reproduce vortex shedding around a cylinder. Considering the complex flow conditions during transient events and low-load operation, Magnoli [14] and Eichhorn [55] recommend the use of hybrid models. Thereby, the model switches between the faster URANS formulation, where the turbulent eddies are not resolved to the level of the smallest eddies, and the LES behavior. The most common hybrid models which are used in the field of numerical simulations of hydro-power plants are the detached eddy simulation (DES) first

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introduced by Spalart [54] and the Scale adaptive simulation (SAS) by Menter et. al [56]. A comparison on two benchmark flow cases leading to similar results is published by Zheng et. al [57]. Moreover, Wunderer [58] mentioned that the SAS model produced similar results in a turbine blade cascade. Jošt et. al [59] and Krappel et. al [60] concluded that the application of the SAS improved the results, in particular with regard to pressure amplitudes and frequency.
In contrast to steady or unsteady simulations in stationary load points, transient simulations include varying boundary conditions like for instance discharge, runner speed. Additionally, to model transient events it is required to consider moving walls like runner or guide vane blades. One of the first attempts of a transient CFD analysis was published by Kolšek et. al [61], who simulated the shutdown process of a bulb turbine. Later on, different studies dealing with load rejection conditions [62] [63] [64] [65] were published. In 2012 Nicolle et. al [66] [67] published a first approach to simulate the startup of a Francis turbine. Moreover, Casartelli et. al [68] published a study dealing with the transient turbine startup process based on 1D hydraulic system simulation. Most of these studies used a standard k–ϵ model and reduced domains to increase numerical stability and decrease simulation time. However, this leads to a worse resolve of especially smaller scaled flow phenomena. Therefore, some recent studies by Trivedi et. al [69] or Melot et. al [70] recommend the SAS hybrid turbulence model to achieve a better accuracy.

1.5.3 Fatigue analysis

The recent requirements of the energy market are resulting in the future challenge of a accurate prediction of fatigue for the hydraulic components of hydro-power plants. Therefore, detailed investigations targeting the turbine runner as one of the most critical component, driven by manufacturers and operators will be necessary as an proper response. Especially in the design phase no expensive measurement data is available. As a result, numerical computations become more and more important. The state of the art industrial design processes is shown in Figure 1.11. The process of obtaining the mean stress by using numerical methods is well established [71]. A steady CFD analysis is performed and the obtained pressure field is used as boundary condition for the FEM simulations as it is presented by Sick et al. [72]. In addition, in some cases the application of harmonic response FEM computations is preferred to evaluate appearing dynamic stress amplitudes at a determined frequency.
In recent years, many studies dealing with the possibility of obtaining time-dependent stress signals by using a unsteady CFD analysis combined with a transient FEM calculation, are published. Compared to the harmonic response analysis, this approach considers a fully resolved signal, which does not depend on deterministic behavior and therefore also considers stochastic peaks. For example, Morissette et al. [73] mentioned that a time-dependent FEM approach would provide more detailed and accurate results, although the computational effort is increasing by far. Some approaches are already published by Doujak and Eichhorn [74] [55] and Nennemann et al. [29] and efforts to validate numerical calculations with measured data are taken (e.g. [75] [76]).

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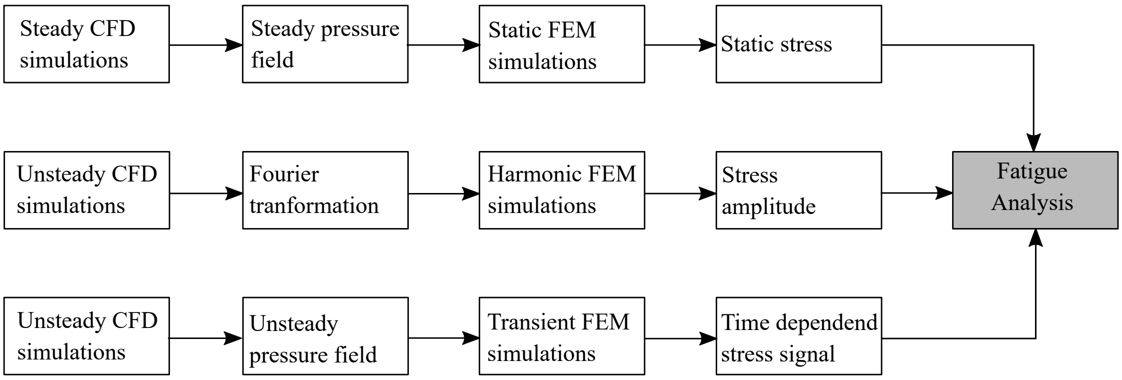

Figure 1.11: Numerical approach for the fatigue analysis of Francis turbines state-of-the-art (line 1 and 2) and approach proposed by Eichhorn and Doujak [55] [74] (line 3)

To assess fatigue life, most of the studies are using a simplified approach according to the Miner rule [77]. As recommended in the FKM guideline [78], the experimental determined S-N curve serves as the limit for failure. However, Huth [79] and later on Gagnon et al. [80][81] [82] were proposing a more complex approach considering crack growth by the Linear Elastic Fracture Mechanics theory. Thibault et al. [83] [84] mentioned the complexity and influence of the the metallurgical aspects and microstructural features.

1.5.4 Structural response

Dynamic load can also lead to altered vibration behavior of the machine if the extinction frequency corresponds to a natural frequency of a component. Therefore, a numerical modal analysis is included in a comprehensive life cycle analysis, as it is depicted in Figure 1.11. In 1987, Dubas and Schuch [85] presented a numerical approach using isoparametric shell elements to determine the natural frequencies and mode shapes of a Francis turbine runner. More recently, Tanaka [18] mentioned that natural frequencies are reduced by the water volume and suggested to take the added mass effect into account. Consequently, Flores et. al [86], Liang et. al [87] and Sick et. al [72] published investigations dealing with the influence of the added mass effect on the natural frequency. Most recent investigations of Østby et. al [88] determine a frequency reduction of around 35% up to 45% for the main blade modes increasing with the nodal diameter. Moreover, experimental validation under operating conditions as the one published by Valentin [89], confirmed reduction values around 50%. Nowadays, it is a key element of the design process to compare the numerical calculated natural frequencies to the results of the harmonic analysis to determine whether or not mechanical resonance is occurring [87].

1.6 Thesis objective

Based on the results of the GreenStorageGrid (GSG) project (FFG project number 836636) a further improvement of fatigue analysis of prototype Francis turbines is

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suggested [55]. During the course of the GSG project, the author was already working on the evaluation of measured data and published the experimental approach for the lifetime assessment of two Francis turbines with different n_q [90]. The follow-up project MDREST (FFG project number 11364358) is meant to further develop the lifetime determination method. A key target of this project is the integration of vibration sensors to assess the behavior of prototype machines operating at off-design conditions. The schema for lifetime assessment, with the state-of-the-art process colored in black, is depicted in Figure 1.12. The modal analysis of the runner including a consideration of the added mass effect, is carried out by a diploma thesis [91].

As part of the research project, the main objective of this thesis work is to contribute an improvement of the numerical lifetime analysis. Therefore, a new approach to model transient events (colored in red), is developed. In contrast to unsteady simulations of Francis turbines, this approach takes into account varying boundary conditions and the integration of a guide vane movement procedure. Moreover, parts of the scheme that have been improved according to the state-of-the-art process are represented by blue color.

In order to validate the numerical simulations detailed prototype site measurements are also a key part of the investigations. In contrast to previous investigations one goal of this thesis is to improve the strain gauge placement and consider different sensor positions by preliminary simulations. Pressure transducers and vibration sensors are mounted to determine damaging operating regions and gather data for the validation of the numerical method. Most of the measurement data evaluation and cross-correlation is still subject to ongoing diploma and bachelor theses and is not part of this work.

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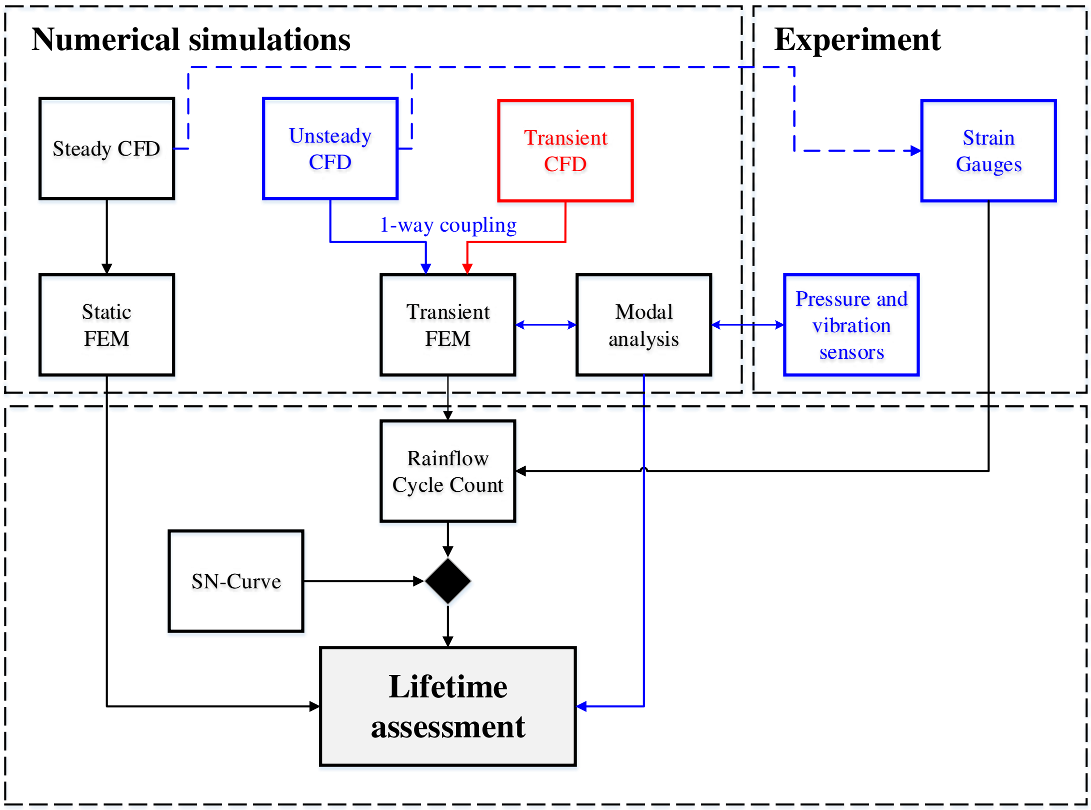

Figure 1.12: Improved schema for the lifetime assessment of prototype Francis turbine runner

To improve the accuracy of the obtained pressure fields two-phase flow and higher resolved meshes are considered. Moreover, the stress blended eddy simulation (SBES) turbulence concept (see Appendix A) is tested. The SBES model decouples the turbulence modeling from the detection of RANS and LES zones [92] and therefore offers advances to detect flow phenomena by means of CFD simulations. Due to the novelty of the model an assessment of applicability and comprehensive validation, especially regarding the simulation of hydraulic turbomachines is currently not existent. Therefore, it is an objective of this thesis to compare the SBES to the SAS model to evaluate advantages when assessing unsteady pressure pulsations.
Furthermore, the research seeks to compare results of the modal analysis with the CFD analysis and the transient FEM to discover excitation mechanisms. In this context, the effect of air injection on runner lifetime and machine vibrations is evaluated by means of experiments and simulations. The discussed research challenge is addressed with four different but coherent contributions to relevant conferences and international journals.

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1.7 Thesis structure

The present thesis work is a collection of peer-reviewed manuscripts published or submitted to international journals.

In **Chapter 1**, a brief introduction about the topic of the thesis is given. The roll of the Francis turbine in Europe's modern energy system is discussed and theoretical background as well as the thesis objective is explained.

In **Chapter 2**, the prototype hydro-power plant and the operating conditions at which the tests are performed are illustrated. Moreover, Paper I includes a brief description about the strain gauge measurement as well as preliminary simulations and a discussion of the experimental results. The runner stresses are converted to a load spectrum by means of a rainflow cycle count and used for a lifetime assessment based on experimental data. Therefore, this paper evaluates and compares all stationary operation points and transient events. In addition, Paper I points out that operation around $44\% \cdot P_{RP}$ as well as transient operation and in particular the startup contribute most to the overall damage. At the end of the paper, a comparison between typical base load and demanding plant operation for grid stabilization is presented. Thus, this paper provides a good basis for the following numerical investigations, highlighting critical operation zones and events. The presented experimental data are further used to validate the numerical simulations.

In **Chapter 3**, the numerical path to the runner lifetime is presented. The mesh of the domains is described in detail and a grid independence study is performed. Moreover, the obtained global parameters like machine head, efficiency and torque are compared to experimental results. The main part of Paper II deals with the critical operation region and discovers damaging flow phenomena by means of a CFD analysis referred to Paper I. Besides steady and unsteady CFD simulations a new transient approach is presented and tested by means of a load rejection scenario. Finally, the numerical results for both cases are compared to the experimental data. In both cases, the dynamic stress and the pressure pulsations in the draft tube cone are below the amplitudes obtained in the experiment. Therefore, article suggests a improvement of CFD-method by considering two-phase flow as well as a further improvement and testing of the transient approach. Moreover, a more detailed investigation of the critical operating point is recommended.

In **Chapter 4**, a detailed investigation of the critical operating point by means of vibration sensors and advanced CFD technologies is presented. The experiments show high vibrations, which are not corresponding to the draft tube vortex discussed in Paper II. Moreover, those vibrations are damped and omitted by the use of an air admission system. Therefore, Paper III deals with the influence of air admission to damp vibrations by the use of numerical simulation and experimental tests. Due to the application of advanced turbulence modeling with higher amount of cells and considering a multiphase model, the use of HPC is required. Besides the higher resolution of the draft tube vortex, which is discussed in Paper II, Paper III also discovers the occurrence of trailing edge vortex-shedding by means of the

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new SBES turbulence model. A simplified modal analysis confirmed the assumption that the vortex-shedding leads to excitation of the draft tube. Furthermore the FSI mapping algorithm and the addition of the normal projection vector is explained in detail. Finally, it can be concluded that the air injection reduces the dynamical stresses and can successfully impede the development or at least shift the detachment frequency of the vortex-shedding.

In **Chapter 5**, the transient approach which is presented in Paper II is improved and used to simulate the startup procedure, since it is the overall most damaging operating mode referred to Paper I. Paper IV compares two different startup schemes to each other and discusses unsteady flow phenomena by means of experimental and numerical methods. Moreover, the transient approach which is presented in Paper II is slightly improved and further validated. The experimental study shows that different startup schemes lead to different damage factors for the whole event. Therefore, a CFD analysis is performed to shed light on the damaging flow phenomena occurring during startup. The CFD analysis reveals a counter rotating draft vortex structure, which induces high pressure pulsations on the runner blade.

In **Chapter 6** a summary of the perspectives, scientific challenges and suggestions for further work based on Paper I – IV is given.

Finally, **Chapter 7** contains a summary of the whole work.

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2

Paper I

2.1 Fatigue Analysis of a Prototype Francis-Turbine based on strain gauge measurements

Reproduced version of:

Unterluggauer, J., Doujak, E., Bauer, C. (2019), *Fatigue Analysis of a Prototype Francis-Turbine based on strain gauge measurements*, WasserWirtschaft Extra; S. 25 - 30

The author's contribution:
The author performed the preliminary simulations in order to evaluate the strain gauge positions. He was in charge of the installation process and did the data post-processing as well as the fatigue analysis. Therefore, he is the first author of this publication.

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Paper II

3.1 Numerical Fatigue Analysis of a Prototype Francis Turbine Runner in Low-Load Operation

Reproduced version of:

Unterluggauer, J., Doujak, E., Bauer, C., (2019), *Numerical Fatigue Analysis of a Prototype Francis Turbine Runner in Low-Load Operation*, Int. J. Turbomach. Propuls. Power, 4(3), 21; doi: 10.3390/ijtpp4030021

The author's contribution:
The author did the meshing including the grid independence study and performed the CFD simulations. He coded an extension for the in [55] used mapping algorithm and performed the FEM simulations. Moreover, the author post-processed the measured data and validated the simulation results by comparing them. Therefore, he is the first author of this publication.

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4 Paper III

4.1 Investigation on the Impact of Air Admission in a Prototype Francis Turbine at Low-Load Operation

Reproduced version of:

Unterluggauer, J., Maly, A., Doujak, E. (2019), *Investigation on the Impact of Air Admission in a Prototype Francis Turbine at Low-Load Operation*, Energies, 12(15), 2893; https://doi.org/10.3390/en12152893

The author's contribution:
The author developed the concept, did the meshing and performed the CFD simulations. He evaluated the measured data and validated the simulation results. Therefore, he is the first author of this publication.

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Paper IV

5.1 Experimental and Numerical Study of a Prototype Francis Turbine Startup

Reproduced version of:

Unterluggauer, J., Doujak, E., Bauer, C. (2019), *Experimental and Numerical Study of a Prototype Francis Turbine Startup*, In Proceedings of the 2th IAHR Asia Symposium on Hydraulic Machinery and Systems, 24th-25th September, 2019, Busan, Korea

The author's contribution:
The author performed the data analysis and was in charge of the strain gauge installation process. Moreover, he designed the second startup procedure and compared it to the regular schema and the startup of two prototype machines that he had previously examined. Furthermore the author performed the CFD analysis, evaluation and validation by means of measurement data. Therefore, he is the first author of this publication.

Remark:
The paper is selected to apply for the further reviewing process of the Elsevier-Journal Renewable Energy special issue: IAHR-Asia 2019. If the paper is finally accepted, only the abstract will be included in the conference proceedings in order to prevent duplicate copies of the paper.

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6 Perspectives and Outlook

The developments achieved in this thesis can serve as base for additional advances in the lifetime assessment of Francis turbine runners. However, the presented work leaves room for further investigations and ongoing work:

- First of all, the presented transient approach needs to be further tested. The validation is done on a prototype machine with limitations in terms of measurement quantities due to inaccessibility. Therefore, it would certainly be advantageous to take a step back and use a model machine with significantly more possibilities to validate the flow field during the process. For example, high frequency pressure measurements in the vaneless space would be an excellent validation parameter.

- Although the numerical approach was successfully validated in this study, further prototype measurements for different hydro power plants are suggested. Especially, the application on pump-turbines or even Pelton runners are possible. This would also include to simulate the changeover process between pump- and turbine-mode.

- Moreover, to decouple the boundary conditions from the prototype measurement the implementation of a 1D approach representing the waterway is suggested. Thereby, based on the transient CFD approach a virtual power plant, which can be operated by the user is created. An integration of the damage factors and operation behavior would definitely be beneficial for an optimization with regard to economic targets.

- In this work, a combination of mesh displacement and remeshing was used to simulate the guide vane movement. It is suggested, to compare different mesh deformations strategies based on the quality the produce for small guide vane openings.

- One of the greatest weaknesses of the lifetime assessment approach discussed in this thesis, which uses S-N curves combined with the Miner's damage accumulation rule, is that the model has only two states: undamaged or damaged. However, to define inspection intervals, a damage tolerance approach is often more appropriate. Therefore, the development of a crack and the variable grow rate with a limiting size, above which repairs are needed, has to be taken into account.

- In order to optimize the startup procedure towards fatigue life, further research is needed. Paper IV discussed to different startup procedures and reveals a counter rotating vortex structure, as main source of the high dynamical stresses. Further research by means of different prototype and model machines are suggested to investigate this phenomenon. Moreover, different solutions to decrease the dynamical stresses during startup, like for instance particular air injection, is suggested.

- Finally, to cover all transient events of a prototype Francis turbine one has to analyze the condenser-mode-operation (CMO) including the dewatering process. However, to evaluate the complex dynamic characteristics of the free-surface and flow phenomena occurring during CMO, bridging the gap between mathematical modeling and applied fluid mechanics will be required.

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7 Conclusion

The main focus of this work has been to further develop a numerical approach for the fatigue assignment of prototype Francis turbines. Therefore, the behavior at different operating regions of a specific prototype machine is analyzed by detailed site measurements and numerical methods. Besides pressure and vibration measurements, strain gauges are used to indirectly determine runner stress. In order to guarantee a proper placement at the stress hotspots, preliminary CFD and FEM simulations are performed. The results revealed the highest mean values at full load operation. However, the highest dynamical values are observed at the low load region ($44\% \cdot P_{RP}$) and overall transient events. Moreover, the measurement reveal extraordinary high vibrations at the most damaging low load operating point. To asses impact on the structural impairment and fatigue strength the accumulation rule of Palmgren and Miner, considering the S-N curve of the runner material, is used.

The numerical CFD approach is validated by global parameters, such as head power and efficiency as well as static pressure measurements. Furthermore, a grid convergence study is performed to calculate the numerical error and ensure reasonable a sufficient accuracy. The obtained pressure fields are used as boundary conditions for the FEM calculations, which are validated by the strain gauge measurement. The numerical investigation of the critical low-load operating point discovered a draft tube vortex with a rotating frequency of $0.2 \cdot f_0$. This flow phenomenon corresponds to the high dynamical stresses in the runner blade, but is not the reason for the excitation of the machine set, which is discussed later on. Moreover, the first approach showed that the wall pressure pulsations in the draft tube cone and thus also the induced runner stresses are underestimated, even if the SAS hybrid turbulence model is used. Therefore, further simulations with higher amounts of grid cells and considering two-phase flow are performed.

As turbulence modeling can have a major impact on the flow behavior besides the SAS model the newer SBES model was tested. The compartment showed similar results and the pressure pulsations in the draft tube are now over predicted related to the two-phase approach. However, a refined runner mesh and the use of the SBES turbulence model revealed the occurrence of a trailing edge vortex shedding, corresponding to a natural frequency of the draft tube determined by

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a modal analysis. Therefore, as already mentioned above, the whole machine set is suffering under high vibrations. The operator deals with this issue by injecting air in the vaneless space between guide vanes and runner, leading to a sufficient reduction of the vibration.
Consequently, the numerical simulations reveals that the air gathers around the hub and shroud of the blade and thus impedes the formation of the vortex shedding. Additionally, the emergence of the draft tube vortex is not particular influenced, but the induced dynamical runner stresses are reduced by damping effects of the air. Therefore, a further development of the numerical approach for stationary operating points is archived by improving the CFD setup.
Furthermore, besides the consideration of operating points a transient approach was developed and validated. This approach is sufficiently capable to simulate the flow behavior during transient events and is used to study load rejection and startup of the prototype Francis turbine. The CFD simulation of two different startup processes revealed the appearance of a counter rotating vortex formation, corresponding to extraordinary high dynamical runner stress. Therefore, this work contributes an improvement of the existing methodology for numerical lifetime assessment of Francis turbine, including transient events.

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APPENDIX A

A.1 SBES Turbulence Model

In recent years, many versions of the original DES formulation targeting on the weakness in terms of the impact of LES components on the boundary layer, have been published. Therefore, Wall-Modelled LES (WMLES) [93] is developed to ensure that the energetic scales of turbulence in the innermost $10-20\%$ of the boundary layer are averaged and treated like standard RANS regions. One of the newest formulation is the SBES [92], an approach which combines strong shielding and protection against the LES application in the boundary layer as well as WMLES. The model provides rapid and distinctive RANS-LES transition. The model can be explained based on the Navier-Stokes equations for continuity and momentum written in Einstein notation

$$\frac{\partial \overline{\boldsymbol{u_i}}}{\partial \boldsymbol{x_i}} \tag{A.1}$$

$$\frac{\partial \overline{\boldsymbol{u_i}}}{\partial t} + \overline{\boldsymbol{u_j}}\frac{\partial \overline{\boldsymbol{u_i}}}{\partial \boldsymbol{x_j}} = \frac{1}{\rho}\frac{\partial \overline{p}}{\partial \boldsymbol{x_i}} + \frac{\partial}{\partial \boldsymbol{x_j}}\left[(\nu + \nu_t)\frac{\partial \overline{\boldsymbol{u_i}}}{\partial \boldsymbol{x_j}}\right] \tag{A.2}$$

where $\overline{u_i}$ and $\overline{p}$ are velocity and pressure fields, ρ the density, ν the kinematic viscosity, ν_t the turbulent eddy viscosity and x_i the length scale vector. According to the Boussinesq hypothesis [94], ν_t approximates the LES or RANS stress tensor $\boldsymbol{\tau_{ij}}$, which represents the non-resolved turbulent momentum transfer [95]. Consequently, the blending or shielding function f_s of the SBES model based on the stress-level

$$\boldsymbol{\tau_{ij}^{SBES}} = \boldsymbol{\tau_{ij}^{RANS}} f_s + \boldsymbol{\tau_{ij}^{LES}}(1-f_s) \qquad 0 \leq f_s \leq 1 \tag{A.3}$$

determines regions where RANS or LES turbulence models are applied. If both models are eddy-viscosity models, the equation can be written in its incompressible form

$$\nu_t^{SBES} = \nu_t^{RANS} f_s + \nu_t^{LES}(1-f_s) \qquad 0 \leq f_s \leq 1 \tag{A.4}$$

The blending function f_s is given by

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$$f_s = \left[e^{\frac{1}{6}\cdot(y^+ - 45)} + 1 \right]^{-\frac{3}{25}} \tag{A.5}$$

and depends on the non-dimensional wall distance y^+. The remaining parameters were determined empirically based on a pipe flow test case with cyclic boundary conditions. Figure A.1 shows f_s and the relation to the RANS, LES and SBES eddy viscosities. It can be seen that in the wall boundary layer $\nu^{SBES} = \nu_{ij}^{RANS}$, while at $y^+ \approx 30$, f_s is decreasing exponentially.

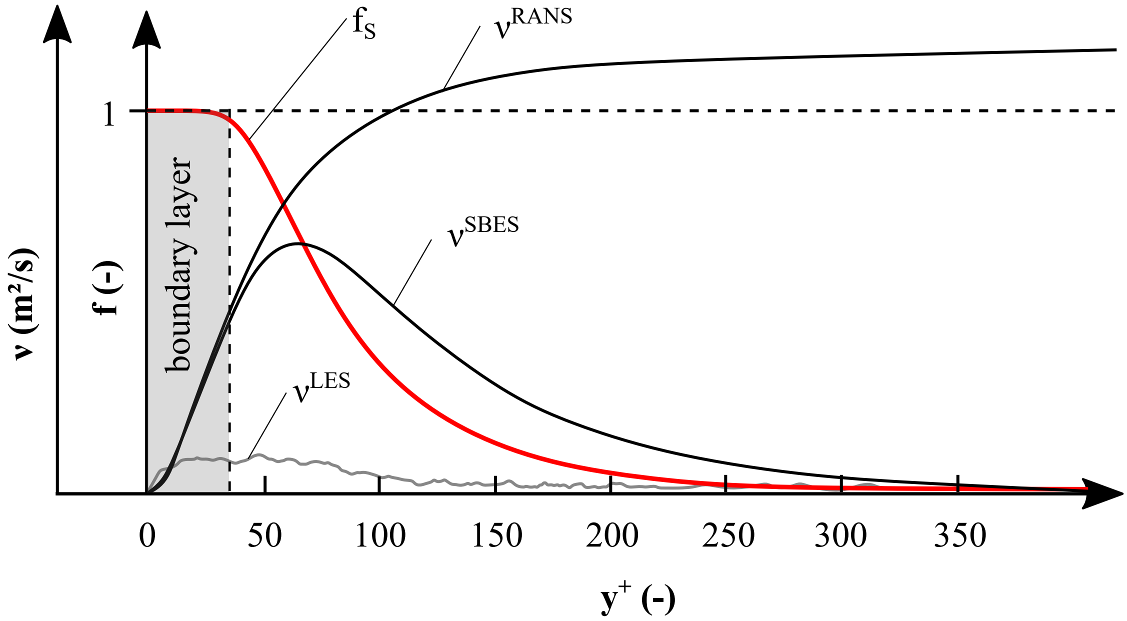

Figure A.1: Near wall treatment of a pipe wall with the SBES model (adapted from [95])

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APPENDIX B

B.1 Guide vane movement

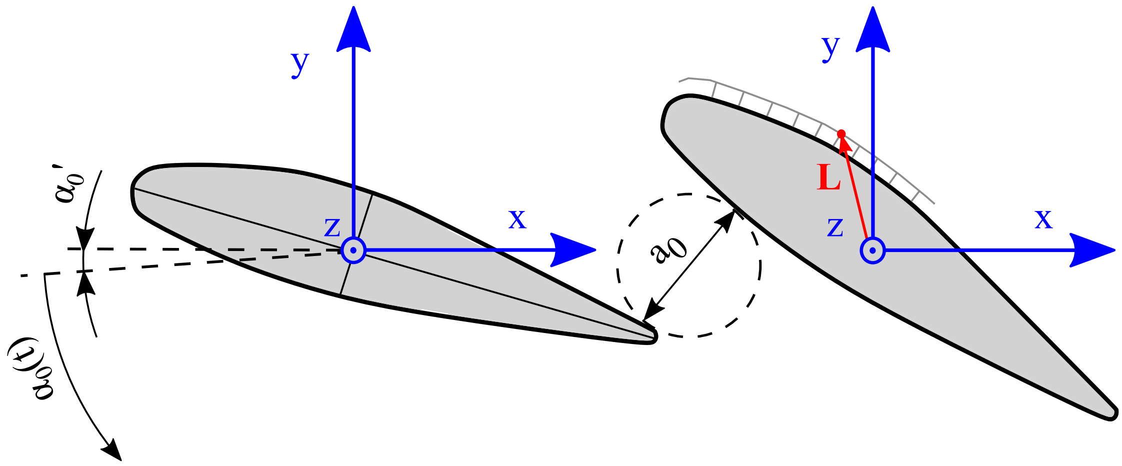

Figure B.1: Guide vane movement of a Francis turbine

As described in Chapter 1, an important part of most transient events is the guide vane movement. In Figure B.1 the opening angle α_0, which depends on the largest circle that fits through the opening between the guide vanes, is depicted. In the simulation, the rotation of a vector $\boldsymbol{L}$ in the Euclidean space is determined by

$$\boldsymbol{L}' = \boldsymbol{M_{rot}}(\boldsymbol{\alpha})\boldsymbol{L} \tag{B.1}$$

The rotation matrix $\boldsymbol{M_{rot}}$ is defined for active rotations of vectors counterclockwise in a right-handed coordinate system. Moreover, the rotation angle α_0 is depending on the time t. The remaining angle displacement from the horizontal axis to the positions until the guide vanes are entirely closed, is referred to as α_0'. However, the movement of the guide vane blade requires mesh deformation modeling. The "Displacement Diffusion" approach, which diffuses the motion of the boundaries to other mesh points is used. The model is explained in Chapter 3 and Chapter 5. Consequently, the vector $\boldsymbol{L}$ includes a relative displacement to the initial mesh. Equation B.1 is modeled in the ANSYS-CFX expression language (CEL) to specify the motion of the grid points depending on the boundaries.

Curriculum Vitæ

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Education

06/2016 – 10/2019 **Doctoral degree in Engineering Sciences Mechanical Engineering - Economics**, *TU Wien*, Vienna, Specialized in the field of renewable energy and numerical methods.

10/2014 – 06/2016 **Master's degree in Mechanical Engineering-Management**, *TU Wien*, Vienna.
Specialized in the field of hydraulic machinery and quality management

10/2010 – 05/2014 **Bachelor's degree in Industrial Engineering**, *TU Wien*, Vienna.
Specialized in the field of fluid mechanics

09/2004 – 06/2009 **Higher school degree**, *HTL Ottakring*, Vienna.
Mechanical Engineering

Community Service

10/2009 – 06/2010 **Austrian red cross**, *Purkersdorf*.
Rescue service

Work Experience

10/2019 – 01/2020 **Postdoctoral researcher**, *TU Wien Institute for Energy Systems and Thermodynamics/E302*, Vienna.
Numerical investigations of hydraulic turbines, Feasibility analysis of energy storage technologies

06/2016 – 10/2019 **Doctoral assistant**, *TU Wien Institute for Energy Systems and Thermodynamics/E302*, Vienna.
Numerical and experimental investigations of hydraulic turbines

09/2015 – 06/2016 **Scientific assistant**, *TU Wien Institute for Energy Systems and Thermodynamics/E302*, Vienna.
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09/2013 & 09/2014 **Teaching assistant**, *TU Wien Institute of Engineering Design and Product Development/E307*, Vienna.
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07 & 08/2007 & 07/2008 **Internship**, *Plasser & Theurer*, Purkersdorf.
Engineering Design

07/2006 **Internship**, *Konrad Klausser Schliessanlagen*, Vienna.
Manufacturing of locking systems

Languages

Native language	German
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Software Skills

Operating systems	MS Windows, Linux (Ubuntu)
Text processing	MS Office, LaTeX
CAD	AUTOCAD, CATIA, Pro/ENGINEER (Creo Elements/Pro), Solid Edge, SolidWorks
CFD	ANSYS (CFX, Fluent, ICEM CFD, Turbogrid), NUMECA Autogrid5, OpenFOAM
FEM	ANSYS Mechanical
Visualization	ANSYS CFD-Post, ParaView, SigmaPlot
Programming codes	APDL, C, Perl, Python
Data processing	FAMOS, MATLAB, LabVIEW

Awards

Best Paper Award 2nd IAHR-Asia Symposium on Hydraulic Machinery and Systems, 2019

Publications and Scientific Work

Conference papers with presentation

09/2019 J. Unterluggauer, E. Doujak and C. Bauer: **Experimental and numerical study of a prototype Francis turbine startup** In *Proceedings of the 2nd IAHR-Asia Symposium on Hydraulic Machinery and Systems*, Busan, Korea, 2019.

04/2019 J. Unterluggauer, E. Doujak and C. Bauer: **Numerical fatigue analysis of a prototype Francis turbine runner in low-load operation** In *Proceedings of the 13th European Conference on Turbomachinery Fluid Dynamics & Thermodynamics*, Lausanne, Swizerland, 2019.

11/2018 J. Unterluggauer, E. Doujak and C. Bauer: **Fatigue analysis of a prototype Francis turbine based on strain gauges** In *Proceedings of the 20th International Seminar on Hydropower Plants*, Laxenburg, Austria, 2018.

11/2016 J. Unterluggauer, M. Eichhorn and E. Doujak: **Fatigue analysis of Francis turbines with different specific speeds using site measurements** In *Proceedings of the 19th International Seminar on Hydropower Plants*, Laxenburg, Austria, 2016.

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Conference papers with contribution (co-author)

11/2018 F. Hahn, J. Unterluggauer and C. Bauer: **Performance improvement of a mixed-flow pump by means of multi-objective optimization** In *Proceedings of the 20th International Seminar on Hydropower Plants*, Laxenburg, Austria, 2018.

07/2018 E. Doujak and J. Unterluggauer: **Fluid-structure interaction of Francis turbines at different load steps** In *Proceedings of the 9th International Symposium on Fluid-Structure Interactions, Flow-Sound Interaction, Flow-induced Vibration & Noise*, Toronto, Canada, 2018.

Journal contributions

submitted J. Unterluggauer, E. Doujak and C. Bauer: **Experimental and numerical study of a prototype Francis turbine startup** In *Renewable Energy*

09/2019 J. Unterluggauer, E. Doujak and C. Bauer: **Fatigue analysis of a prototype Francis turbine based on strain gauges** In *Wasserwirtschaft Extra*, S. 25 - 30, 2019.

09/2019 J. Unterluggauer, M. Eichhorn and E. Doujak: **Fatigue analysis of Francis turbines with different specific speeds using site measurements** In *Wasserwirtschaft Extra*, S. 52 - 57, 2019.

08/2019 J. Unterluggauer, E. Doujak and C. Bauer: **Investigation on the Impact of Air Admission in a Prototype Francis Turbine at Low-Load Operation** In *Energies*, 12(15), 2893; https://doi.org/10.3390/en12152893, 2019.

07/2019 J. Unterluggauer, E. Doujak and C. Bauer: **Numerical fatigue analysis of a prototype Francis turbine runner in low-load operation** In *Int. J. Turbomach. Propuls. Power*, 4(3), 21; https://doi.org/10.3390/ijtpp4030021, 2019.

Journal contributions (co-author)

09/2019 F. Hahn, J. Unterluggauer and C. Bauer: **Performance improvement of a mixed-flow pump by means of multi-objective optimization** In *Wasserwirtschaft Extra*, S. 32 - 37, 2019.

Doctoral thesis

10/2019 J. Unterluggauer: **Investigations on Critical Low-Load and Transient Operation of a Prototype Francis Turbine** *Doctoral thesis*, TU Wien, Vienna, 2019.

Diploma thesis

06/2016 J. Unterluggauer: **Lifetime analysis of a Francis turbine from data of a prototype measurement** *Diploma thesis*, TU Wien, Vienna, 2016.

Supervised Diploma theses

in progress S. Stadler: **Numerische Lebensdauerberechnung einer Francis Turbine** *Diploma thesis*, TU Wien, Vienna, 2019.

in progress M. Maier: **Ausgewählte Kapitel der Festigkeits- und Leistungsbetrachtung an einer Francis Turbine** *Diploma thesis*, TU Wien, Vienna, 2019.

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10/2019 K. Mühlbacher: **Auswertung der Messdaten einer Prototypanlagenmessung** *Diploma thesis*, TU Wien, Vienna, 2019.

09/2019 A. Schmelz: **Numerische Simulation von Stabilisierungsluft in einer Francis Turbine** *Diploma thesis*, TU Wien, Vienna, 2019.

09/2018 F. Hahn: **Performance Improvement of a High Specific Speed Mixed-Flow Pump by Means of Multi-objective Optimization** *Diploma thesis*, TU Wien, Vienna, 2018.

10/2018 G. Fillinger: **Eigenschwingungsanalyse eines Francis-Turbinenlaufrads** *Diploma thesis*, TU Wien, Vienna, 2018.

Cooperation in scientific projects

10/2017 – 01/2020 "MDREST - Method development for lifetime assessment of hydropower turbines, BRIDGE, FFG project number 11364358

06/2016 – 10/2017 "MODPROT - Investigation on Modular Pump Turbines for laboratory and field testing", energy research, FFG project number 853658

09/2015 – 05/2017 "GSG - Green Storage Grid, Sub-project PSP-LowLoad: Unsteady behavior of turbines and pump turbines in low-load operation", „COMET K-Project", FFG project number 836636

Contribution to the application of scientific projects

09/2018 "FLEXPRIME: Flexibilisation of existing hydro power plants to increase the primary control reserve in the electricity grid", Energy research, FFG project number 836636

02/2018 "FLOWCELL: Flexible Hybrid Storage for Web of Cells", Energy research, FFG project number 16767456

09/2017 "FLEXSTO: Flexible storage systems to support distribution, Energy research", FFG project number 14060920

10/2017 "MDREST: Method development for lifetime assessment of hydro power turbines", BRIDGE, FFG project number 11364358

01/2017 "DECHSTO: Decentralised hybrid Storage System"', Energy research, FFG project number 860951

09/2016 Life+: "Method development for the fatigue analysis of hydraulic turbines using condition optimized maintenance strategies"., Energy research, FFG project number 9271981

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