Design of a Turbine That Utilizes the Energy in Spill Water of Large Ships

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Norwegian University of Science and Technology
Department of Energy and Process Engineering
MASTER THESIS

for

Student Christoffer Meek

Spring 2018

Design of a turbine which utilize the energy in spill water of large ships
Design av en turbin som utnytter overskuddsvann fra store skip

Background

Shipping continues to be one of the highest polluting industries on the planet. Exhaust emissions from marine engines comprise of nitrogen, oxygen, carbon dioxide and water vapour due to the heavy oil used in their operations. These emissions include smaller quantities of different oxides and various hydrocarbons. It is the smaller quantities, together: with CO2, that are of most concern to human health and the environment. As emission limits become more stringent, exhaust gas treatment becomes more challenging and costly. Ships, in their design, have massive amount of water circulating during operation which has the potential to create large amounts of kinetic energy.

The company QRRNT AS is developing turbines to capture the kinetic energy as water flows through the ship’s systems, returning the recovered energy directly to the ship. This system will ultimately save the ship owners on fuel costs, simultaneously reducing the emissions from the ship through recovering energy. The technology is designed to be able to be retrofit onto any ship with an open-loop system. Through creating custom designed turbines, QRRNT has the potential to capture this lost kinetic energy. This project will assist them in testing and verifying and optimising their technology.

Objective

Design and test a small axial turbine in the Waterpower Laboratory at NTNU

The following tasks are to be considered:

1. Literature study
   a. Design of low head hydro turbines
2. Software knowledge
   a. Ansys, Inventor and Labview
3. Turbine design
   a. New runner and guide vanes
4. Laboratory preparations
   a. Calibration
   b. Instrumentation and data acquisition setup
5. Efficiency measurements
6. If the student will go to Nepal for a excursion, earlier and further work will be presented as a publication and presented at the conference; 8th International symposium on Current Research in Hydraulic Turbines (CRHT-VIII) at Kathmandu University in March 2018.
7. If there is time available, the student will carry out CFD-analysis of the new runner design and compare the numerical and experimental data.
Within 14 days of receiving the written text on the master thesis, the candidate shall submit a research plan for his project to the department.

When the thesis is evaluated, emphasis is put on processing of the results, and that they are presented in tabular and/or graphic form in a clear manner, and that they are analyzed carefully.

The thesis should be formulated as a research report with summary both in English and Norwegian, conclusion, literature references, table of contents etc. During the preparation of the text, the candidate should make an effort to produce a well-structured and easily readable report. In order to ease the evaluation of the thesis, it is important that the cross-references are correct. In the making of the report, strong emphasis should be placed on both a thorough discussion of the results and an orderly presentation.

The candidate is requested to initiate and keep close contact with his/her academic supervisor(s) throughout the working period. The candidate must follow the rules and regulations of NTNU as well as passive directions given by the Department of Energy and Process Engineering.

Risk assessment of the candidate's work shall be carried out according to the department's procedures. The risk assessment must be documented and included as part of the final report. Events related to the candidate's work adversely affecting the health, safety or security, must be documented and included as part of the final report. If the documentation on risk assessment represents a large number of pages, the full version is to be submitted electronically to the supervisor and an excerpt is included in the report.

Pursuant to "Regulations concerning the supplementary provisions to the technology study program/Master of Science" at NTNU §20, the Department reserves the permission to utilize all the results and data for teaching and research purposes as well as in future publications.

The final report is to be submitted digitally in DAIM. An executive summary of the thesis including title, student's name, supervisor's name, year, department name, and NTNU's logo and name, shall be submitted to the department as a separate pdf file. Based on an agreement with the supervisor, the final report and other material and documents may be given to the supervisor in digital format.

☐ Work to be done in the Waterpower laboratory
☐ Field work

Department of Energy and Process Engineering, 15. January 2018

[Signature]
Ole Gunnar Dahlhaug
Academic Supervisor

Co-Supervisors:
- Bjørn Winther Solemslie
- Chirag Trivedi
Preface

This master thesis has been a practical and theoretical evaluation of QRRNT’s turbines, continued from the Project work. It has been very educational to work with a real turbine and to use theory to explain the physical phenomena. Furthermore, it has been very motivational for me that this work will be of great value for QRRNT and their future customers.

I would like to thank Julia Navarsete and the rest of the QRRNT team for giving me the opportunity to work with this turbine and providing all the necessary equipment. Particularly thanks to Fredrik Linge for a great collaboration during this project and for teaching me ANSYS CFX, which was essential to accomplish the second part of this thesis. It has been a pleasure to work with all of you, and I wish the company good luck in the future.

Furthermore, I would like to thank my supervisor Professor Ole Gunnar Dahlhaug for being an excellent advisor during this master thesis and for always taking the time to assist me when problems have occurred. Your positivity and engagement, have been motivating during difficult times. Also, thanks to the Ph.D. candidate Chirag Trivedi for assistance when Ole Gunnar has been unavailable.

Finally, thanks to Bård Aslak Brandåstrø and Joar Grilstad for consulting me during the practical preparations in the laboratory. Especially thanks to Trygve Opland for all the hours spent working on the test rig.

NTNU Gløshaugen, June 2018

Christoffer Meek
Abstract

In this master thesis, a Straflo turbine prototype used for energy recovering in large ships was tested, evaluated, and redesigned. The main goals were first to evaluate the performance of the prototype based on efficiency measurements from tests, and secondly to design a new improved turbine and evaluate the design. The prototype turbine was provided by the turbine development company QRRNT AS, and was designed for the circulation system of the fish tanks on a well boat. To map its performance at different operation points, a test rig was set up in the Waterpower laboratory to perform efficiency measurements. Based on experiences from these tests, another turbine was designed for a different ship. Then, CFD simulations was performed to evaluate and optimize the turbine design.

During the prototype testing, some challenges occurred due to the poor mechanical design of the turbine, which caused the runner to detach from the shroud and then the generator magnets to loosen. However, measurements showed that the prototype turbine was able to generate a maximum of 17-18 kW at an efficiency of approximately 72 % and a partly performance chart was created. To enhance future models, improvements include using runner blade modules and improving the hub and runner blade design.

The new turbine was designed and optimized according to these findings, and simulations predicted a hydraulic efficiency of 88.3 %, which is higher than the simulated and measured performance of the prototype turbine. The goals of this thesis were therefore achieved, although unplanned events limited the performance mapping of the prototype. Future work includes finding a solution to re-attach the magnets to the shroud and pursuing the performance mapping and performing additional CFD-simulations of the new turbine to optimize its blade thickness and study its transient behavior.

1Computational Fluid Dynamics
Sammendrag

Denne masteroppgaven består av en praktisk undersøkelse og re-design av en prototype Straflo turbin, som skal brukes til å gjenvinne energi fra spillvann på store skip. Hovedmålet med oppgaven har vært å først kartlegge turbinens ytelse ved hjelp av laboratorietesting, for å deretter utvikle et nytt og forbedret design basert på erfaringer fra testene. Prototypen ble levert av turbinleverandøren QRRNT AS, og var designet for installasjon i sirkulasjonssystemet til fisketanken på en brønnbåt. For å kartlegge turbinens virkningsgrad ved ulike driftsområder, ble det satt opp en testrigg på Vannkraftlaboratoriet og utført tester og målinger. Basert på disse testresultatene, ble det designet en ny og forbedret turbin til et annet skip. Designet ble deretter evaluert og optimalisert ved hjelp av numeriske strømningsberegninger.

Under testingen av prototypen oppstod det noen utfordringer som var knyttet til dårlig mekanisk utforming av turbinen, som førte til at først løpehjulet og deretter magnetene i generatoren løsnet fra rotoren. Til tross for dette, ble det målt en maksimal effekt på 17-18 kW ved en virkningsgrad på rundt 72 %, og et ufullkommen Hill diagram ble laget. For å forbedre ytelsen i fremtidige modeller kan modulbaserte løpehjul erstatte 3D-printede deler, i tillegg til at designet av løpehjulsskovler og nav kan forbedres.

I den andre delen av oppgaven, ble en ny turbin designet og optimalisert med hensyn på disse forbedringene. Det endelige designet har en simulert hydraulisk virkningsgrad på 88.3 %, som er høyere en prototypens målte og simulerte ytelse. Målet for denne masteroppgaven ble derfor oppnådd, selv om uventede utfordringer begrenset kartleggingen av ytelsen. Fremtidig arbeid inkluderer å feste magnetene på nytt og gjenoppta testing, i tillegg til å utføre flere simuleringer av den nye turbinen for å optimalisere bladtykkelsen og utelukke kavitasjon.
Contents

1 Introduction and Background .................................................. 3
  1.1 Background ........................................................................ 3
  1.2 QRRNT’s Turbines ............................................................... 4
  1.3 Introduction ...................................................................... 6

2 Theory .................................................................................. 7
  2.1 Fundamentals of Hydropower ............................................... 7
  2.2 Hydraulic Turbines and Efficiency Measurements .................. 8
    2.2.1 Straflo Turbines ............................................................ 10
  2.3 Turbine Design .................................................................. 14

3 Turbine Testing ....................................................................... 20

4 Test Results and Discussion ..................................................... 27
  4.1 Performance Test, 24.01.18 ................................................ 27
  4.2 Frequency Transformer Tuning, 09.03.18 ............................ 30
  4.3 Second Frequency Transformer Tuning, 13.03.18 ................. 31
  4.4 Flow Meter and Frequency Transformer Calibration, 07.06.18 . 33

5 Final Thoughts and Improvements ............................................ 35
## Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>Cross sectional area</td>
<td>$[m^2]$</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>Angle between Cu and Cm</td>
<td>[-]</td>
</tr>
<tr>
<td>$\beta$</td>
<td>Angle between U and W</td>
<td>[-]</td>
</tr>
<tr>
<td>$C$</td>
<td>Absolute velocity of the water</td>
<td>$[m/s]$</td>
</tr>
<tr>
<td>$C_L$</td>
<td>Coefficient of lift for a 3D airfoil in a cascade</td>
<td>[-]</td>
</tr>
<tr>
<td>$C_{L1}$</td>
<td>Coefficient of lift for a 3D airfoil</td>
<td>[-]</td>
</tr>
<tr>
<td>$Cm$</td>
<td>Axial (meridional) velocity component</td>
<td>$[m/s]$</td>
</tr>
<tr>
<td>$Cu$</td>
<td>Tangential velocity component</td>
<td>$[m/s]$</td>
</tr>
<tr>
<td>$D$</td>
<td>Turbine diameter</td>
<td>$[m]$</td>
</tr>
<tr>
<td>$d$</td>
<td>Bulb diameter</td>
<td>$[m]$</td>
</tr>
<tr>
<td>$\eta$</td>
<td>Total efficiency</td>
<td>[-]</td>
</tr>
<tr>
<td>$\eta_g$</td>
<td>Generator efficiency</td>
<td>[-]</td>
</tr>
<tr>
<td>$\eta_h$</td>
<td>Hydraulic efficiency</td>
<td>[-]</td>
</tr>
<tr>
<td>$f$</td>
<td>Frequency of the electrical power output</td>
<td>$[Hz]$</td>
</tr>
<tr>
<td>$f_x$</td>
<td>Uncertainty of variable &quot;x&quot;</td>
<td>[-]</td>
</tr>
<tr>
<td>$g$</td>
<td>Gravitational acceleration in the laboratory</td>
<td>$[m/s^2]$</td>
</tr>
<tr>
<td>$H$</td>
<td>Net head</td>
<td>$[m]$</td>
</tr>
<tr>
<td>$H_0$</td>
<td>Reservoir height</td>
<td>$[m]$</td>
</tr>
<tr>
<td>$H_s$</td>
<td>Turbine suction height</td>
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<td>$h_b$</td>
<td>Atmospheric pressure in meter water column (mWc)</td>
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<td>Friction losses</td>
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<td>$h_{va}$</td>
<td>Water vapor pressure</td>
<td>$[m]$</td>
</tr>
<tr>
<td>$K_u$</td>
<td>Peripheral velocity coefficient</td>
<td>[-]</td>
</tr>
<tr>
<td>$L$</td>
<td>Lift force</td>
<td>$[N]$</td>
</tr>
</tbody>
</table>
\( L_{hub} \) Axial length of the hub at the outlet [m]
\( l \) Blade chord length [m]
\( MT \) Maximum thickness of airfoil [m]
\( \dot{m} \) Mass flow rate [kg/s]
\( N_{ed} \) Speed factor [-]
\( N_q \) Specific speed [-]
\( NPSH_t \) Net Pressure Suction Head for turbine [m]
\( n \) Rotational velocity [RPM]
\( \Omega \) Speed number [-]
\( \omega \) Rotational velocity [rad/s]
\( P \) Shaft power [W]
\( P_{diff} \) Differential pressure over turbine [Bar]
\( P_e \) Electrical power output [W]
\( P_h \) Hydraulic power [W]
\( P_{inlet} \) Turbine inlet pressure [Bar]
\( P_{Lm} \) Dissipated power in turbine bearings [W]
\( P_{outlet} \) Turbine outlet pressure [Bar]
\( Q \) Volume flow rate \([m^3/s]\)
\( Q_{ed} \) Specific volume flow [-]
\( R \) Reaction ratio of a turbine [-]
\( \rho \) Water density \([kg/m^3]\)
\( s \) Airfoil radial length [m]
\( T \) Temperature [Celsius]
\( t \) Blade pitch [m]
\( \tau \) Torque [Nm]
\( U \) Peripheral velocity of the runner [m/s]
\( V \) Electrical voltage/ Volume \([V]/[m^3]\)
\( W \) Relative velocity [m/s]
\( Z \) Vertical height [m]
\( Z_b \) Number of runner blades [-]
\( Z_{gv} \) Number of guide vanes [-]
\( \varnothing \) Pipe diameter [mm]
Chapter 1

Introduction and Background

This chapter considers the background for this thesis and its span.

1.1 Background

Shipping continues to be one of the highest polluting industries on the planet. Exhaust emissions from marine engines comprise of nitrogen, carbon dioxide and water vapor due to the heavy oil used in their operations. These emissions include smaller quantities of different oxides and various hydrocarbons. It is the smaller quantities, together with $CO_2$, that are of most concern to human health and the environment. As emission limits become more stringent, exhaust gas treatment becomes more challenging and costly. Ships, in their design, have massive amounts of water circulating during operation and in some cases, the excess energy can be extracted.

The company QRRNT AS is developing turbines to capture the kinetic energy as water flows through the ship's systems, returning the recovered energy directly to the ship's power system. This installation will ultimately save the ship owners on fuel costs, simultaneously reducing the emissions from the ship through recovering energy. The technology is designed to be able to be retrofit onto any ship with an open-loop system\[1\]. Through creating custom designed turbines, QRRNT has the potential to capture the excessive energy. This project will assist them in testing, verifying and optimizing their technology.

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\[1\] Defined here as open to the atmosphere at some point in the system
1.2 QRRNT’s Turbines

QRRNT’s first prototype was originally designed for the fish-circulation system of a well boat. These boats have large elevated fish tanks that are used to transport living fish from fish farms to onshore facilities. The tanks require a considerable volume of circulating water, driven by a pump. Some of these boats have an open system, which enables power extraction using a hydraulic turbine placed at the outlet of the system (see Figure 1.1 under). More specifically, the prototype was designed for a volume flow rate of $0.7 \, \text{m}^3/\text{s}$ and an elevation of 5 meters from the sea level to the top of the tank. The turbine is a *Straflo turbine* and utilizes a 3D-printed runner and guide vanes in alimid and Nylon 12PA. With a retrofit design, it was intended to replace an existing Ø500 pipe-section. Theoretically, the turbine would produce around 30 kW of power (by Equation (2.1) and assuming a 90% efficiency), which is almost enough to cover the power consumption of the pump. In the long run, the ship will save fuel expenses without affecting the ship’s primary operation.

![Figure 1.1: Illustration of the QRRNT concept on a well boat.](image-url)
Another application for this type of turbine is in the scrubber system of cruise ships. As new maritime regulations require large ships to clean their exhaust gases using scrubbers, this application might cover a much larger customer base. Maritime scrubber systems utilize fresh sea water to spray the exhaust gases to absorb pollution and dust particles, which is then extracted in a filter (orange box in Figure 1.2) before the water is released into the sea.

The concept of installing a turbine at the outlet of the scrubber system is quite similar to the well boat application, except that this system requires a certain minimum water level in the upper reservoir for the scrubber to work. Therefore, this installation requires either controllable guide vanes or a frequency transformer to control the turbine's rotational velocity to manage the upper water level. A frequency transformer will be utilized in these turbines. The utilization of such system at the outlet of a scrubber system enables both power recovering and water level control. The upper reservoir of a scrubber system can typically be around 18 meters above the sea level and have volume flow rates of 0.16 m³/s, where potentially 25 kW of power could be recovered. However, the pipe dimensions are usually of Ø300 size, which will make the turbine quite small and might be challenging to design.

Figure 1.2: Illustration of the QRRNT turbine in scrubber systems.
1.3 Introduction

To assist QRRNT in their work, the main objectives of this thesis was firstly to evaluate the performance of the prototype turbine based on efficiency measurements from tests (continued from the Project work) and compare these to CFD-simulations performed by EDR Medeso. Secondly, a new turbine for scrubber systems was to be designed and optimized. To achieve these goals, a practical examination of the turbine and a theoretical analysis was done. In addition, several CFD-simulations were performed in collaboration with QRRNT, to evaluate the design of the new turbine. The necessary theory, methods, and results are presented in this thesis.

To understand the concepts in this thesis, the first chapter provides a review of the initial literature study that was done to obtain knowledge on hydroelectric power generation and turbine design. More specifically, it considers the fundamental equations of hydropower, the structure of hydraulic turbines and the development of a new method used to design the "scrubber turbine". Next, the two main parts follow.

The first main part of this thesis considers the practical preparations, experimental setup and test results from the laboratory testing. The practical preparations include sensor calibration, design and assembling of an improved test rig, and planning of necessary test methods. Challenges that occurred and main results obtained during testing can be found in this part. Design improvements found during testing is then further considered in the next part.

The second part involves the procedure of designing the "scrubber turbine". It considers the process of finding the main design parameters using the developed design method, creation of a turbine geometry and evaluation of performed CFD-simulations in ANSYS CFX. The latter part involves evaluating the turbine's estimated hydraulic efficiency and changing the design accordingly before a final optimized design was found.

Finally, the last chapter presents the main results and conclusions from both parts of the thesis. This part includes the main test results and potential improvements of the prototype, and the final design of the "scrubber turbine". Eventually, future work is discussed.
Chapter 2

Theory

This chapter is a review of the theoretical literature study that was done at the beginning of this thesis to understand the theory behind turbine design and testing. The theory is partly taken from the Project Work [15].

2.1 Fundamentals of Hydropower

The concept of hydropower is to extract the potential energy of an elevated water reservoir at a height $H_0$. At rest, the energy of the reservoir is $E = \rho \cdot V \cdot g \cdot H_0$. By differentiating the volume with respect to time, defining the net head as $H = H_0 - h_f$, and introducing the hydraulic efficiency $\eta_h$, to account for turbine losses, the power potential of the reservoir becomes:

$$P_h = \eta_h \cdot \rho \cdot g \cdot Q \cdot H \quad [W] \quad (2.1)$$

An enclosed pipe is utilized to transfer the energy from the reservoir into a hydraulic turbine at a lower altitude. The turbine transforms the potential energy into mechanical energy, which is then transferred to a generator for conversion into electrical energy. The generator is connected to a transformer and/or a frequency transformer that feeds the produced energy to a power grid. Equation (2.1) can be used to calculate the expected total power output from a reservoir by using the total efficiency, which is defined here as $\eta = \eta_h \cdot \eta_g$, assuming that other losses are negligible. Note that from this point, the net head will be addressed as the head.
2.2 Hydraulic Turbines and Efficiency Measurements

Hydraulic turbines convert kinetic and pressure energy into rotational energy that contributes to torque $\tau$ in the rotating runner and gives shaft power equal to $P = \tau \cdot \omega$. A measurement of the extracted energy in a turbine is the effective head, and is calculated by Bernoulli’s equation. By neglecting change in height (for horizontal turbines) and assuming continuity and incompressibility ($Q_{inlet} = Q_{outlet} = Q$), the effective head becomes:

$$H = \frac{P_{inlet} - P_{outlet}}{\rho \cdot g} + \frac{Q^2 \left( \frac{1}{A_{in}} - \frac{1}{A_{out}} \right)}{2 \cdot g} \quad [\text{mWc}]$$ (2.2)

Using this equation, the head can be calculated by measuring the pressure difference across the turbine and the volume flow rate. In addition, the water temperature needs to be measured, such that accurate values for the density $\rho$ can be found in thermodynamic tables. Some of the energy is dissipated in turbine bearings and is defined as $P_{Lm} = P_h - P$. This quantity is however difficult to measure but can be assumed negligible here. The hydraulic efficiency of a turbine is a measurement of the ability to convert hydraulic energy into mechanical shaft power, and is defined by IEC as [1]:

$$\eta_h = \frac{P + P_{Lm}}{\rho \cdot g \cdot Q \cdot H} \approx \frac{P_e}{\eta_g \rho \cdot g \cdot Q \cdot H} \quad [-]$$ (2.3)

As bearing losses are assumed negligible, the hydraulic efficiency can be estimated using the measured electrical power output $P_e$ and the known generator efficiency $\eta_g$, as shown in Equation (2.3). By including the generator efficiency, the total efficiency can be calculated by:

$$\eta = \frac{P_e}{\rho \cdot g \cdot Q \cdot H} \quad [-]$$ (2.4)

Hydraulic turbines are designed for a specific head $^*H$, volume flow rate $^*Q$, and rotational velocity, $^*n$. The maximum hydraulic efficiency is usually achieved at these design conditions, and is called the Best Efficiency Point or BEP. A Hill chart can be used to predict the turbine performance outside this point. A typical diagram can be seen in Figure 2.1.
Figure 2.1: Hill chart for a Kaplan turbine with $\Omega=1.3$. $\phi$ is the runner blade angle and $S_0$ is the guide vane angle, which both are fixed in QRRNT’s turbines.

The performance characteristic of a turbine is usually mapped by the turbine producer by prototype testing. Instead of using relative volume flow rate and rotational velocities as in Figure 2.1, Hill charts are usually given by non-dimensional (reduced) parameters to make the results applicable in full-scale systems. This definition use a turbine with $D=1\, m$, $Q=1\, m^3/s$ and $H=1\, m$ as reference, and defines the unit volume flow (flow factor) and the unit speed (speed factor) as \[9]\:

\[ Q_{ed} = \frac{Q}{D^2 \cdot \sqrt{g \cdot H}} \quad [-] \quad (2.5) \]

\[ N_{ed} = \frac{n \cdot D}{\sqrt{g \cdot H}} \quad [-] \quad (2.6) \]

For the prototype, these values becomes $Q_{ed}=0.4$ and $N_{ed}=35.7$ at design point. As two turbines of the same speed number have an equal Hill chart, this parameter can be used to compare turbine characteristics. It utilizes values at the BEP and is defined as:

\[ \Omega = \frac{\omega \cdot \sqrt[3]{Q}}{(2 \cdot g \cdot H)^{3/4}} \quad [-] \quad (2.7) \]

This equation gives a speed number of $\Omega = 1.4$, which is almost similar to the turbine
CHAPTER 2. THEORY

used to produce the Hill chart in Figure [2.1]. Therefore, the measured prototype characteristic should look similar to this. Another non-dimensional turbine parameter is the specific speed, which is used in turbine design. It is defined as:

\[ N_q = \frac{n \cdot Q^{0.5}}{H^{0.75}} \]  \hspace{1cm} (2.8)

The choice of turbine type in a hydropower system depends on the available head and volume flow rate, as each turbine have an ideal working range. For low heads and large volume flow rates, Kaplan turbines (also called propeller turbines) are mostly utilized. There are also multiple variants of Kaplans, such as the Straflo turbine.

### 2.2.1 Straflo Turbines

QRRNT are utilizing Straflo turbines, which is an axial Kaplan turbine variant. These turbines are quite similar to bulb turbines, except that the generator is fixed at the perimeter of the runner. This turbine type is utilized for relative small heads and large volume flow rates. The structure of these turbines, make them ideal to use in pipe-sections (see figure under).

![Figure 2.2: Illustration of a Straflo Turbine. The guide vanes are marked in green, and the runner and generator marked in red [5].](image)

The figure above illustrates how Straflo turbines work. Water flows into the turbine from the left and flows around the fixed bulb, which is stabilized by support beams. Then the water enters the guide vanes (green) that are fixed in QRRNT’s turbines, which induces rotation.
to the flow before it is extracted in the runner (red). If the turbine is running at the BEP, the water will have nearly no rotation at the turbine outlet. This means that the velocity components of the water follows the runner blades’ geometry as designed.

Similar to other Kaplan turbine variants, the guide vanes and runner blades have an airfoil shape, which can be modeled in 2D as in Figure 2.3. In order to obtain the highest turbine efficiency, the angle at the inlet and outlet of the runner blades must be optimized according to the velocity components of the flow. The absolute velocity of the water $C$, can be decomposed into the axial $C_m$, and tangential $Cu$ direction. $U$ is the peripheral velocity of the runner blades at a certain radial distance from the center $r$, given by Equation (2.9):

$$U = \frac{n \cdot r \cdot \pi}{30} \text{ [m/s]}$$

(2.9)

Furthermore, the water velocity relative to the runner blade is calculated by $\vec{W} = \vec{C} - \vec{U}$. The subscripts 1, 2 and 3 means respectively the positions at the guide vane inlet, runner inlet, and runner outlet. As can be seen in the figure under, $\alpha$ is defined as the angle between $Cu$ and $C$, and $\beta$ as the angle between $U$ and $W$. To ensure the most optimal flow in the turbine, the geometry of the blades must match these angles, such that $\alpha_2$ becomes equal to the guide vane exit angle, $\beta_2$ equal to the runner inlet angle and $\beta_3$ equal to the runner exit angle.

![Figure 2.3: Illustration of a guide vane and runner blade, and the velocity components.](image)

By using Euler’s turbine equation, the hydraulic efficiency of a turbine can be estimated by the difference in rotational momentum at the runner inlet and outlet [11]:

$$\text{Efficiency} = \frac{U_2 - U_1}{C_2 - C_1}$$
\[ \eta_h = \frac{1}{g \cdot H} (U_2 \cdot C_{u2} - U_3 \cdot C_{u3}) = \frac{U}{g \cdot H} (C_{u2} - C_{u3}) \quad [-] \quad (2.10) \]

To calculate the blade geometry for a specific turbine, irrotational flow at the runner outlet at design point is assumed, thus \( C_{u3} = 0 \). Then, the Euler equation simplifies to: \( \eta_h = \frac{U \cdot C_{u2}}{g \cdot H} \). By assuming an hydraulic efficiency and inserting \( \ast \) \( H \) and \( U \) (using \( \ast n \)), this equation can be used to calculate the \( C_{u} \) component along the blade. Furthermore, the axial velocity components are given by continuity as \( Q \) and \( A \) is constant, such that \( C_{m2} = C_{m3} = C_{m} \). This velocity component can be calculated by the following equation:

\[ C_{m} = \frac{Q}{A} = \frac{4 \cdot Q}{\pi \cdot (D^2 - d^2)} \quad [m/s] \quad (2.11) \]

Finally, from the velocity triangles in Figure 2.3, the following relations for the blade angles can be derived:

\[ \alpha_2 = \tan^{-1} \left( \frac{C_{m}}{C_{u2}} \right) \quad [-] \quad (2.12) \]

\[ \beta_2 = \tan^{-1} \left( \frac{C_{m}}{U - C_{u2}} \right) \quad [-] \quad (2.13) \]

\[ \beta_3 = \tan^{-1} \left( \frac{C_{m}}{U} \right) \quad [-] \quad (2.14) \]

The inlet and outlet angle of the runner blade are fixed by these relations, but the in-between shape is defined by choosing a \( U_{C_{u}} \)-distribution along the blade. As the blade angle is decreased from the inlet to the outlet, the rotational velocity component of the water \( (C_{u}) \) is decreased accordingly as the energy is transferred to the runner. The energy decrease can occur linearly from the inlet to the outlet or by extracting 75 % in the first half of the blade and 25 % in the remaining half. This distribution determines the curvature between the inlet and outlet of the runner blades.

Another aspect to consider is the thickness distribution along the blade. The blade thickness is important to minimize the boundary layer around the blades and to prevent flow separation. A widely used thickness distribution is the symmetric NACA 4-digit airfoil series, where the thickness varies as a "droplet"-shape along the center line. This distribution min-
CHAPTER 2. THEORY

imizes the effects of surface roughness and has good stall characteristics, which mean that it can handle a wide range of inflow angles without flow separation occurring. When a fluid flows over an airfoil at a certain angle of attack, a lifting force is induced. This force is called lift $L$, and is the force that contributes to favorable torque in the runner. If the angle of attack is too large, stall occurs, and the lift is reduced. It is common to represent an airfoil’s lift force abilities using the dimensionless lift coefficient, which is defined as [7]:

$$C_{L1} = \frac{L}{\frac{1}{2} \cdot \rho \cdot W^2 \cdot l \cdot s} \quad [-]$$

In this equation, $l$ is the airfoil’s chord length, which is the straight distance from the airfoil inlet (leading edge) to the outlet (trailing edge). For thin airfoils, the $s$-parameter is the radial length of the airfoil, such that $l \cdot s$ approximates the airfoil surface. However, when multiple airfoils are put together in a cascade, the effective lift coefficient of each airfoil changes ($C_L$). The pitch $t$, is defined as the distance between the leading edge of two airfoils, as can be seen in Figure 2.4a under:

(a) Illustration of Kaplan blades, which is equivalent to a cascade of airfoils.

(b) Cascade lift coefficient ratio versus the pitch-chord ratio [4].

Figure 2.4: Cascade effects in Kaplan turbines.

The effective lift coefficient of each airfoil in a cascade depends strongly on the pitch-chord ratio $t/l$, and the angle of attack of the flow $\beta$. As can be seen in Figure 2.4b, the highest effective coefficient of lift is found when $t/l = 1$. This ratio is therefore important to consider when designing Kaplan turbines.
2.3 Turbine Design

The concept of designing turbines is optimizing the geometry of the runner and guide vanes to achieve the best hydraulic efficiency. By using the relations from the previous chapters in addition to empirical relations from literature, new custom turbines can be designed to meet new customer demands [10]. The necessary design parameters are the volume flow rate $Q$, the head $H$, and the outer diameter $D$, which all are determined by the system where the turbine will be installed.

Since Kaplan turbines are reaction turbines, the geometry must ideally be designed such that the energy at the runner inlet consists of approximately 50% kinetic energy and 50% pressure. The ratio of pressure to total energy is called the reaction ratio, and is calculated by $R = 1 - \frac{c^2}{2gH}$ using Bernoulli’s equation. In practice, it is difficult to achieve an ideal reaction ratio as many of QRRNT’s potential customers have hydraulic systems that are not ideal for Straflo turbines. Axial Kaplan turbines are normally utilized in systems with large volume flow rates and low heads, contrary to the high-pressure conditions in the scrubber systems. As a consequence, conventional design methods cannot be used to create the turbine design for these systems. Therefore, a new and iterative method has been developed to determine design parameters for all hydraulic systems.

The main challenge when designing a turbine for systems with a large head and low volume flow rate is finding the design rotational velocity and the hub versus shroud diameter ratio of the turbine $d/D$. These are usually found using empirical relations for the specific speed parameter (see Equation (2.8)), which are however useless in this case, as they predict unrealistically high rotational velocities. The solution to this was therefore to use an empirical relation for the peripheral velocity coefficient $K_u$, to estimate a rotational velocity that was re-inserted into Equation (2.8) to calculate a new specific speed. This iterative step was executed until the rotational velocity converged. It was then rounded off to the nearest hundred rpm and chosen as the design rotational velocity. Based on this value, the other design parameters could be found from literature. The full design method goes as follows:
1) Choose a rotational velocity based on \( \ast H, \ast Q \) and \( D \)

In order to make an initial guess on the ideal rotational velocity, an empirical relation for \( N_q \) is utilized. For bulb turbines, the specific speed can be be estimated to \([14]\):

\[
N_q = 1059.2 \cdot \ast H^{-0.625} \quad [-]
\]  
(2.16)

Using this guess, a correlation between \( N_q \) and the peripheral velocity coefficient is used to find the rotational velocity \([14]\): \( K_u = 1 + 0.0038N_q \). The \( n \) is then calculated by:

\[
\ast n = K_u \cdot 60 \cdot \frac{\sqrt{2 \cdot g \cdot \ast H}}{\pi \cdot D} \quad [rpm]
\]  
(2.17)

Next, the new \( N_q \) is calculated by Equation (2.8). As this rotational speed tends to be too high, iteration is necessary to obtain a useful value. Thus, iteration of the \( K_u \) correlation, Equation (2.17) and (2.8), is repeated until the rotational velocity converges (see Figure 2.6).
Finally, the rotational velocity is rounded off to the nearest hundred for practical purposes. The MATLAB-script `Initializer.m` (see Appendix G) was created to perform this step.

2) Find the d/D ratio

To find the diameter of the hub, an empirical relation is utilized. By calculating \( N_s = 3N_q \) for the chosen \( *n, *H \) and \( *Q \), the \( d/D \) ratio can be read from Figure 9.18 in "Pumper og Turbiner" or from Figure 2.7 [10]. The hub diameter can then be found by \( d = D \cdot d/D \).

3) Find the number of runner blades and guide vanes

Figure 2.7: Chart used to find the number of runner blades [19].
To find the number of runner blades $Z_b$, experimental data from literature is utilized [19]. By using the final value for $N_q$ calculated in step 1, the number of runner blades can be extrapolated and read in Figure 2.7. Next step is then to determine the number of guide vanes, $Z_{g_v}$. This is done intuitively by choosing around twice as many guide vanes as runner blades, but selecting a number that is not divisible on $Z_b$ to ensure a smooth flow. The optimal number of guide vanes is found by simulation.

4) Calculate the velocity components and blade angles

When the essential design parameters are set, the next step is to calculate the velocity components and the blade geometry. By assuming a hydraulic efficiency of 90 % and using Equation (2.10), the tangential velocity component at the runner inlet becomes $C_u_2 = \eta \cdot g \cdot H \cdot U$. Then, the guide vane outlet angle $\alpha_2$, the runner inlet angle $\beta_2$ and the runner outlet angle $\beta_3$, can be calculated using Equations (2.12), (2.13), and (2.14) respectively.

5) Evaluate the reaction ratio and velocity triangles

To obtain the most optimal geometry, it is desirable that $W_2 \sim U$, $C_u_2 \sim 2U$ and that the runner blade inlet angles are not too large or the outlet angles too small [13]. Furthermore, to achieve the best efficiency, the reaction ratio should be around 50 %. If these requirements are not met, the design rotational velocity needs to be changed.

6) Check if cavitation might occur

It is important to verify that cavitation not can occur in the chosen geometry due to undesirable high velocities. This can be examined by the turbine *Net Pressure Suction Head* ($NPSh_t$), which is estimated by [10]:

$$NPSh_t = 1.1 \cdot C m^2 + 0.1 \cdot U^2 \quad [m]$$

(2.18)

It is required that the $NPSh_t$ produced by the turbine plus the height to the free water surface $H_s$, are less than or equal to the atmospheric pressure $h_b$, minus the vapor pressure
of water $h_{va}$. Thus:

$$H_s + NPSH_t \leq h_b - h_{va} \quad (2.19)$$

If this is not satisfied, the rotational velocity must be changed to avoid cavitation risk.

7) Create a model based on chosen design parameters

If the blade geometry passes all the requirements, the full 3D geometry can be computed. It is then necessary to choose the following design control parameters:

- **Energy distribution:** The energy distribution determine how the energy is extracted in the runner blades. A linear or a custom distribution can be chosen here.

- **Pitch-chord ratio:** As mentioned earlier in this chapter, the ideal pitch-chord ratio is $t/l = 1$. To obtain an average ratio of 1, the ratio is set to 0.9 at the hub and 1.1 at the shroud [14].

- **Blade thickness distribution:** A maximum blade thickness for the runner blades and guide vanes must be set, in addition to a thickness distribution along the blades. A custom distribution or the more conventional NACA 4-series thickness distribution can be chosen.

After choosing all design parameters, they can be inserted into QRRNT’s copyrighted ANSYS Workbench-wizard, which creates a 3D model of the turbine and sets up a CFD-simulation.

8) Perform a CFD-simulation

When the 3D model is created, the inlet and outlet region and the boundary conditions must be specified. Then, a CFD-simulation is performed to evaluate the turbine’s performance. Either a full 360-degree model or an axis-symmetric single passage model can be used. For more details on this step, see Chapter 6.2.
9) Evaluation

After simulation, the estimated hydraulic efficiency and the flow at the inlet and outlet of the blades must be examined. If the hydraulic efficiency is less than around 85%, the geometry needs to be improved to obtain higher performance. Secondly, the blade inlet must be examined to make sure the flow enters the blade smoothly at a correct angle of attack. At the outlet it is crucial to see if slip occur. This is the phenomenon that water flow fails to "stick" to the lower surface of the runner blades and separates, which changes $\beta_3$ (see Figure 2.8). Accordingly, the velocity components at the turbine outlet are changed, which affects the volume flow through the turbine.

Figure 2.8: Illustration of slip, which is deflection of the outlet flow. $\beta_3$ is the actual angle of the flow, and $\beta_{3\infty}$ is the outlet angle of the blade.

Therefore, if slip occurs, this must be accounted for to ensure the desired volume flow rate. This is important to not affect the ships' system by accumulating water. If the CFD-simulations indicate a too small volume flow rate, the blade geometry needs to be adjusted and the method repeated.

If all requirements are met, and the turbine efficiency is satisfactory, a final 3D transient simulation has to be performed. This can reveal transient phenomena such as cavitation, vortices and pressure pulsations. The final turbine design is found if no such behavior can be seen in a transient simulation.
Part I

Turbine Testing
Chapter 3

Experimental Setup and Method

This chapter considers the experimental setup and methods used for testing QRRNT’s prototype turbine in the Waterpower laboratory. It describes the test rig, the hydraulic system in the laboratory, utilized equipment, and what type of tests that were performed.

3.1 Laboratory Setup

A test rig was designed to supply the turbine with water at the same conditions as in the fish circulation system on the well boat. The existing rig from the Project Work was modified with a larger bend upstream of the turbine and was then re-assembled. A sketch of the improved test rig and its main dimensions can be seen in the figure under:

![Figure 3.1: Sketch of the test rig, showing the main dimensions in mm.](image-url)
By replacing the old Ø300 bend with a larger Ø600 pipe, the cavitation that occurred at large volume flow rates should be eliminated. Since cavitation is caused by too low pressure due to high water velocities, will pipes with increased cross-sectional area decrease the water velocity by continuity \((Q = C \cdot A)\), and thus decrease the risk of cavitation. Besides, using larger pipes required a narrowing part before the Ø500-section, which can contribute to a more uniform flow at the turbine inlet. Since the length of the Ø500 pipe is equal to 10D, a fully uniform flow can be assumed at the turbine inlet [6]. The modification also required a new vertical Ø600 pipe-section and a Ø600 butterfly valve to control the flow. These parts can be seen in the 3D-drawing of the test rig below:

![3D model of the test rig with the turbine seen in turquoise.](image)

The Ø600 butterfly valve was assembled onto the main pipe of the hydraulic system in the laboratory, which can be seen in Figure [3.3]. Instead of performing tests using the upper reservoir as in the Project Work, a pressure tank was utilized, which enabled improved control of the head and flow. This tank can be seen next to the test rig in Figure [3.3].

By using the pressure tank, the water path goes from the pump in the lower reservoir (the blue machine to the left in Figure [3.3]) and into the pressure tank, before it flows down to the test rig, through the turbine and back into the lower reservoir. The volume flow rate and the head are controlled by adjusting the rotational velocity of the pump and by adjusting the level of opening of the valve downstream of the turbine. This test configuration ensures more efficient testing, as the hydraulic conditions can be adjusted with less delay in the system.
Figure 3.3: 3D model of the hydraulic system in the laboratory. The test rig is marked in red.

### 3.2 Sensors and Equipment

When using the pressure tank, the water does not flow through the previously used flow meter. Therefore, an ultrasonic flow meter was utilized on the Ø500 pipe-section upstream of the turbine to measure the volume flow rate. Instead of using a load bank to dump the generated power, a frequency transformer was installed, which enabled control of the turbine’s rotational velocity using LabView. The frequency transformer, a Yaskawa U1000 MATRIX Drive, enables power feeding to the local grid. It also measures the power output and the frequency, which can be imported into the LabView monitoring program. The rotational velocity can then be computed by: $n = \frac{f}{7.5}$ [16].

The same pressure transducers and temperature meter that was used in previous tests were utilized here. Ring manifolds were used approximately 10 cm upstream of the turbine inlet and within the inlet of the downstream Ø600 pipe-section, to tap-mount the pressure transducers to obtain accurate pressure measurements. In the following picture (Figure 3.4) of the experimental setup, the frequency transformer can be seen next to the test rig.
All sensors that were utilized in the tests were re-calibrated and the corresponding uncertainties re-calculated (can be found in Appendix B.3). A LabView program from the laboratory was utilized to calibrate the pressure sensors. A complete list of sensors and the corresponding calibration uncertainty can be found in the following table:

<table>
<thead>
<tr>
<th>Type</th>
<th>Variable</th>
<th>Sensor</th>
<th>Calibration Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Absolute pressure</td>
<td>$P_{inlet}$</td>
<td>Druck PTX 610</td>
<td>±0.01029%</td>
</tr>
<tr>
<td>Absolute pressure</td>
<td>$P_{outlet}$</td>
<td>Druck PTX 610</td>
<td>±0.01048%</td>
</tr>
<tr>
<td>Differential pressure</td>
<td>$P_{diff}$</td>
<td>FHCX 36W1-AKBYY</td>
<td>±0.02967%</td>
</tr>
<tr>
<td>Volume flow rate</td>
<td>$Q$</td>
<td>Fuji Portaflow C</td>
<td>±1.036%</td>
</tr>
<tr>
<td>Temperature</td>
<td>$T$</td>
<td>Systemteknik S1220 PT100</td>
<td>±0.05%</td>
</tr>
<tr>
<td>Rotational velocity</td>
<td>$n$</td>
<td>Yaskawa U1000</td>
<td>±0.39%</td>
</tr>
<tr>
<td>Electrical Power</td>
<td>$P_e$</td>
<td>Yaskawa U1000</td>
<td>Unknown</td>
</tr>
</tbody>
</table>

Table 3.1: Sensors used in experiments.
To convert the electrical signals from the sensors into physical values, a monitoring program in LabView was created and utilized during testing. This program was used to monitor the flow and turbine conditions, log measurement data and to control the rotational velocity of the turbine from the frequency transformer. The program was also modified to enable real-time data streaming to EDR Medeso’s online server, to create a digital twin of the turbine.

3.3 Test Method

The main goal of testing the turbine was to map its performance and produce a Hill chart based on efficiency measurements. Therefore, the two main laboratory tests that were planned in this thesis was a full performance test of the original design and another using a 3D-printed plastic cone attached to the rear end of the hub. The test procedure is described in detail together with the potential HSE risks (in Norwegian) in Appendix F.

3.3.1 Performance Test

In order to produce a Hill chart to demonstrate the turbine’s efficiency over a wide range of operation points, several measurements at different heads and rotational velocities were required. To ensure a small uncertainty, sensor data for each measurement point was recorded over a 60 second time interval. To obtain a full Hill chart, the measurement points must range from $\pm 50\%$ of the BEP. This corresponds to the following range of measurement points [17]:

<table>
<thead>
<tr>
<th>Head</th>
<th>Rotational Velocity Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>3 m</td>
<td>300-700 rpm</td>
</tr>
<tr>
<td>4 m</td>
<td>300-700 rpm</td>
</tr>
<tr>
<td>5 m</td>
<td>300-700 rpm</td>
</tr>
<tr>
<td>6 m</td>
<td>300-700 rpm</td>
</tr>
<tr>
<td>7 m</td>
<td>300-700 rpm</td>
</tr>
</tbody>
</table>

Table 3.2: Planned test parameters for the performance test.

From CFD-simulations performed by EDR Medeso, the expected maximum efficiency at the design point is $\eta_h = 87\%$, which correspond to a total efficiency of $\eta = 83.3\%$ including the generator efficiency [16]. This remains to be measured to verify the CFD-model.
3.3.2 Performance Test with Enhancing Cone

After mapping the performance of the original design, the same measurements will be performed with a smooth cone attached to the rear end of the hub (see Figure 3.5). This cone was designed and 3D-printed in the laboratory with the intention to ensure a smoother backflow in the turbine. CFD-simulations of the turbine with this smooth hub extension indicates an efficiency increase to $\eta_h = 90\%$. It is therefore interesting to investigate this in practice, as this is a significant performance improvement.

Figure 3.5: Picture if the 3D-printed cone attached to the rear end of the hub.
Chapter 4

Test Results and Discussion

The testing of the prototype turbine was continued from the Project work [18]. This chapter presents the main results of the tests performed in this thesis. Uncertainty calculations for the tests can be found in Appendix B.3 and the full test results can be found in Appendix D. The resulting plots presented in this chapter have been created using MATLAB scripts to analyze the measurement data, and can be found in Appendix G.

4.1 Performance Test, 24.01.18

With a new test rig in place, a new performance test was performed to obtain improved results. It was discovered that the resistance in the load bank utilized in previous tests, could be manually adjusted. Therefore, with an improved test rig and the ability to partly control the rotational velocity, the turbine would hopefully operate closer to the design point. The ultrasonic volume flow meter was utilized for flow measurements, which enabled the use of the pressure tank instead of the upper reservoir. Accordingly, the turbine inlet pressure was then directly controlled by the pump's rotational velocity.

During testing, no noise or vibrations could be observed from the upstream bend, which indicates that no cavitation occurred. At higher volume flow rates, cavitation seemed to occur in the downstream bend, which however not did affect the test. Besides this, the test was successfully completed, but the BEP was not reached. The measurements were analyzed, and the main results can be seen in the following plots:
CHAPTER 4. TEST RESULTS AND DISCUSSION

Figure 4.1: Measured power versus head and theoretical value.

Figure 4.2: Total measured efficiency versus the speed factor.
CHAPTER 4. TEST RESULTS AND DISCUSSION

29

Figure 4.3: An intuitive performance chart from the test.

As can be seen in both Figure 4.2 and 4.3, the highest measured value was not sufficiently near the design point to produce a satisfactory Hill chart. However, the measured Hill chart follows the same shape as the constant $\phi$-lines in Figure 2.1 as expected. Figure 4.1 shows a maximum power output of around 13 kW, which is the highest power output the prototype has delivered. Some of these results are yet quite surprising. The maximum efficiency in this test was measured at $N_{ed} = 15$, which correspond to $H = 2\ m$. In Figure 4.1 a substantial deviation from the expected power output can be seen at this point. The same efficiency peak was also observed (but not logged) during a demo test. The reason for this efficiency peak is not easy to explain, as the efficiency is expected to increase smoothly towards the BEP.

Unfortunately, the uncertainty of the flow factor and efficiency were quite large, due to manual measurements of $Q$, $n$ and $P_e$ as the digital recordings failed. This issue must be fixed before further testing to obtain more accurate results. Nevertheless, a maximum efficiency of around 60 % is still far from the expected 83 %. In order to make the turbine operate closer to the designed conditions, the rotational velocity of the turbine needs to be increased. A frequency transformer is therefore necessary to obtain better results.
4.2 Frequency Transformer Tuning, 09.03.18

After the previous test, QRRNT provided a frequency transformer that was compatible with the permanent magnet generator in the turbine. This device required to be tested and adjusted for future tests. After successfully using it to run the turbine as a motor, next step was to tune in the frequency transformer to enable power generation. There were a lot of failed runs due to missing control parameters. Therefore, the inductance in the windings of each phase in the generator was manually measured to improve the generator control. After providing these values, the turbine was for the first time able to feed power to the grid.

In an attempt to reach the BEP, the applied volume flow rate and the head were gradually increased together with the rotational velocity. At an instance, the power output was measured to 17-18 kW, at $H = 4.5$ m, $Q = 0.55 \text{ m}^3/\text{s}$ and $n = 300$ rpm (unfortunately only the power was logged). This correspond to $Q_{ed} = 0.3$ and $N_{ed} = 22.6$, which is still far from the BEP. At this point however, the total efficiency was approximately $\eta = 72\%$, which is the highest measured. Shortly after this point, when the rotational velocity was slightly decreased, a sudden large drop in the head was observed (see Figure 4.4). Accordingly, the current levels in the generator were significantly increased, which resulted in the frequency transformer faulting and stopping the generator. Then, the test was aborted.

![Figure 4.4: Sudden head drop observation from LabView.](image)

The sudden head-drop was assumed to be caused by the pump in the laboratory reaching its volume flow rate limit. Therefore, a new test was planned using two pumps in parallel to ensure a sufficient volume flow rate.
4.3 Second Frequency Transformer Tuning, 13.03.18

For this test, two pumps in parallel were used to provide the necessary head at large volume flow rates. As in the previous test, the rotational velocity was gradually increased. In contrast to last time, the turbine was vibrating significantly and causing resonance in the upstream pipes. This vibration was temporarily limited by increasing the rotational velocity. The volume flow rate was increased to 0.45 $m^3/s$, but still only a small head was measured. At this point, the turbine generated alarmingly less power than it did on the last test.

When the rotational velocity was decreased as an attempt to produce more power, the vibration returned, which caused large oscillations in both pressure and power output (see Figure 4.5). To see if the instability was caused by the dynamics of two parallel pumps, the same test was carried out with a single pump, but the same phenomenon was observed. This time, a loud scratching sound was noticed from the runner. The test was therefore aborted.

![Time Serie, RPM Change](image)

**Figure 4.5:** Measurements showing instability after minor drop in rotational velocity.
A mechanical fault was suspected after this test, which needed to be examined by disassembling the turbine. After disconnecting the turbine from the test rig, it was noticed that the runner could not be rotated properly by hand. Also, the gap between the shroud and the turbine casing was significantly enlarged. Further disassembling of the turbine revealed that the runner was detached from the shroud. Scratching marks on the runner (see Figure 4.6) indicated that the runner probably has been rotating freely inside the rotor, and might have caused the limited power generation and vibration. The guide vanes were found to be undesirable flexible, which meant they might have failed to keep the bulb fixed during testing. Accordingly, the hub might have moved in the axial direction and applied a large force on the runner, causing increased friction between the runner and the turbine casing that eventually might have caused the detachment.

![Figure 4.6: Picture of the scratches seen at the turbine after testing.](image)

After this discovery, QRRNT decided to acquire new guide vanes and runner in aluminum, in an attempt to make the turbine operational. This time, the runner was sufficiently attached to the shroud using bolts. Similarly, the guide vanes were fixed to the turbine casing to prevent axial movements. To minimize the influence of the bolts on the flow, a sealing compound was applied on the screw heads to provide a smooth surface. The turbine was then reassembled and was finally operational on the 4th of June.
4.4 Flow Meter and Frequency Transformer Calibration, 07.06.18

After the modified turbine was reassembled, the frequency transformer and ultrasonic flow meter needed to be calibrated to enable digital measurements. The volume flow rate $Q$, rotational velocity $n$, and the power output $P_e$ were calibrated simultaneously as the flow and head were gradually increased. Unfortunately, the output signal for the power output saturated at 6 kW, which prevented calibration of the electrical power output. The measurements indicated oscillating conditions at low volume flow rates, but suddenly a very smooth state was achieved in the turbine. With almost no noise at all compared to the previous test, the turbine was assumed to operate close to the BEP. The head and flow were smoothly increased towards the design point meanwhile calibration points were measured.

Shortly after reaching the designed volume flow rate, at $Q = 0.7 \, \text{m}^3/\text{s}$, $H = 5 \, \text{m}$ and $n = 450 \, \text{rpm}$, a loud noise was noticed from the runner before it suddenly became jammed. The resulting water hammer caused the water to escape through the flange at the inlet before the pump was shut down. The cause of this sudden breakdown was assumed to be mechanical. After dissembling the turbine, it was discovered that the magnets attached to the shroud were moved out of position and jammed the shroud (see Figure 4.7). The reason for the magnet movement might be due to the high rotational velocity that the turbine has never experienced earlier, but should be able to withstand. This fault was once again caused by the poor mechanical design by the turbine producer and prevented further testing.

![Figure 4.7: Picture of the loose magnets between the shroud and windings.](image-url)
Despite the breakdown of the turbine, values for the power, rotational velocity, and the power output were logged during the calibration. Since the head was not measured in this test, a linear relation was assumed between Q and H to estimate the head and calculate the efficiency. The uncertainty is therefore not considered in the resulting performance chart:

Figure 4.8: An estimated performance chart. Uncertainties are not considered.

As can be seen in the figure above, the efficiency was sufficiently reduced compared to the previous test (see Figure 4.3). In addition, the $Q_{ed} - N_{ed}$ line seems to be shifted compared to previously measured points, and it seems to miss the original design point. The main suspected reason for this reduced performance is that the new runner and guide vanes were not exactly equal to the original parts, as they were manufactured by hand. Only a small change in blade angle, can have a significant impact on the velocity triangles of the flow and change the turbine's BEP. During testing, rotation of the flow was observed at the outlet near design point, which strengthens this theory. Also, the bolts and welded joints in the runner might have influenced the flow additionally.

Besides the fact that the turbine geometry seems to be changed, the turbine generated almost half of the expected power at its original BEP, which makes it somewhat unnecessary to fix the turbine and pursue the mapping of its performance. This is up to QRRNT to decide.
Chapter 5

Final Thoughts and Improvements

Despite the many challenges that have occurred during testing, which have limited the performance analysis, some experience has still been gained. Firstly, the original mechanical design of the turbine was not sufficiently robust to withstand the forces that occur in the desired area of application. The detachment of the magnets in the generator is not supposed to happen at all, which indicate a poor manner of fastening by the manufacturer. Additionally, by using 3D-printed guide vanes, the bulb and runner was able to move out of position, supposedly causing detachment of the runner. Therefore, flexible guide vanes should be avoided in future models.

On the other hand, the 3D-printed runner blades were able to withstand the huge loads despite axial movements, cavitation, and large drag forces during the runner jam [18]. If a solution that can ensure the attachment of the runner to the shroud is found, 3D-printed runner blades could be used in future models. However, to obtain more robust turbines, runner blades in metal might be a better solution. QRRNT is already looking into using runner blade modules in future turbines. With this runner design, the blades can be manufactured individually and then mounted into a shroud casing. This solution also simplifies maintenance and makes it easy to test different blade designs.

From the limited test results, it is hard to conclude some immediate weaknesses with the hydraulic design of the turbine. Besides, as the original turbine was never tested at the designed conditions, the CFD-simulations performed by EDR Medeso have not yet been verified. However, inspection of the simulation results revealed some deficiencies with the orig-
• Separation at the trailing edge of the hub: The simulation predicted flow separation right before the downstream end of the hub (after the runner outlet). This separation would cause inefficient backflow downstream of the turbine, which increases the back pressure and reduces the head over the turbine. As mentioned in Chapter 3.3.2, a cone was 3D-printed and was planned to be attached at the end of the hub to test whether this would increase the efficiency, which was predicted by simulation. Such cone could therefore be included in the design of the next turbine to improve its performance. Unfortunately, this was not tested and verified in the laboratory.

• Separation at the trailing edge of the guide vanes: The simulation also predicted circulation in the wake-zone of the guide vanes, which mean leak-flow around the trailing edge from the lower to the upper side of the blades. This can propagate and induce rough flow in the runner, which can cause a drop in lift force and thus decreased efficiency. Separation at the trailing edge of an airfoil is often caused by a wrong thickness distribution along the blade. In the prototype, a typical thickness distribution of a Francis runner was chosen. A possible improvement is to choose a NACA 4-series thickness distribution (streamlined) in future models to avoid separation.

In addition to these simulated deficiencies, the pitch-chord ratio of the runner blades could also be improved in future models. For the prototype, the pitch-chord ratio was chosen to 1 at the hub and 1.1 at the shroud. Contrary, Professor Hermond Brekke recommends a ratio of 0.9 at the hub and 1.1 at the shroud to ensure an average ratio of 1, which is the ideal ratio to obtain maximum lift in the blades [14]. Furthermore, Professor Ole Gunnar Dahlhaug recommends a linear energy distribution in the runner, instead of the nonlinear distribution used in the prototype.

These four mentioned improvements, which are the hub design, thickness distribution of the blades, pitch-chord ratio and the energy distribution, are the main findings in this part of the thesis. In the following part, these aspects will be taken into consideration to obtain an improved design for the next QRRNT turbine prototype.
Part II

Design of a "Scrubber Turbine"
Chapter 6

Design Procedure

This part of the thesis considers the design of a new turbine for the scrubber system of cruise ships (see Section 1.2). The deficiencies and improvements found in Part 1 were taken into consideration in this design process, to obtain a robust and optimized turbine for QRRNT. As explained in Chapter 2.3, the design procedure involves finding the optimal design parameters, evaluating the turbine geometry and then performing CFD-simulations to evaluate the performance of the design and consider adjusting parameters. The process of choosing initial design parameters and setting up the CFD-simulations is presented in this chapter.

6.1 Choosing Design Parameters

The parameters needed to design the new turbine were partly determined by the physical conditions at the ship, and partly optimized through CFD-simulations. The design head, volume flow rate and shroud diameter were fixed from the ship’s requirements. For this specific turbine, the fixed design parameters were as follows:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>*H</td>
<td>18 m</td>
</tr>
<tr>
<td>*Q</td>
<td>0.16 m³/s</td>
</tr>
<tr>
<td>D</td>
<td>0.3 m</td>
</tr>
</tbody>
</table>

Table 6.1: Design conditions in the scrubber system.
Based on these values, the rotational velocity was found in the first step of the developed design method (see Chapter 2.3). By using the *Initializer* MATLAB script, which can be found in Appendix G, the ideal rotational velocity was found to be $n = 1500$ rpm. The hub-shroud ratio was then found to be $d/D = 0.693$, the number of runner blades $Z_b = 8$, and the number of guide vanes was chosen to $Z_{gv} = 17$. The hub design was initially chosen equal to the prototype. The hub length-to-diameter ratio in the prototype was $L_{hub}/D = 0.6$, which correspond to $L_{hub} = 0.18$ m for this turbine. Based on values from literature, the pitch-chord ratio was set to 0.9 at the shroud and 1.1 at the hub, to achieve an ideal average ratio of 1 [14]. Furthermore, a linear energy distribution was chosen for the runner blades, with a NACA 4-series blade thickness distribution to prevent separation. These parameters resulted in the following geometry:

Figure 6.1: 3D-model of the initial geometry. Created in QRRNT’s copyrighted design script.
This design resulted in a reaction ratio of 72.7 %, and expected power output of \( P = 25.4 \, kW \) (by Equation (2.1)). As can be seen in Figure 6.1, the blade outlet angles are quite small. They are on average 12.5 degrees, which mean that slip might occur and must be examined by CFD-simulations.

6.2 CFD-Simulation Setup in ANSYS CFX

After importing the guide vane and runner geometry into ANSYS Workbench, the inlet and outlet flow region needed to be drawn. These domains included a sufficient distance to the inlet and outlet to ensure the most accurate results, in addition to a hub design. They were drawn in sections, such that one inlet region corresponded to a single guide vane section and equally with an outlet region and a runner blade section. Next step was the meshing.

Meshing

The meshing of the guide vanes and runner blades were automatically performed in ANSYS Turbogrid and then checked. The inlet and outlet regions were manually meshed and can be seen in Figure 6.2. As recommended by EDR Medeso, the following mesh types were utilized:

- **Body sizing:** Tetrahedrons of size 0.015 (Path Conforming method)
- **Inflation (boundary layers):** 30 layers with growth rate of 1.15 on walls (hub and shroud)
- **Face sizing:** Extra fine sizing of 0.002 at the turbine outlet/inlet face
- **Sphere of influence (outlet only):** Extra fine sizing of 0.002 the first 0.4 m downstream

(a) Inlet region.  
(b) Outlet region.

Figure 6.2: Mesh of the inlet and outlet region.
Boundary Conditions and Configurations

After successfully meshing all flow domains and defining the different faces, the boundary conditions were specified. The main boundary conditions were:

- **Walls**: No-slip condition on the shroud, hub, runner blades and guide vanes
- **Outlet**: Static back pressure estimated to: $P_s = h_b - NPSH_t$ (expressed in Pa)
- **Inlet**: Mass flow rate boundary ($\dot{m} = \dot{Q} \cdot \rho$) or total pressure boundary ($P_0 = \rho \cdot g \cdot \eta$)
- **Interfaces**: Mixing-plane general connection between the interfaces of the inlet, guide vanes, runner and outlet
- **Runner**: Moving reference system, rotating at $\omega$
- **If axis-symmetric model**: Rotational periodic interface on the sides

Furthermore, the $K - \epsilon$ turbulence model was utilized, together with a first-order high-resolution advection scheme. In order to obtain an accurate solution, the following solver configurations were used in most of the simulations:

<table>
<thead>
<tr>
<th>Setting</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum number of iterations</td>
<td>500-1000</td>
</tr>
<tr>
<td>Conservation target</td>
<td>0.01</td>
</tr>
<tr>
<td>Residual convergence target</td>
<td>$10^{-5}$</td>
</tr>
<tr>
<td>Simulation type</td>
<td>Steady-state</td>
</tr>
<tr>
<td>Timescale</td>
<td>Automatic</td>
</tr>
</tbody>
</table>

Table 6.2: Main settings in CFX Setup.

During simulation, the hydraulic efficiency was set as a monitoring point. After convergence or when a stable solution was achieved, the results were post-processed using streamlines and vector plots to visualize the flow. After a solution is obtained at the design point, a parametric analysis can be used to vary the applied head and the rotational velocity of the turbine to produce a Hill chart, which can be used as an indicator of the turbine’s performance. A MATLAB-script was utilized to produce these Hill charts, and can be found in Appendix C.
Chapter 7

Simulation Results and Discussion

This chapter presents results from CFD-simulations and discusses design adjustments performed to optimize the design. The interpretation of these simulation results must, however, be regarded with caution. As the CFD-simulations of the first prototype have not been validated by experimental tests, the accuracy of the model is unknown. Furthermore, the simulations in this chapter have been performed using the $K - \epsilon$ turbulence model, which is not very accurate in modeling near-wall flow such as flow close to the runner blades and guide vanes. To compensate for this, the mesh was chosen to be very fine close to the walls and blades, to make sure the inner layer is smaller than the boundary layer of the flow.

For the scope of this thesis, the same mesh has been used in all simulations, which makes it hard to conclude whether the solution is mesh-dependent. The uncertainty of the results can therefore not be quantified, but they can still provide useful information. As the simulations have been carried out using a small residual convergence criterion, a large number of iterations and a strict requirement of conservation, the physics can be trusted. Thus, when it comes to evaluating the results, the flow regime can be assumed to be more or less realistic, whereas the derived turbine efficiency has to be considered only as an indicator with unknown accuracy. The majority of the following simulations have been performed by Fredrik Linge from QRRNT, as new design adjustments were made.
Initial Parametric Simulation

To investigate whether the design parameters found in the previous chapter were well chosen, an initial simulation was performed with a range of rotational velocities and applied heads using the same design. An inlet pressure boundary condition was utilized, together with an axis-symmetric model. A Hill chart of the simulation results can be seen in Figure 7.1 under, and the complete results can be found in Appendix E.

![Hill Chart Parametric 1](image)

Figure 7.1: Hill chart from the results of the initial parametric simulation.

As can be seen, the highest hydraulic efficiency was obtained at $n = 1400$ rpm, even though the turbine was designed for 1500 rpm. At this point $H = 19$ m, which is larger than the desired design head. At the design rotational velocity, the streamlines indicated rotation of the outlet flow, which mean that the velocity components were not as designed. This can be caused by non-ideal $^*H$ or $^*n$ values. Thus, in order to move the BEP closer to the design point, the geometry needed to be adjusted. However, an efficiency of 87.9 % is higher than the simulated efficiency of the first prototype, which can indicate that the changing the pitch-chord ratio and the blade energy distribution might have been beneficial to ensure a more efficient flow in the runner.
Main discoveries: The design parameters must be changed to achieve the BEP closer to design point, well choice of pitch-chord ratio and energy distribution.

Simulation Using a Larger Diameter Ratio

The next simulation was therefore carried out using another source to find the optimal hub-shroud ratio. This time, the ratio was calculated to be $d/D = 0.729$ using Figure 9.18 in the referred literature [10]. By using this value, the area between the hub and the shroud was reduced, which increased the axial velocity component. Accordingly, the tangential velocity component was reduced, which increased the average runner blade outlet angle from 13.5 to 14.8 degrees. Using this new geometry and keeping the other parameters constant, a new CFD-simulation was performed using a mass flow boundary condition at the inlet. The simulation result indicated a slightly less efficiency of 87.3 %, which can be a result of the decrease in the tangential velocity component as $\eta_h = \frac{U_C n_s}{g H}$. However, the flow seemed to follow the lower blade surface, which indicates minimal slip tendencies (see figure under).

Figure 7.2: Close-up view of the flow in a single guide vane and runner blade.
The vector plot shows that the flow attacks the blades at desired angles with almost no rotation at the outlet, which indicate nearly ideal velocity triangles. Furthermore, the pressure gradient along the runner blades (see Figure 7.3a) showed a smooth pressure distribution, which indicates that the NACA thickness was a smart choice to ensure maximum lift, and thus torque. The shape of the pressure gradient along the upper and lower surface of the blade (see Figure 7.3b) is almost identical to an ideal pressure distribution of a NACA 4-series airfoil, except for minor oscillations assumed caused by turbulence at the runner inlet [8].

(a) Pressure gradient contours along a single runner blade.  
(b) Plot showing static pressure along the center line of the upper and lower blade surface.

Figure 7.3: Pressure distribution of runner blade.

Even though the blade design seems to work, further inspections of the outlet region revealed an undesired vortex at the hub (see Figure 7.4). This vortex might affect the efficiency, as it can increase the back pressure in the outlet region and thus decrease the head.

(a) Streamlines at the outlet revealing a vortex.  
(b) Shape of the original hub design.

Figure 7.4: Simulation of the outlet region using the original hub.

**Main discoveries:** NACA thickness is favorable, the original hub design causes a vortex.
Simulation of Different Hub Variations

In order to improve the secondary flow at the outlet, another simulation was carried out using a smooth rear hub, which is equivalent to attaching a cone at the outlet (see Chapter 3.3.2). The other parameters remained unchanged to investigate the effect of the hub only. The simulation indicated a slightly decreased efficiency of 87.2 %. By inspection (see Figure 7.5), a large vortex could be seen at the hub, which might have caused the efficiency drop.

This result was quite surprising, as a round shape of the rear hub was expected to increase the efficiency, which was simulated for the prototype in Part 1 of this thesis. However, as the average axial velocity in this turbine is 13 % less than in the prototype, another hub design might be needed to avoid flow separation. Based on literature regarding blade-induced vortices, a sharp cone design with $L_{hub} = 0.125 \, m$ was utilized in the next simulation [12]. By using this hub design, the resulting backflow was improved, which can be seen in Figure 7.6 under:

![Streamlines showing a large vortex.](image1)

![Shape of the "cone" hub.](image2)

Figure 7.5: Simulation of the outlet region with a "cone" hub.

![Streamlines showing a smooth outlet flow.](image3)

![Shape of the redesigned hub.](image4)

Figure 7.6: Simulation of the outlet region with redesigned hub.
This design indicated an increased hydraulic efficiency of 87.5 % and was therefore utilized in all subsequent simulations. On the other hand, the effects of the hub design were quite small and might be negligible in reality.

**Main discovery:** A sharp-ended hub ensures a smooth outlet flow.

### Parametric Simulation Using a Larger Diameter Ratio

To investigate the performance of the modified design outside the design point, another parametric simulation was performed. This time, a pressure gradient was utilized as the boundary condition instead of mass flow at the inlet. The simulation resulted in the Hill chart that can be seen in Figure 7.7, and the complete results can be found in Appendix E.

![Hill Chart Parametric 2](image)

Figure 7.7: Hill chart from the results of the new parametric simulation.

This design obtained a BEP closer to the design point, compared to the initial design (see Figure 7.1), which might be due to more optimal velocity triangles, as the reaction ratio was changed from 76.2 % to 75.8 % with the enlarged hub diameter. However, the maximum efficiency was decreased, supposedly because of the reduced tangential velocity component,
as mentioned. The original hub-shroud ratio might therefore be a better choice. These re-
sults also show that using a pressure boundary condition at the inlet instead of mass flow
decreases the efficiency additionally. This can be because a different volume flow rate is
achieved using a pressure difference, which gives inefficient flow in the runner as the veloc-
ity components not become idle.

**Main discovery:** A \( \frac{d}{D} \) ratio of 0.693 gives better performance.

**Parametric Simulation with Increased Number of Guide Vanes**

To investigate the effect of the number of guide vanes, another simulation was performed
using 19 guide vanes instead of 17. The hub and runner design remained unchanged. This
modification resulted in a decrease in the maximum efficiency to 86.9 \%, which might have
been caused by a reduction in volume flow through the turbine that affected the velocity
triangles and causing rotation at the outlet. Therefore, \( Z_{gv} = 17 \) was found as the optimal
amount and was utilized in all future simulations.

**Main discovery:** Choosing 17 guide vanes instead of 19, gives better performance.

**Parametric Simulations with Hill Chart Correction**

By going back to the initial \( \frac{d}{D} \) ratio, another adjustment needed to be tried to obtain a de-
sign that ensures a BEP closer to designed conditions. Therefore, a straight line was drawn
between the BEP and design point in Figure 7.1 and extrapolated an equal distance, which
gave \( Q_{ed} = 0.137 \) and \( N_{ed} = 37.13 \). By using these values to redesign the geometry, the new
design parameters became * \( n = 1600 \) and * \( H = 17 \ m \). This design would hopefully shift the
BEP closer to the design point.

However, the results indicated an even lower performance of 86.3 \%, and the BEP shifted
further away from the design point. This shift might be because the \( \frac{U}{C_2} \)-ratio was in-
creased, which result in a smaller efficiency by Euler’s equation (Equation (2.10)). There-
fore another simulation was performed, by extrapolating in the other direction and using a design head of 19 meters instead. The resulting Hill chart follows (full results in Appendix E):

![Hill Chart Parametric 5](image)

Figure 7.8: Hill chart from the results of parametric simulation of the $N_{ed}$-corrected design.

As seen in the figure, this attempt was unsuccessful, as also this design shifted the BEP further away from the design point. On the other hand, the efficiency was slightly increased to 86.8%, which might be due to the $U/C_2$-ratio of 2.09, which is closer to the ideal value of 2 [13]. Thus, another design change was needed to increase the efficiency and move the BEP.

**Main discoveries:** Changing design head and rotational velocity decreases the efficiency.

**Mass Flow Boundary Simulation of Initial Design**

Since all the performed changes in geometry failed to move the BEP closer to the design point, a further analysis was done on the first simulation as it resulted in the highest efficiency. After using the simulated results to calculate the actual volume flow rate and head in the turbine, it was discovered that the values appeared to be smaller than the simulation output values. The actual volume flow rate was found to be $Q = 0.14 \text{ m}^3/\text{s}$, which explains
why the BEP did not occur at the design point. Therefore, a new simulation was performed on the initial design using a mass flow rate boundary at the inlet to enforce the desired volume flow rate of $Q = 0.16 \, m^3/s$ at an 18-meter head. The results showed that the BEP was successfully achieved as designed, which can be seen in the figure under:

![Parametric Plot 6](image)

Figure 7.9: Hill chart from the results of the parametric simulation of the initial design, using a mass flow boundary condition.

These results indicate that the method used to find the design parameters is very accurate in predicting its BEP at designed conditions. On the other hand, a mass flow boundary condition is not a realistic model of a real system. As the turbine will be applied in a hydraulic system that is open to the atmosphere at an 18-meter altitude, a pressure boundary is more accurate in simulating the physics. The final turbine design should therefore be able to ensure a $0.16 \, m^3/s$ flow rate using a total pressure boundary corresponding to a head of 18 meters. As the volume flow rate in the initial simulation was smaller than this, the geometry needed to be changed. A new geometry with a higher design volume flow rate might be the solution as this increases the blade angle, which ensures more flow through the turbine.

**Main discovery:** The initial design achieves the BEP at enforced design conditions, but it must be modified to ensure a higher volume flow with a pressure boundary condition.
Final Design by Volume Flow Modification

Another geometry was therefore computed, by using the initial hub-shroud ratio and increasing the design volume flow rate. This time, a pressure difference boundary condition was used in the simulation. First, a design flow of 0.17 m$^3$/s was tried, but the simulation indicated a too large volume flow rate through the turbine. After a few iterations of changing the design volume flow rate, the nearly optimal value was found to be $^*Q = 0.168$ m$^3$/s. The resulting design was successfully able to ensure 0.161 m$^3$/s at an efficiency of 88.5%.

Next, a parametric simulation was performed to investigate the turbine's performance over a range of operating conditions and to locate its BEP. The complete results can be found in Appendix E. The resulting Hill chart (see Figure 7.10) shows that the BEP is very close to the design point. Even though the BEP occurred at $H = 17$ m, it is not necessary to change the design, as the efficiency is almost identical at 18 m. The $Q_{ed} - N_{ed}$ relation is also quite similar to the measured results in Chapter 4, which make these results more reliable than the previous simulations.

![Figure 7.10: Hill chart from parametric simulation of the *Q - modified design.](image)

After successfully passing all design requirements, a final full 3D transient simulation was
required to verify the turbine’s performance using a more realistic model and make sure no transient flow phenomena occur.

**Main discovery:** After changing the design flow rate the BEP was obtained as designed.

### Final 3D Simulation

As a final verification of the turbine design, a 3D transient (time variant) simulation was initiated at the designed conditions. To obtain robust initial values for the transient part, a full 3D steady-state simulation was initially performed. This simulation resulted in almost identical results as in the axis-symmetric model. It indicated a hydraulic efficiency of 88.3 % and a volume flow rate of 0.160 \( m^3/s \), which is exactly as designed. The following streamline plot under shows the flow through the turbine:

![Streamlines of the turbine flow in the steady-state simulation.](image)

The streamlines show a smooth flow through the guide vanes and runner, and a very efficient backflow, as the flow seems to meet at the tip of the hub and flows smoothly downstream. However, the flow at the shroud seems to be rotating, which seems to be induced by
the rotating shroud in the runner. A closer examination of the flow around the blades indicates a smooth and efficient flow along the guide vanes and runner blades, with no apparent slip or separation tendencies to be seen:

Figure 7.12: Vector plot of the flow in the guide vanes and runner blades in the steady-state simulation.

Despite these promising results, this stationary 3D simulation had difficulties converging, which can be caused by transient phenomena such as cavitation or a vortex. The oscillating residuals might indicate that a stationary solution is unrealistic due to these time-dependent phenomena. This remains to be investigated with a transient simulation.
Chapter 8

Validation of Method and Final Design

After optimizing the design volume flow rate, the d/D ratio, the hub and the number of guide vanes, a final optimized design was found. As was seen in the parametric simulation using a mass flow boundary condition to enforce the desired flow, the method used to find the design parameters, successfully predicted values that resulted in a BEP precisely as designed. It can therefore be concluded that the method is valid and can be used as an additional tool to design future turbines for QRRNT. However, for the systems that QRRNT will be using the turbines, it is more realistic to perform simulations using a pressure boundary condition, as they are open systems. Therefore, to ensure enough flow through the turbine, the design volume flow rate might need to be adjusted by manual iteration based on simulation results, as was done in this design process. Nevertheless, the method is still useful to choose design parameters.

The final design was successfully able to ensure the designed volume flow rate through the turbine, at a maximum hydraulic efficiency of 88.3 %. As state-of-the-art bulb turbines typically have a maximum hydraulic efficiency of around 90 %, this result is quite good considering that the design conditions are not ideal. At design point, the reaction ratio of the turbine is approximately 75 %, which is somewhat higher than the ideal 50 %. This large ratio is mainly due to the high head in the scrubber system. The average angle of the runner blade inlet is 22.3 degrees and 13 degrees at the outlet, due to a high design rotational velocity of 1500 rpm. The velocity triangles of the turbine can be seen in Figure 8.1.
As was discovered from the CFD-simulations, the ideal number of blades was 17 guide vanes and 8 runner blades. Besides, the initial d/D ratio of 0.693 resulted in the highest efficiency. It was also discovered that a NACA thickness distribution prevented separation on the guide vanes and runner blades. A sharp-ended hub was found to provide a smooth flow at the turbine outlet and was chosen for the final design. A 3D-model of the turbine can be seen in the figure under:

![Turbine Model](image)

**Figure 8.1: Velocity triangles in the turbine at design point.**

In this design however, the guide vanes are unrealistically thin, which would make them impossible to manufacture. Thus, the thickness of runner blades and guide vanes remains to be optimized. Furthermore, it was not enough time to perform a mesh analysis to quantify the uncertainty of the simulations. Also, a transient simulation was not performed, which is essential to validate the design using a more realistic simulation.
Part III

Final Conclusions
Chapter 9

Conclusion

The primary goals for this thesis have been first to evaluate the performance of the prototype turbine based on efficiency measurements from tests, and secondly to design and optimize a new turbine for scrubber systems using CFD-simulations. The practical tests were mostly performed initially in this project to find improvements that were then implemented in the new design. Next, the incremental design procedure was initiated and completed.

During the practical testing, several challenges occurred due to the poor mechanical design of the prototype, which eventually caused the runner and the generator magnets to detach from the shroud. The reason for the loosening of the runner was supposedly due to the flexibility of the 3D-printed guide vanes, which enabled the runner to move in the axial direction causing a large stress on the glued runner. Therefore, QRRNT acquired a new runner and guide vanes in aluminum to pursue the tests. However, during calibration, the magnets in the generator detached from the shroud and jammed the runner, which prevented further testing. The measurements from this session showed a decreased performance, with a maximum efficiency of 50%. This was assumed to be caused by the new blades not being exactly equal to the original design, which changed the velocity triangles and decreased the turbine’s hydraulic efficiency. Due to the poor performance at designed conditions, further mapping of its performance might not be necessary as the original design is changed. Thus, the testing was concluded, and the measurements from the previous test considered the main results.

Based on these limited tests results and examination of the CFD-simulation of the prototype, improvements were found to enhance the performance of future models. To improve
the mechanical design QRRNT is looking into using runner blade modules to prevent using 3D-printed parts, and find a new solution to attach the magnets to the shroud. Regarding the hydraulic design, it was discovered from simulation results that flow separation was occurring on the hub and leak flow occurring around the trailing edge of the guide vanes. Also, the energy distribution and pitch-chord ratio of the prototype were not optimal. Enhancing modifications were therefore performed in the design of the "scrubber turbine".

The new turbine was designed using the developed method to determine the main design parameters. By using CFD-simulations to estimate the turbine's performance, it was found that the method successfully predicted design parameters that ensured a BEP as designed. However, when using a more realistic pressure boundary condition in the simulations, the desired volume flow was not obtained, which caused the BEP to not occur as designed. Therefore, several failed geometry changes were attempted, using an increased hub-shroud ratio and Hill chart corrections. The final solution was to increase the design volume flow rate by iteration, which resulted in a high performance as designed.

Simulations of the final design indicate a hydraulic efficiency of 88.3 %, which is close to state-of-the-art and quite good considering that the applied conditions are not ideal for a Kaplan turbine. Furthermore, the design ensures a smooth backflow due to an optimized hub. Due to the high pressure in the system, the design rotational velocity was set to 1500 rpm to prevent cavitation and ensure decent velocity components. A linear energy distribution was utilized for the runner blades, and a NACA 4-series thickness distribution prevents flow separation.

To summarize this thesis, it was measured from experimental tests that the original prototype turbine was able to generate 17-18 kW of power at a maximum efficiency of approximately 72 %. A partly Hill chart showing its performance was created, but the turbine broke down before new measurements were taken. In addition, a "scrubber turbine" was designed and optimized, and simulations predict a hydraulic efficiency of 88.3 %, which is higher than the simulated and measured performance of the prototype. The goals of this thesis were therefore achieved, although unplanned events limited the performance mapping of the prototype. However, there is some future work required to establish a final design.
Chapter 10

Future Work

Despite all the work done in this thesis, some aspects could be further improved if more time was available. When it comes to the prototype testing, future work will be to either find a solution to fasten the magnets and pursue the mapping of the turbine’s performance or conclude with a failed design and manufacture a new improved prototype to test. If the turbine is repaired, the other planned performance test using a cone at the hub could be performed, which is necessary to verify the CFD-simulation (see Chapter 3.3.2). However, with the poor performance of the turbine, acquiring a new turbine might be a better use of resources than spending more time on the failed prototype. This is up to QRRNT to decide.

Besides, future work for the second part of the thesis includes performing a transient simulation to study the turbine’s transient behavior, performing simulations using another mesh to quantify the uncertainty and to optimize the maximum thickness of the runner blades and guide vanes. By performing the same simulations with a different mesh, the uncertainty of the results can be quantified, which makes the results more reliable. Furthermore, the simulations performed in this thesis utilized only default values for the maximum thickness of the guide vanes and runner blades. Different values could have been tested, to find optimal values. However, there was not enough time available for this.

Finally, if the design of the ”scrubber turbine” had been developed earlier, the turbine could have been manufactured and tested in the laboratory. This remains to be done, in order to verify the CFD-simulations performed in the second part of this thesis.
Bibliography


Appendices
Appendix A

List of Figures and Tables

List of Figures

1.1 Illustration of the QRRNT concept on a well boat. . . . . . . . . . . . . . . . . . . 4
1.2 Illustration of the QRRNT turbine in scrubber systems. . . . . . . . . . . . . . . . 5
2.1 Hill chart for a Kaplan turbine with $\Omega=1.3$ [10]. $\phi$ is the runner blade angle and $S_0$ is the guide vane angle, which both are fixed in QRRNT’s turbines. . . . . . . 9
2.2 Illustration of a Straflo Turbine. The guide vanes are marked in green, and the runner and generator marked in red [5]. . . . . . . . . . . . . . . . . . . . . . . . . 10
2.3 Illustration of a guide vane and runner blade, and the velocity components . . . 11
2.4 Cascade effects in Kaplan turbines. . . . . . . . . . . . . . . . . . . . . . . . . . . 13
2.5 Float chart of the design method. . . . . . . . . . . . . . . . . . . . . . . . . . . . . 15
2.6 Float chart of the iterative method used to find $^\ast n$. . . . . . . . . . . . . . . . 16
2.7 Chart used to find the number of runner blades [19]. . . . . . . . . . . . . . . . . 16
2.8 Illustration of slip, which is deflection of the outlet flow. $\beta_3$ is the actual angle of the flow, and $\beta_{3\infty}$ is the outlet angle of the blade. . . . . . . . . . . . . . . . . . . . . . . . 19
3.1 Sketch of the test rig, showing the main dimensions in mm. . . . . . . . . . . . . 21
3.2 3D model of the test rig with the turbine seen in turquoise. .......................... 22
3.3 3D model of the hydraulic system in the laboratory. The test rig is marked in red. 23
3.4 Picture of the test rig showing ring manifolds, the downstream valve, and the frequency transformer. ................................................................. 24
3.5 Picture if the 3D-printed cone attached to the rear end of the hub. .................. 26
4.1 Measured power versus head and theoretical value. ........................................ 28
4.2 Total measured efficiency versus the speed factor. ........................................ 28
4.3 An intuitive performance chart from the test. .................................................. 29
4.4 Sudden head drop observation from LabView. ............................................... 30
4.5 Measurements showing instability after minor drop in rotational velocity. ....... 31
4.6 Picture of the scratches seen at the turbine after testing. .............................. 32
4.7 Picture of the loose magnets between the shroud and windings. .................... 33
4.8 An estimated performance chart. Uncertainties are not considered. ............... 34
6.1 3D-model of the initial geometry. Created in QRRNT’s copyrighted design script. 39
6.2 Mesh of the inlet and outlet region. ............................................................. 40
7.1 Hill chart from the results of the initial parametric simulation. ......................... 43
7.2 Close-up view of the flow in a single guide vane and runner blade. ................. 44
7.3 Pressure distribution of runner blade. .......................................................... 45
7.4 Simulation of the outlet region using the original hub. ................................... 45
7.5 Simulation of the outlet region with a “cone” hub. ...................................... 46
7.6 Simulation of the outlet region with redesigned hub. ................................... 46
7.7 Hill chart from the results of the new parametric simulation. ......................... 47
7.8 Hill chart from the results of parametric simulation of the $N_{eq}$-corrected design 49
7.9 Hill chart from the results of the parametric simulation of the initial design, using a mass flow boundary condition. .................................................... 50
7.10 Hill chart from parametric simulation of the *Q*-modified design. .................. 51
7.11 Streamlines of the turbine flow in the steady-state simulation. ...................... 52
7.12 Vector plot of the flow in the guide vanes and runner blades in the steady-state simulation. ................................................................. 53
8.1 Velocity triangles in the turbine at design point. .......................................... 55
8.2 The final "scrubber turbine" design

55
## List of Tables

<table>
<thead>
<tr>
<th>Table</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.1</td>
<td>Sensors used in experiments</td>
<td>24</td>
</tr>
<tr>
<td>3.2</td>
<td>Planned test parameters for the performance test</td>
<td>25</td>
</tr>
<tr>
<td>6.1</td>
<td>Design conditions in the scrubber system</td>
<td>38</td>
</tr>
<tr>
<td>6.2</td>
<td>Main settings in CFX Setup</td>
<td>41</td>
</tr>
<tr>
<td>B.1</td>
<td>IEC defined calibration errors</td>
<td>VI</td>
</tr>
<tr>
<td>B.2</td>
<td>Calibration uncertainties for the absolute pressure sensor at the turbine inlet</td>
<td>VI</td>
</tr>
<tr>
<td>B.3</td>
<td>Calibration uncertainties for the absolute pressure sensor at the turbine outlet</td>
<td>VII</td>
</tr>
<tr>
<td>B.4</td>
<td>Calibration uncertainties for the differential pressure sensor over the turbine</td>
<td>VII</td>
</tr>
<tr>
<td>B.5</td>
<td>Calibration uncertainty for the ultrasonic volume flow meter</td>
<td>VII</td>
</tr>
<tr>
<td>B.6</td>
<td>Uncertainty of the temperature meter</td>
<td>VIII</td>
</tr>
<tr>
<td>B.7</td>
<td>Uncertainty of the rotational velocity signal from the frequency transformer</td>
<td>VIII</td>
</tr>
<tr>
<td>B.8</td>
<td>Total uncertainties of parameters. *Estimated values due to manual measurement</td>
<td>XI</td>
</tr>
<tr>
<td>D.1</td>
<td>Test results from the performance test</td>
<td>XVIII</td>
</tr>
<tr>
<td>D.2</td>
<td>Estimated results from the frequency transformer calibration</td>
<td>XIX</td>
</tr>
<tr>
<td>E.1</td>
<td>Results from initial parametric simulation</td>
<td>XX</td>
</tr>
<tr>
<td>E.2</td>
<td>Results from the second parametric simulation</td>
<td>XXI</td>
</tr>
<tr>
<td>E.3</td>
<td>Parametric simulation results of the Hill chart corrected design</td>
<td>XXII</td>
</tr>
<tr>
<td>E.4</td>
<td>Parametric simulation results of the final design</td>
<td>XXII</td>
</tr>
<tr>
<td>G.1</td>
<td>Summary of MATLAB scripts and functions</td>
<td>LIV</td>
</tr>
</tbody>
</table>
Appendix B

Calibration and Uncertainty Calculations

In order to translate electrical signals into physical values, all sensors need to be calibrated. This is done by assuming a linear relationship between the sensor output voltage $V$ and the physical value and then finding the best regression line. The scaled signal comes with a corresponding error $\Delta x$, which is the difference between the measurement and the actual value of the quantity. The uncertainty of the measurement is defined as "range within which the true value of a measured quantity can be expected to lie, with a suitable high probability" [1]. This chapter considers calibration of the different sensors utilized in the laboratory testing, and uncertainty calculations of derived parameters and experimental measurements.

B.1 Sensor Calibration

There are different sources of error that must be considered when calibrating instruments. These are defined by IEC and are shown in Table B.1. The total uncertainty of an instrument can be calculated using the Root-Sum-Square (RSS) - method, to combine all sources of error [3].
Uncertainty Description (source)

<table>
<thead>
<tr>
<th>Uncertainty</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>±fx_a</td>
<td>Systematic error of the primary calibration method</td>
</tr>
<tr>
<td>±fx_b</td>
<td>Random error of the primary calibration method</td>
</tr>
<tr>
<td>±fx_c</td>
<td>Systematic error of the secondary instrument</td>
</tr>
<tr>
<td>±fx_d</td>
<td>Random error of the secondary calibration method</td>
</tr>
<tr>
<td>±fx_e</td>
<td>Physical phenomena and external influences</td>
</tr>
<tr>
<td>±fx_f</td>
<td>Error in physical properties</td>
</tr>
</tbody>
</table>

Table B.1: IEC defined calibration errors.

In the following sensor calibrations, it is assumed that the random error in the voltage measurements is Student’s *t*-distributed, and the mean values are found using a 95% confidence interval. Furthermore, negligible external and physical influences are assumed. Then, using the RSS-method, the maximum uncertainty for each sensor becomes:

\[
max(f_x) = \pm \sqrt{(fx_{ab})^2 + (max(f_{s\text{regression}}))^2}
\]  

(B.1)

The \(max(f_{s\text{regression}})\) of this expression is the maximum regression uncertainty, which is calculated using the method in Appendix C.3 of Bjørn Solemslie’s compendium [3]. In the following sensor calibrations, the systematic and random error of the instrument is included in the regression uncertainty. The pressure transducers were calibrated using a calibration program in LabView (see Appendix ??), whereas the other sensors were calibrated manually using the created *Calibration.m*- MATLAB script (see Appendix G). Calibration reports for the following uncertainty calculations can be found in Appendix C.

### Inlet Pressure Sensor

<table>
<thead>
<tr>
<th>Uncertainty</th>
<th>Description</th>
<th>Magnitude</th>
</tr>
</thead>
<tbody>
<tr>
<td>±fp_{in,a}</td>
<td>Systematic uncertainty in deadweight calibrator</td>
<td>±0.008%</td>
</tr>
<tr>
<td>±fp_{in,regression}</td>
<td>Maximum regression uncertainty</td>
<td>±0.006472%</td>
</tr>
</tbody>
</table>

Table B.2: Calibration uncertainties for the absolute pressure sensor at the turbine inlet.

By adding these together, the total calibration uncertainty becomes:

\[
max(fp_{in}) = \pm \sqrt{(0.008)^2 + (0.006472)^2} = \pm 0.01029\%
\]

(B.2)
Outlet Pressure Sensor

<table>
<thead>
<tr>
<th>Uncertainty</th>
<th>Description</th>
<th>Magnitude</th>
</tr>
</thead>
<tbody>
<tr>
<td>±$f_{P_{out,a}}$</td>
<td>Systematic uncertainty in deadweight calibrator</td>
<td>±0.008%</td>
</tr>
<tr>
<td>±$f_{P_{out,regression}}$</td>
<td>Maximum regression uncertainty</td>
<td>±0.006767%</td>
</tr>
</tbody>
</table>

Table B.3: Calibration uncertainties for the absolute pressure sensor at the turbine outlet.

By adding these together, the total calibration uncertainty becomes:

$$max(f_{P_{out}}) = \pm \sqrt{(0.008)^2 + (0.006767)^2} = \pm 0.01048\%$$  \hspace{1cm} (B.3)

Differential Pressure Sensor

<table>
<thead>
<tr>
<th>Uncertainty</th>
<th>Description</th>
<th>Magnitude</th>
</tr>
</thead>
<tbody>
<tr>
<td>±$f_{P_{diff,a}}$</td>
<td>Systematic uncertainty in deadweight calibrator</td>
<td>±0.008%</td>
</tr>
<tr>
<td>±$f_{P_{diff,regression}}$</td>
<td>Maximum regression uncertainty</td>
<td>±0.028571%</td>
</tr>
</tbody>
</table>

Table B.4: Calibration uncertainties for the differential pressure sensor over the turbine.

By adding these together, the total calibration uncertainty becomes:

$$max(f_{P_{diff}}) = \pm \sqrt{(0.008)^2 + (0.028571)^2} = \pm 0.02967\%$$  \hspace{1cm} (B.4)

Ultrasonic Flow Sensor

<table>
<thead>
<tr>
<th>Uncertainty</th>
<th>Description</th>
<th>Magnitude</th>
</tr>
</thead>
<tbody>
<tr>
<td>±$f_{Q,a}$</td>
<td>Systematic uncertainty in electromagnetic flow meter</td>
<td>±1.00%</td>
</tr>
<tr>
<td>±$f_{Q,regression}$</td>
<td>Maximum regression uncertainty</td>
<td>±0.27%</td>
</tr>
</tbody>
</table>

Table B.5: Calibration uncertainty for the ultrasonic volume flow meter.

By adding these together, the total calibration uncertainty becomes:

$$max(f_{Q}) = \pm \sqrt{(1.00)^2 + (0.27)^2} = \pm 1.036\%$$  \hspace{1cm} (B.5)
Temperature Sensor

<table>
<thead>
<tr>
<th>Uncertainty</th>
<th>Description</th>
<th>Magnitude</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\pm f_{T,a}$</td>
<td>Systematic uncertainty in the temperature meter</td>
<td>$\pm 0.50%$</td>
</tr>
</tbody>
</table>

Table B.6: Uncertainty of the temperature meter.

Since no calibration reports was found for the temperature sensor, an uncertainty of 0.5 % was assumed. Thus:

$$\text{max}(f_T) = \pm 0.50\%$$  \hspace{1cm} (B.6)

Frequency Transformer

Rotational Velocity

<table>
<thead>
<tr>
<th>Uncertainty</th>
<th>Description</th>
<th>Magnitude</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\pm f_{n,\text{regression}}$</td>
<td>Maximum regression uncertainty</td>
<td>$\pm 0.39%$</td>
</tr>
</tbody>
</table>

Table B.7: Uncertainty of the rotational velocity signal from the frequency transformer.

Similarly for the rotational velocity, the calibration uncertainty becomes:

$$\text{max}(f_n) = \pm 0.39\%$$  \hspace{1cm} (B.7)
APPENDIX B. CALIBRATION AND UNCERTAINTY CALCULATIONS

B.2 Uncertainty of Derived Values

When measurements from the sensors are used to calculate new parameters, the uncertainty propagates and must be calculated using the Gauss’ law of error propagation. For a variable \( Y = Y + \Delta Y \), which is a function of \( x_1, x_2, ..., x_n \) such that \( Y = y(x_1 \pm \Delta x_1, x_2 \pm \Delta x_2, ..., x_n \pm \Delta x_n) \), the error \( \Delta Y \) caused by the individual errors \( \Delta x_i \) is given by [3]:

\[
\Delta Y = \pm \sqrt{\left( \frac{\partial y}{\partial x_1} \Delta x_1 \right)^2 + \left( \frac{\partial y}{\partial x_2} \Delta x_2 \right)^2 + \cdots + \left( \frac{\partial y}{\partial x_n} \Delta x_n \right)^2}
\]  

(B.8)

If the function is on the form \( y = \frac{x_1^a x_2^b}{x_3^c x_4^d} \), the equation simplifies to:

\[
\Delta Y = \pm \sqrt{(a \cdot \Delta x_1)^2 + (b \cdot \Delta x_2)^2 + \cdots + (z \cdot \Delta x_n)^2}
\]  

(B.9)

The uncertainty is then given by: \( f_Y = \frac{\Delta Y}{Y} \). In the following uncertainty calculations, the uncertainty of density of water is assumed to be negligible. Even though IEC states that the systematic error of estimating the density is in the order of 0.01 %, it is due to Professor Pål-Tore Selbo, not significant when calculating the uncertainty of derived variables [2].

Pressure Head

Head equation (Equation (2.2)): \( H = \frac{\Delta P}{\rho g} + \frac{Q^2}{2g} \left( \frac{1}{x_{in}} - \frac{1}{x_{out}} \right) = \frac{\Delta P}{\rho g} + \frac{Q^2}{2ag} \)

Define \( a = \frac{A_{out} A_{in}^2}{A_{out}^2 - A_{in}^2} \)

Note: \( P \) is the differential pressure over the turbine

Error equation:

\[
\Delta H = \pm \sqrt{\left( \frac{\partial H}{\partial \Delta P} \right)^2 + \left( \frac{\partial H}{\partial Q} \Delta Q \right)^2} = \pm \sqrt{\left( \frac{1}{\rho g} \Delta P \right)^2 + \left( \frac{Q}{g a} \Delta Q \right)^2}
\]

Uncertainty equation:

\[
f_H = \frac{\Delta H}{H} = \pm \sqrt{\left( \frac{1}{\rho g} \Delta P \right)^2 + \left( \frac{Q}{g a} \Delta Q \right)^2} = \pm \sqrt{\frac{1}{\rho g^2} \frac{\Delta P^2}{P^2} + \frac{Q^2}{g a^2} \frac{\Delta Q^2}{Q^2}} = \pm \sqrt{\frac{1}{\rho g^2} \frac{\Delta P^2}{P^2} + \frac{Q^2}{g a^2} \frac{\Delta Q^2}{Q^2}}
\]
APPENDIX B. CALIBRATION AND UNCERTAINTY CALCULATIONS

\[ \eta = \pm \sqrt{\frac{P_2 a^2 (f_P)^2 + Q_2 P^2 (f_Q)^2}{\left(\frac{Q_2 P}{2} + Pa\right)^2}} = \pm \sqrt{\left(\frac{Pa}{\frac{Q_2 P}{2} + Pa}\right)^2 f_P^2 + \left(\frac{Q_2 P}{\frac{Q_2 P}{2} + Pa}\right)^2 f_Q^2} \]

\[ = \pm \sqrt{\left(1 + \frac{Q_2 P}{Pa} + 1\right)^2 f_P^2 + \left(\frac{1}{Q_2 P}\right)^2 f_Q^2} \]

Total Efficiency

Total efficiency equation (Equation (2.4)):

\[ \eta = \frac{P_e}{\rho Q H} = \frac{P_e}{Q(P_{diff} + \frac{1}{2} Q^2)} = \frac{P_e}{Q(P_{diff} + kQ)} \]

Define \( k = \frac{Q}{2a} \), where \( a = \frac{(A_{out} - A_{in})}{2} \)

Note: From this point, \( P \) is the differential pressure over the turbine

Error equation:

\[ \Delta \eta = \pm \sqrt{\left(\frac{\partial \eta}{\partial P_e} \Delta P_e\right)^2 + \left(\frac{\partial \eta}{\partial Q} \Delta Q\right)^2 + \left(\frac{\partial \eta}{\partial P} \Delta P\right)^2} = \pm \sqrt{\left(\frac{1}{QP + kQ^2} \Delta P_e\right)^2 + \left(\frac{P_e (3kQ^2 + P)}{(QP + kQ^2)^2} \Delta Q\right)^2 + \left(\frac{P_e Q}{(QP + kQ^2)^2} \Delta P\right)^2} \]

Uncertainty equation:

\[ f_{\eta} = \frac{\Delta \eta}{\eta} = \pm \sqrt{\left(\frac{1}{QP + kQ^2} \Delta P_e^2 + \left(\frac{P_e (3kQ^2 + P)}{(QP + kQ^2)^2} \Delta Q\right)^2 + \left(\frac{P_e Q}{(QP + kQ^2)^2} \Delta P\right)^2\right)} = \]

\[ = \pm \sqrt{\left(\frac{\Delta P}{P_e}\right)^2 + \left(\frac{3kQ^2 + P}{(QP + kQ^2)^2}\right) (\Delta Q)^2 + \left(\frac{Q^2}{(QP + kQ^2)^2}\right) (\Delta P)^2} \]

\[ f_{\eta} = \pm \sqrt{f_P^2 + \left(\frac{3kQ^2 + P}{(P + kQ)^2}\right) f_Q^2 + \left(\frac{1}{1 + \frac{kQ}{P}}\right)^2 f_P^2} \]
B.3 Total Uncertainties in Tests

In this section, the uncertainties of the measurements from the test are presented and used to calculate the total uncertainties using the RSS-method. These are found using a 95 % confidence interval, and by using the standard deviation from the mean of each measuring point as an estimation of the variance. Equation (26) in Appendix C.3 of Bjørn Solemslie’s compendium was utilized to find the uncertainty [3].

Performance Test 24.01.2018

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Calibration uncertainty</th>
<th>Mean Measured Uncertainty</th>
<th>Total Mean Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_{in}$</td>
<td>±0.01029%</td>
<td>±0.235%</td>
<td>±0.236%</td>
</tr>
<tr>
<td>$P_{out}$</td>
<td>±0.01048%</td>
<td>±0.070%</td>
<td>±0.072%</td>
</tr>
<tr>
<td>$P_{diff}$</td>
<td>±0.02967%</td>
<td>±0.026%</td>
<td>±0.039%</td>
</tr>
<tr>
<td>$Q^*$</td>
<td>±3%</td>
<td>±0.009%</td>
<td>±3%</td>
</tr>
<tr>
<td>$T$</td>
<td>±0.050%</td>
<td>±0.002%</td>
<td>±0.050%</td>
</tr>
<tr>
<td>$n^*$</td>
<td>±5%</td>
<td>±5%</td>
<td>±5%</td>
</tr>
<tr>
<td>$P_e^*$</td>
<td>±5%</td>
<td>±5%</td>
<td>±5%</td>
</tr>
<tr>
<td>$H^*$</td>
<td>–</td>
<td>±6%</td>
<td>±6%</td>
</tr>
<tr>
<td>$\eta^*$</td>
<td>–</td>
<td>±5%</td>
<td>±5%</td>
</tr>
<tr>
<td>$Qed$</td>
<td>–</td>
<td>±4.242%</td>
<td>±4.242%</td>
</tr>
<tr>
<td>$ned$</td>
<td>–</td>
<td>±5.831%</td>
<td>±5.831%</td>
</tr>
</tbody>
</table>

Table B.8: Total uncertainties of parameters. *Estimated values due to manual measurement.
Appendix C

Calibration Reports

Inlet Pressure Sensor

CALIBRATION REPORT

CALIBRATION PROPERTIES
Calibrated by: Christoffer Meek
Type/Producer: DRUCK PTX810
SN: 2738456
Range: 1-2 bar
Unit: Bar

CALIBRATION SOURCE PROPERTIES
Type/Producer: Pressure measurements deadweight tester P3023-8-P
SN: 66611
Uncertainty [%]: 0.008

POLYFIT EQUATION:
Y = -62.94121074E+0X^0 + 31.30471854E+0X^1

CALIBRATION SUMMARY:
Max Uncertainty : 0.006767 [%]
Max Uncertainty : 0.006957 [Bar]
RSQ : 1.000000
Calibration points : 38

Figure 1: Calibration chart (The uncertainty band is multiplied by 10)
OUTLET PRESSURE SENSOR

CALIBRATION REPORT

CALIBRATION PROPERTIES
Calibrated by: Christoffer Meek
Type/Producer: DRUCK PTX810
SN: 3811122
Range: 1-2 bar
Unit: Bar

CALIBRATION SOURCE PROPERTIES
Type/Producer: Pressuremets deadweight tester P3023-6-P
SN: 66611
Uncertainty [%]: 0.008

POLY FIT EQUATION:
Y = -63.04387189E+0X^0 + 31.30132167E+0X^1

CALIBRATION SUMMARY:
Max Uncertainty : 0.006472 [%]
Max Uncertainty : 0.006647 [Bar]
RSQ : 1.000000
Calibration points : 38

Figure 1 : Calibration chart (The uncertainty band is multiplied by 10)
APPENDIX C. CALIBRATION REPORTS

Differential Pressure Sensor

CALIBRATION REPORT

CALIBRATION PROPERTIES
Calibrated by: Christoffer Meek
Type/Producer: FHCX 36W1-AKBYY
SN: 9602 N 0004 CK1
Range: 0-1 bar
Unit: Bar

CALIBRATION SOURCE PROPERTIES
Type/Producer: Pressurization deadweight tester P3023-6-P
SN: 66611
Uncertainty [%]: 0.008

POLY FIT EQUATION:
Y = -123.02468417E+0X^0 + 62.72488141E+0X^1

CALIBRATION SUMMARY:
Max Uncertainty : 0.028571 [%]
Max Uncertainty : 0.029672 [Bar]
RSQ : 1.000000
Calibration points : 36

Figure 1: Calibration chart (The uncertainty band is multiplied by 10)
Ultrasonic Flow Sensor

### Performance specifications

<table>
<thead>
<tr>
<th>Pipe inner diameter</th>
<th>Flow velocity range</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>φ13 to φ50mm</td>
<td>2 to 32 m/s</td>
<td>±1.5% of rate</td>
</tr>
<tr>
<td></td>
<td>0 to 2 m/s</td>
<td>±0.03 m/s</td>
</tr>
<tr>
<td>φ50 to φ300mm</td>
<td>2 to 32 m/s</td>
<td>±1.0% of rate</td>
</tr>
<tr>
<td></td>
<td>0 to 2 m/s</td>
<td>±0.02 m/s</td>
</tr>
<tr>
<td>φ300 to φ6000mm</td>
<td>1 to 32 m/s</td>
<td>±1.0% of rate</td>
</tr>
<tr>
<td></td>
<td>0 to 1 m/s</td>
<td>±0.01 m/s</td>
</tr>
</tbody>
</table>

Note) Reference conditions are based on JEMIS-032.
CALIBRATION REPORT

CALIBRATION PROPERTIES
Calibrated by: Christoffer Meek
Type/Producer: Ultrasonic flow meter
SN:
Range: 0-0.9
Unit: m³/s

CALIBRATION SOURCE PROPERTIES
Type/Producer: Ultrasonic flow meter
SN:
Uncertainty [%]: 1.0

POLY FIT EQUATION:
Y = 0.1127020x + 0.2272796

CALIBRATION SUMMARY:
Max Uncertainty: 0.27 [%]
Max Uncertainty: 0.01 [m³/s]
RSQ: 0.9996
Calibration points: 22

Figure 1: Calibration chart
Frequency Transformer (rotational velocity only)

CALIBRATION REPORT

CALIBRATION PROPERTIES
Calibrated by: Christoffer Meek
Type/Producer: Frequency Transformer
SN:
Range: 0-656
Unit: rpm

CALIBRATION SOURCE PROPERTIES
Type/Producer: Frequency Transformer
SN:
Uncertainty [%]: 0

POLY FIT EQUATION:
$Y = 65.9840713^*x + -0.2353826$

CALIBRATION SUMMARY:
Max Uncertainty: 0.390 [%]
Max Uncertainty: 8.69 [rpm]
R$^2$: 0.9999
Calibration points: 22

Figure 1: Calibration chart
Appendix D

Experimental Test Results

This chapter contains the measurements from the laboratory tests performed in Part 1 of the thesis.

Performance Test 24.01.18

<table>
<thead>
<tr>
<th>Q [m^3/s]</th>
<th>n [rpm]</th>
<th>P [W]</th>
<th>H [m]</th>
<th>η [-]</th>
<th>Qed [-]</th>
<th>Ned [-]</th>
<th>ρ [kg/m^3]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.119</td>
<td>150.0</td>
<td>110.9</td>
<td>0.34</td>
<td>0.4006</td>
<td>0.273</td>
<td>42.861</td>
<td>999.1</td>
</tr>
<tr>
<td>0.153</td>
<td>83.3</td>
<td>225.2</td>
<td>0.55</td>
<td>0.2939</td>
<td>0.267</td>
<td>18.196</td>
<td>999.1</td>
</tr>
<tr>
<td>0.192</td>
<td>67.5</td>
<td>499.6</td>
<td>0.68</td>
<td>0.4124</td>
<td>0.298</td>
<td>13.126</td>
<td>999.0</td>
</tr>
<tr>
<td>0.262</td>
<td>58.5</td>
<td>955.2</td>
<td>1.52</td>
<td>0.2526</td>
<td>0.271</td>
<td>7.571</td>
<td>999.1</td>
</tr>
<tr>
<td>0.282</td>
<td>117.0</td>
<td>1818.7</td>
<td>1.34</td>
<td>0.5161</td>
<td>0.312</td>
<td>16.188</td>
<td>999.0</td>
</tr>
<tr>
<td>0.314</td>
<td>82.5</td>
<td>2500.4</td>
<td>2.14</td>
<td>0.3926</td>
<td>0.274</td>
<td>9.007</td>
<td>999.1</td>
</tr>
<tr>
<td>0.348</td>
<td>142.5</td>
<td>4304.3</td>
<td>2.24</td>
<td>0.5878</td>
<td>0.297</td>
<td>15.223</td>
<td>999.0</td>
</tr>
<tr>
<td>0.395</td>
<td>112.5</td>
<td>4442.8</td>
<td>3.33</td>
<td>0.3566</td>
<td>0.276</td>
<td>9.842</td>
<td>999.1</td>
</tr>
<tr>
<td>0.452</td>
<td>120.0</td>
<td>5449.2</td>
<td>4.48</td>
<td>0.2832</td>
<td>0.272</td>
<td>9.045</td>
<td>999.1</td>
</tr>
<tr>
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Table D.1: Test results from the performance test.
Flow Meter and Frequency Transformer Calibration, 07.06.18

<table>
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<th>Q [m$^3$/s]</th>
<th>n [rpm]</th>
<th>P [W]</th>
<th>H [m]</th>
<th>η [-]</th>
<th>Qed [-]</th>
<th>Ned [-]</th>
<th>ρ [kg/m$^3$]</th>
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<td>0.73</td>
<td>0.000</td>
<td>0.164</td>
<td>7.42</td>
<td>999.10</td>
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<tr>
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<td>50.0</td>
<td>0</td>
<td>0.93</td>
<td>0.000</td>
<td>0.185</td>
<td>8.28</td>
<td>999.10</td>
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<td>0.21</td>
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<td>400</td>
<td>1.40</td>
<td>0.139</td>
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<td>800</td>
<td>1.66</td>
<td>0.196</td>
<td>0.248</td>
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<td>100.0</td>
<td>1600</td>
<td>1.97</td>
<td>0.280</td>
<td>0.269</td>
<td>11.38</td>
<td>999.03</td>
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<td>2200</td>
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<td>0.353</td>
<td>0.309</td>
<td>17.83</td>
<td>999.07</td>
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<td>199.5</td>
<td>3900</td>
<td>2.72</td>
<td>0.356</td>
<td>0.317</td>
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<td>999.06</td>
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<td>2.99</td>
<td>0.417</td>
<td>0.332</td>
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<td>0.417</td>
<td>0.340</td>
<td>22.58</td>
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<td>6500</td>
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<td>0.399</td>
<td>0.361</td>
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<td>9500</td>
<td>3.65</td>
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<td>8000</td>
<td>3.65</td>
<td>0.406</td>
<td>0.367</td>
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<td>12200</td>
<td>4.05</td>
<td>0.503</td>
<td>0.387</td>
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<td>4.45</td>
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Table D.2: Estimated results from the frequency transformer calibration.
Appendix E

ANSYS Simulation Results

This chapter include the simulation results from ANSYS CFX.

Initial Parametric Simulation

<table>
<thead>
<tr>
<th>H [m]</th>
<th>Q [m³/s]</th>
<th>n [rpm]</th>
<th>ηₜ [⁻]</th>
</tr>
</thead>
<tbody>
<tr>
<td>18</td>
<td>0.160</td>
<td>1500</td>
<td>86.4%</td>
</tr>
<tr>
<td>18</td>
<td>0.170</td>
<td>1500</td>
<td>86.3%</td>
</tr>
<tr>
<td>18</td>
<td>0.180</td>
<td>1500</td>
<td>86.3%</td>
</tr>
<tr>
<td>18</td>
<td>0.150</td>
<td>1500</td>
<td>86.4%</td>
</tr>
<tr>
<td>17</td>
<td>0.160</td>
<td>1500</td>
<td>85.8%</td>
</tr>
<tr>
<td>16</td>
<td>0.160</td>
<td>1500</td>
<td>85.2%</td>
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<tr>
<td>15</td>
<td>0.160</td>
<td>1500</td>
<td>84.5%</td>
</tr>
<tr>
<td>19</td>
<td>0.160</td>
<td>1500</td>
<td>86.8%</td>
</tr>
<tr>
<td>18</td>
<td>0.160</td>
<td>1450</td>
<td>86.9%</td>
</tr>
<tr>
<td>18</td>
<td>0.160</td>
<td>1400</td>
<td>87.4%</td>
</tr>
<tr>
<td>18</td>
<td>0.160</td>
<td>1550</td>
<td>85.7%</td>
</tr>
<tr>
<td>18</td>
<td>0.160</td>
<td>1600</td>
<td>85.1%</td>
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Table E.1: Results from initial parametric simulation
Parametric Simulation - Increased Diameter Ratio

The following output parameters was obtained from the parametric simulation:

<table>
<thead>
<tr>
<th>H [m]</th>
<th>Q [m$^3$/s]</th>
<th>n [rpm]</th>
<th>$\eta_h$ [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>0.160</td>
<td>1000</td>
<td>0.868</td>
</tr>
<tr>
<td>16</td>
<td>0.160</td>
<td>1000</td>
<td>0.866</td>
</tr>
<tr>
<td>17</td>
<td>0.160</td>
<td>1000</td>
<td>0.863</td>
</tr>
<tr>
<td>18</td>
<td>0.160</td>
<td>1000</td>
<td>0.858</td>
</tr>
<tr>
<td>19</td>
<td>0.160</td>
<td>1000</td>
<td>0.855</td>
</tr>
<tr>
<td>20</td>
<td>0.160</td>
<td>1000</td>
<td>0.851</td>
</tr>
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<td>0.160</td>
<td>1400</td>
<td>0.850</td>
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<tr>
<td>16</td>
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<td>1400</td>
<td>0.856</td>
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<tr>
<td>17</td>
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<td>1400</td>
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<td>0.160</td>
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<td>0.869</td>
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<td>1400</td>
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<td>19</td>
<td>0.160</td>
<td>1500</td>
<td>0.859</td>
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<td>1500</td>
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Table E.2: Results from the second parametric simulation
## Parametric Simulation - $N_{ed}$ Correction

<table>
<thead>
<tr>
<th>H [m]</th>
<th>Q $[m^3/s]$</th>
<th>n [rpm]</th>
<th>$\eta_h$ [-]</th>
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<tbody>
<tr>
<td>17</td>
<td>0.160</td>
<td>1000</td>
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<td>0.160</td>
<td>1000</td>
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<td>1500</td>
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<td>0.859</td>
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<td>1600</td>
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Table E.3: Parametric simulation results of the Hill chart corrected design

## Parametric Simulation - Volume Flow Correction

<table>
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<th>n [rpm]</th>
<th>$\eta_h$ [-]</th>
<th>Q $[m^3/s]$</th>
</tr>
</thead>
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<td>0.156</td>
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<td>17</td>
<td>1400</td>
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<tr>
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<tr>
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<td>0.866</td>
<td>0.163</td>
</tr>
<tr>
<td>18</td>
<td>1600</td>
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<tr>
<td>19</td>
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Table E.4: Parametric simulation results of the final design
Appendix F

HSE Report
Risikovurderingsrapport

**QRRNT-turbin**

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<th>Prosjektittel</th>
<th>Design av en turbin som utnytter overskuddsvann fra store skip</th>
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<td>Enhet</td>
<td>Vannkraftlaboratoriet</td>
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<td>Ole Gunnar Dahlhaug</td>
</tr>
<tr>
<td>Prosjektleder</td>
<td>Ole Gunnar Dahlhaug</td>
</tr>
<tr>
<td>HMS-koordinator</td>
<td>Morten Grønli</td>
</tr>
<tr>
<td>HMS-ansvarlig (linjeleder)</td>
<td>Terese Løvås</td>
</tr>
<tr>
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<td>Vannkraftlaboratoriet</td>
</tr>
<tr>
<td>Romnummer</td>
<td>-</td>
</tr>
<tr>
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<td>Christoffer Meek</td>
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**Godkjenning:**

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<td>Forsøk pågår kort (EXPERIMENT IN PROGRESS) valid for:</td>
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<td>Morten Grønli</td>
<td>86-2018</td>
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<td>HMS ansvarlig (linjeleder)</td>
<td>Terese Løvås</td>
<td>13-2018</td>
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# INNHOLDSFORTEGNELSE

1 INNLEDNING .......................................................................................................................... 1  
2 ORGANISERING .................................................................................................................... 1  
3 RISIKOSTYRING AV PROSJEKTET .................................................................................. 1  
4 TEGNINGER, FOTO, BESKRIVELSER AV FORSØKSOPPSETT ........................................... 2  
5 EVAKUERING FRA FORSØKSOPPSETNINGEN ................................................................... 2  
6 VARSリング .......................................................................................................................... 2  
6.1 Før forsøkskjøring ........................................................................................................... 2  
6.2 Ved uønskede hendelser ................................................................................................. 3  
7 VURDERING AV TEKNISK SIKKERHET .......................................................................... 3  
7.1 Fareidentifikasjon, HAZOP .............................................................................................. 3  
7.2 Brannfarlig, reaksjonsfarlig og trykksatt stoff og gass .................................................. 4  
7.3 Trykkgåjent utstyr .......................................................................................................... 4  
7.4 Påvirkning av ytre miljø (utsipp til luft/vann, støy, temperatur, rystelser, lukt) ............ 4  
7.5 Stråling .......................................................................................................................... 4  
7.6 Bruk og behandling av kjemikalier ................................................................................ 4  
7.7 El sikkerhet (behov for å avvike fra gjeldende forskrifter og normer) ......................... 5  
8 VURDERING AV OPERASJONELL SIKKERHET .................................................................. 5  
8.1 Prosedyre HAZOP .......................................................................................................... 5  
8.2 Drifts og nødstopp prosedyre ....................................................................................... 5  
8.3 Opplæring av operatører ............................................................................................... 5  
8.4 Tekniske modifikasjoner ............................................................................................... 5  
8.5 Personlig verneutstyr .................................................................................................... 6  
8.6 Generelt ........................................................................................................................ 6  
8.7 Sikkerhetsutrustning .................................................................................................... 6  
8.8 Spesielle tiltak ............................................................................................................. 6  
9 TALLFESTING AV RESTRISIKO – RISIKOMATRISE .......................................................... 6  
10 KONKLUSJON ..................................................................................................................... 6  
11 LOVER FORSKRIFTER OG PÅLEGG SOM GJELDER .......................................................... 7  
12 DOKUMENTASJON ........................................................................................................... 7  
13 VEILEDNING TIL RAPPORTMAL .................................................................................... 8
1 INNLEDNING

Forsøket går ut på å utføre effektmålinger av turbin.

2 ORGANISERING

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<tr>
<td>Apparaturansvarlig</td>
<td>Christoffer Meek</td>
</tr>
<tr>
<td>Romansvarlig</td>
<td>Torbjørn K. Nielsen</td>
</tr>
<tr>
<td>HMS koordinator</td>
<td>Morten Grønli</td>
</tr>
<tr>
<td>HMS ansvarlig (linjeleder):</td>
<td>Terese Løvås</td>
</tr>
</tbody>
</table>

3 RISIKOSTYRING AV PROSJEKTET

<table>
<thead>
<tr>
<th>Hovedaktiviteter risikostyring</th>
<th>Nødvendige tiltak, dokumentasjon</th>
<th>DTG</th>
</tr>
</thead>
<tbody>
<tr>
<td>Prosjekt initiering</td>
<td>Prosjekt initiering mal</td>
<td></td>
</tr>
<tr>
<td>Veiledningsmøte</td>
<td>Skjema for Veiledningsmøte med pre-risikovurdering</td>
<td></td>
</tr>
<tr>
<td>Innledende risikovurdering</td>
<td>Fareidentifikasjon – HAZID</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Skjema grovanalyse</td>
<td></td>
</tr>
<tr>
<td>Vurdering av teknisk sikkerhet</td>
<td>Prosess-HAZOP</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Tekniske dokumentasjoner</td>
<td></td>
</tr>
<tr>
<td>Vurdering av operasjonell sikkerhet</td>
<td>Prosedyre-HAZOP</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Opplæringsplan for operatører</td>
<td></td>
</tr>
<tr>
<td>Sluttvurdering, kvalitetssikring</td>
<td>Uavhengig kontroll</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Utstedelse av apparaturkort</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Utstedelse av forsøk pågår kort</td>
<td></td>
</tr>
</tbody>
</table>
4 BESKRIVELSER AV FORSØKSOPPSETT

- Tegninger og bilder som beskriver forsøksoppsetningen.

- Prosess og Instrumenterings Diagram, (PID) med komponentliste Se Vedlegg A
- Hvor oppholder operatør seg, hvor er gassflasker, avstegningsventiler for vann/luft. Operatør oppholder seg ved testrigg

5 EVAKUERING FRA FORSØKSOPPSETNINGEN

Evakuering skjer på signal fra alarmklokker eller lokale gassalarmstasjon med egen lokal varsling med lyd og lys utenfor aktuelle rom, se 6.2

Evakuering fra rigg området foregår igjennom merkede nødutganger til møteplass, (hjørnet gamle kjemi/kjelhuset eller parkeringsplass 1a-b.)

Aksjon på rigg ved evakuering:
Beskriv i hvilken tilstand riggen skal forlates ved en evakueringssituasjon (nødavstegning, vann, gass, spenning).

6 VARSLING

6.1 Før forsøkskjøring

Varsling per e-post, til iept-experiments@ivt.ntnu.no

I e-posten skal det stå:
- Navn på forsøksleder:
- Navn på forsøksrigg: QRRNT-turbin
- Tid for start: (dato og klokkelslett)
- Tid for stop: (dato og klokkelslett)
All forsøkskjøringen skal planlegges og legges inn i aktivitetskalender for lab. Forsøksleder må få bekreftelse på at forsøkene er klarer med øvrig labdrift før forsøk kan iverksettes.

6.2 Ved uønskede hendelser

**BRANN**
Ved brann en ikke selv er i stand til å slukke med rimelige lokalt tilgjengelige slukkemidler, skal nærmeste brannalarm utløses og arealet evakuieres raskest mulig. En skal så være tilgjengelig for brannvesen/bygningsvaktmester for å påvise brannsted. Om mulig varsles så:

<table>
<thead>
<tr>
<th>NTNU</th>
<th>SINTEF</th>
</tr>
</thead>
<tbody>
<tr>
<td>Morten Grønli, Mob: 918 97 515</td>
<td>Harald Mæhlum, Mob: 930 14 986</td>
</tr>
<tr>
<td>Terese Løvås: Mob: 735 93 709</td>
<td>Anne Karin T. Hemmingsen Mob: 930 19 669</td>
</tr>
<tr>
<td>NTNU – SINTEF Beredskapstelefon</td>
<td>800 80 388</td>
</tr>
</tbody>
</table>

**GASSALARM**

**PERSONSKADE**
- Førstehjelpsutstyr i Brann/førstehjelpletsstasjoner,
- Rop på hjelp,
- Start livreddende førstehjelp
- Ring 113 hvis det er eller det er tvil om det er alvorlig skade.

**ANDRE UØNSKEDE HENDELSER (AVVIK)**

**NTNU:**
Rapportering av uønskede hendelser, Innsida, avviksmeldinger
[https://innsida.ntnu.no/wiki/-/wiki/Norsk/Melde+avvik](https://innsida.ntnu.no/wiki/-/wiki/Norsk/Melde+avvik)

**SINTEF:**
Synergi

7 VURDERING AV TEKNISK SIKKERHET

7.1 HAZOP
Se kapittel 13 “Veiledning til rapport mal.
Forsøksoppsetningen deles inn i følgende noder:

<table>
<thead>
<tr>
<th>Node 1</th>
<th>Testrigg</th>
</tr>
</thead>
<tbody>
<tr>
<td>Node 2</td>
<td></td>
</tr>
</tbody>
</table>
7.2 Brannfarlig, reaksjonsfarlig og trykksatt stoff og gass

Se kapittel 13 "Veiledning til rapport mal.
Inneholder forsøkene brannfarlig, reaksjonsfarlig og trykksatt stoff

NEI

Vedlegg Ex-sonekart: Ta for eksempel utgangspunkt i PID
Konklusjon: Ut fra ovennevnte vurdering og funn ansees det nødvendig å utstyre anlegget med Ex sikkert utstyr, sonekart definerer hvor dette skal monteres.

7.3 Trykkpåkjent utstyr

Inneholder forsøksoppsetningen trykkpåkjent utstyr?

JA
Utstyret trykktetestes i henhold til norm og dokumenteres

Vedlegg: Sertifikat for trykkpåkjent utstyr (se vedlegg til Risikovurdering).
Konklusjon:

7.4 Påvirkning av ytre miljø (utslipp til luft/vann, støy, temperatur, rystelser, lukt)

Vil eksperimentene generere utslipp av røyk, gass, lukt eller unormalt avfall.? Mengder/konsistens. Er det behov for utslippstillatelse, ekstraordinære tiltak?

Se kapittel 13 "Veiledning til rapport mal..

NEI

Vedlegg:
Konklusjon:

7.5 Stråling

Se kapittel 13 "Veiledning til rapport mal.

NEI

Vedlegg:
Konklusjon:

7.6 Kjemikalier

Inneholder eksperimentene bruk og behandling av kjemikalier Hvilke og hvilke mengder? Hvordan skal dette avhendes, oppbevares? Risikovurder i henhold til sikkerhetsdatablad Er det behov for beskyttelses tiltak tillegges disse i operasjonell prosedyre.

Se kapittel 13 "Veiledning til rapport mal.

NEI


Vedlegg: MSDS
Konklusjon:

7.7 El sikkerhet (behov for å avvike fra gjeldende forskriver og normer)
Her forstås montasje og bruk i forhold til normer og forskriver med tanke på berøringsfare

| NEJ |

Vedlegg: Montasje utført av NTNU Drift, ikke behov for modifikasjoner

8 VURDERING AV OPERASJONELL SIKKERHET
Sikrer at etablerte prosedyrer dekker alle identifiserte risikoforhold som må håndteres gjennom operasjonelle barrierer og at peratører og teknisk utførende har tilstrekkelig kompetanse.

8.1 Prosedyre HAZOP
Se kapittel 13 “Veiledning til rapport mal.
Metoden er en undersøkelse av operasjonsprosedyrer, og identifiserer årsaker og farekilder for operasjonelle problemer.

Vedlegg: HAZOP_MAL_Prosedyre

8.2 Forsøksprosedyre og nødstopp prosedyre
Se kapittel 13 “Veiledning til rapport mal.
Driftsprosedyren er en sjekkliste som skal fylles ut for hvert forsøk.
Nødstopp prosedyren skal sette forsøksoppsetningen i en harmløs tilstand ved uforutsette hendelser.

Vedlegg: Forsøksprosedyre
Nødstopp prosedyre:

8.3 Opplæring av operatører
Dokument som viser Opplæringsplan for operatører utarbeides for alle forøksrigger.
- *Hvilke krav er det til opplæring av operatører.*
- *Hva skal til for å bli selvstendig operatør*
- *Arbeidsbeskrivelse for operatører*

Vedlegg: Opplæringsplan for operatører

8.4 Tekniske modifikasjoner
- Tekniske modifikasjoner som kan gjøres av Operatør (for eksempel, skifting av komponenter, likt mot likt)
- Tekniske modifikasjoner som må gjøres av Teknisk personale: (for eksempel modifikasjon på trykkpåkjent utstyr).
• Hvilke tekniske modifikasjoner utløser krav om ny risikovurdering (ved endring av risikobildet)?

Konklusjon:

8.5 Personlig verneutstyr
• Det er påbudt med vernebriller i sonen anlegget er plassert i.
• Det er påbudt med vernesko i sonen anlegget er plassert i.
• Det skal benyttes hansker når det er mulighet for kontakt med varme flater. 
• Det skal benyttes åndedrettsvern

Konklusjon:

8.6 Generell sikkerhet
• Området rundt forsøksoppsetningen avskjermer.
• Traverskran og truck kjøring skal ikke foregå i nærheten under eksperimentet.
• Gassflasker skal plasseres i godkjent stativ med avstengningsventil lett tilgjengelig.
• Vann og trykklufttilførsel i slanger skal stenges/kobles fra ved nærmeste fastpunkt når riggen ikke er i bruk.

8.7 Sikkerhetsutrustning
• Portable gassdetektorer skal benyttes under forsøkskjøring.
• Fare skilting, se Forskrift om Sikkerhetsskilting og signalgivning på arbeidsplassen

8.8 Spesielle tiltak
For eksempel:
• Overvåkning.
• Beredskap.
• Sikker jobb analyse ved modifikasjoner, (SJA)
• Arbeid i høyden
• Brannfarlig/giftig gass eller kjemikalier

9 TALLFESTING AV RESTRISIKO – RISIKOMATRISERE

Se kapittel 13 "Veiledning til rapport mal.
Risikomatrisen vil gi en visualisering og en samlet oversikt over aktivitetens risikoforhold slik at ledelse og brukere får et mest mulig komplett bilde av risikoforhold.

<table>
<thead>
<tr>
<th>IDnr</th>
<th>Aktivitet-hendelse</th>
<th>Frekv-Sans</th>
<th>Kons</th>
<th>RV</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Vannlekkasje og sprut</td>
<td>4</td>
<td>A</td>
<td>A4</td>
</tr>
<tr>
<td>1</td>
<td>Mye støy, personer uten verneutstyr kan komme inn i riggområde</td>
<td>4</td>
<td>A</td>
<td>A4</td>
</tr>
<tr>
<td>1</td>
<td>Strømgang fra elektriske koblinger</td>
<td>1</td>
<td>C</td>
<td>C1</td>
</tr>
<tr>
<td>1</td>
<td>Klemskade</td>
<td>4</td>
<td>A</td>
<td>A4</td>
</tr>
<tr>
<td>1</td>
<td>Varmegang i turbin</td>
<td>3</td>
<td>A</td>
<td>A3</td>
</tr>
</tbody>
</table>

Vurdering restrisiko: Deltakerne foretar en helhetsvurdering for å avgjøre om gjenværende risiko ved aktiviteten/prosessen er akseptabel. Avsperring og kjøring utenom arbeidstid
10  LOVER FORSKRIFTER OG PÅLEGG SOM GJELDER

Se [http://www.arbeidstilsynet.no/regelverk/index.html](http://www.arbeidstilsynet.no/regelverk/index.html)
- Lov om tilsyn med elektriske anlegg og elektrisk utstyr (1929)
- Arbeidsmiljøloven
- Forskrift om systematisk helse-, miljø- og sikkerhetsarbeid (HMS Internkontrollforskrift)
- Forskrift om sikkerhet ved arbeid og drift av elektriske anlegg (FSE 2006)
- Forskrift om elektriske forsyningsanlegg (FEF 2006)
- Forskrift om utstyr og sikkerhetssystem til bruk i eksplosjonsfarlig område NEK 420
- Forskrift om håndtering av brannfarlig, reaksjonsfarlig og trykksatt stoff samt utstyr og anlegg som benyttes ved håndteringen
- Forskrift om Håndtering av eksplosjonsfarlig stoff
- Forskrift om bruk av arbeidsutstyr.
- Forskrift om Arbeidsplasser og arbeidslokaler
- Forskrift om Bruk av personlig verneutstyr på arbeidsplassen
- Forskrift om Helse og sikkerhet i eksplosjonsfarlige atmosfærer
- Forskrift om Høytrykksspyling
- Forskrift om Maskiner
- Forskrift om Sikkerhetskilsting og signalgivning på arbeidsplassen
- Forskrift om Stillaser, stiger og arbeid på tak m.m.
- Forskrift om Sveising, termisk skjæring, termisk sprøyting, kullbuemeisling, lodding og sliping (varmt arbeid)
- Forskrift om Tekniske innretninger
- Forskrift om Tungt og ensformig arbeid
- Forskrift om Vern mot eksponering for kjemikalier på arbeidsplassen (Kjemikalieforskriften)
- Forskrift om Vern mot kunstig optisk stråling på arbeidsplassen
- Forskrift om Vern mot mekaniske vibrasjoner
- Forskrift om Vern mot støy på arbeidsplassen

Veiledninger fra arbeidstilsynet
se: [http://www.arbeidstilsynet.no/regelverk/veiledninger.html](http://www.arbeidstilsynet.no/regelverk/veiledninger.html)

11  DOKUMENTASJON

- Tegninger, foto, beskrivelser av forsøksoppsetningen
- Hazop_mal
- Sertifikat for trykkpåkjent utstyr
- Håndtering avfall i NTNU
- Sikker bruk av LASERE, retningslinje
- HAZOP_MAL_Proseydeyre
- Forsøksprosedyre
- Opplæringsplan for operatører
- Skjema for sikker jobb analyse, (SJA)
- Apparaturkortet
- Forsøk pågår kort
12 VEILEDNING TIL RAPPORTMAL

Kapittel 7 Vurdering av teknisk sikkerhet

Sikre at design av apparatur er optimalisert i forhold til teknisk sikkerhet.
Identifiser risikoforhold knyttet til valgt design, og eventuelt å initiere re-design for å sikre at størst mulig andel av risiko elimineres gjennom teknisk sikkerhet.
Punktene skal beskrive hva forsøksoppsettningen faktisk er i stand til å tåle og aksept for utslipp.

7.1 Fareidentifikasjon, HAZOP

Forsøksoppsettningen deles inn i noder: (eks Motorenheter, pumpeenheter, kjøleenhet.)
Ved hjelp av ledeord identifiseres årsak, konsekvens og sikkerhets tiltak. Konkluderes det med at tiltak er nødvendig anbefales disse på bakgrunn av dette. Tiltakene lukkes når de er utført og Hazop slutføres.
(eks ”No flow”, årsak: rør er deformert, konsekvens: pumpe går varm, sikkerhetsforanstaltning: måling av flow med kobling opp mot nødstopp eller hvis konsevens ikke er kritisk benyttes manuell overvåkning og punktet legges inn i den operasjonelle prosedyren.)

7.2 Brannfarlig, reaksjonsfarlig og trykksatt stoff.

I henhold til Forskrift om håndtering av brannfarlig, reaksjonsfarlig og trykksatt stoff samt utstyr og anlegg som benyttes ved håndteringen

| Brannfarlig stoff: Fast, flytende eller gassformig stoff, stoffblanding, samt stoff som forekommer i kombinasjoner av slike tilstander, som i kraft av sitt flammepunkt, kontakt med andre stoffer, trykk, temperatur eller andre kjemiske egenskaper representerer en fare for brann. |
| Reaksjonsfarlig stoff: Fast, flytende, eller gassformig stoff, stoffblanding, samt stoff som forekommer i kombinasjoner av slike tilstander, som ved kontakt med vann, ved sitt trykk, temperatur eller andre kjemiske forhold, representerer en fare for farlig reaksjon, eksplosjon eller utslipp av farlig gass, damp, støv eller tåke. |
| Trykksatt stoff: Annet fast, flytende eller gassformig stoff eller stoffblanding enn brann- eller reaksjonsfarlig stoff, som er under trykk, og som derved kan representerer en fare ved ukontrollert utslipp. |

Nærmere kriterier for klassifisering av brannfarlig, reaksjonsfarlig og trykksatt stoff er fastsatt i vedlegg 1 i veiledningen til forskriften ”Brannfarlig, reaksjonsfarlig og trykksatt stoff”


Rigg og areal skal gjennomgås med hensyn på vurdering av Ex sone

- Sone 0: Alltid eksplosiv atmosfære, for eksempel inne i tanker med gass, brennbare væske.
- Sone 1: Primær sone, tidvis eksplosiv atmosfære for eksempel et fylle tappe punkt
7.4 Påvirkning av ytre miljø

Med forurensning forstås: tilførsel av fast stoff, væske eller gass til luft, vann eller i grunnen støy og rystelser påvirkning av temperaturen som er eller kan være til skade eller ulempe for miljøet.

Regelverk: [http://www.lovdata.no/all/hl-19810313-006.html#6](http://www.lovdata.no/all/hl-19810313-006.html#6)

NTNU retningslinjer for avfall se: [http://www.ntnu.no/hms/retningslinjer/HMSR18B.pdf](http://www.ntnu.no/hms/retningslinjer/HMSR18B.pdf)

7.5 Stråling

Stråling defineres som

- **Ioniserende stråling**: Elektromagnetisk stråling (i strålevernsammenheng med bølgelengde <100 nm) eller hurtige atomære partikler (f.eks alfa- og beta-partikler) som har evne til å ionisere atomer eller molekyler.

- **Ikke-ioniserende stråling**: Elektromagnetisk stråling (bølgelengde >100 nm), og ultralyd, som har liten eller ingen evne til å ionisere.

**Strålekilder**:
- Alle ioniserende og sterke ikke-ioniserende strålekilder.
- **Ioniserende strålekilder**: Kilder som avgir ioniserende stråling, f.eks alle typer radioaktive kilder, røntgenapparater, elektronmikroskop.

1 Ultralyd er akustisk stråling ("lyd") over det hørbare frekvensområdet (>20 kHz). I strålevernforskriften er ultralyd omtalt sammen med elektromagnetisk ikke-ioniserende stråling.

2 MR (eg. NMR) - kjernemagnetisk resonans, metode som nyttes til å «avbilde» indre strukturer i ulike materialer.

3 UVC er elektromagnetisk stråling i bølgelengdeområdet 100-280 nm.

4 IR er elektromagnetisk stråling i bølgelengdeområdet 700 nm – 1 mm.

For hver laser skal det finnes en informasjonsperm(HMSRV3404B) som skal inneholde:

- Generell informasjon
- Navn på instrumentansvarlig og stedfortreder, og lokal strålevernskoordinator
- Sentrale data om apparaturen
- Instrumentespesifikk dokumentasjon
- Referanser til (evt kopier av) datablader, strålevernbestemmelser, o.l.
- Vurderinger av risikomomenter
- Instruks for brukere
- Instruks for praktisk bruk; oppstart, drift, avstenging, sikkerhetsforholdsregler, loggføring, avlåsing, evt. bruk av strålingsmåler, osv.
- Nødprosedyrer

Se ellers retningslinjen til NTNU for laser: [http://www.ntnu.no/hms/retningslinjer/HMSR34B.pdf](http://www.ntnu.no/hms/retningslinjer/HMSR34B.pdf)

7.6 Bruk og behandling av kjemikalier.

Her forstås kjemikalier som grunnstoff som kan utgjøre en fare for arbeidstakers sikkerhet og helse.


Sikkerhetsdatablær skal være i forøkenes HMS perm og kjemikaliene registrert i Stoffkartoteket.
Kapittel 8 Vurdering av operasjonell sikkerhet
Sikrer at etablerte prosedyrer dekker alle identifiserte risikoforhold som må håndteres gjennom operasjonelle barrierer og at operatører og teknisk utførende har tilstrekkelig kompetanse.

8.1 Prosedyre Hazop
Prosedyre-HAZOP gjennomføres som en systematisk gjennomgang av den aktuelle prosedyren ved hjelp av fastlagt HAZOP-metodikk og definerte ledeord. Prosedyren brytes ned i enkeltstående arbeidsoperasjoner (noder) og analyseres ved hjelp av ledeordene for å avdekke mulige avvik, uklarheter eller kilder til mangelfull gjennomføring og feil.

8.2 Drifts og nødstopp prosedyrer
Utarbeides for alle forsøksoppsetninger. Driftsprosedyren skal stegvis beskrive gjennomføringen av et forsøk, inndelt i oppstart, under drift og avslutning. Prosedyren skal beskrive forutsetninger og tilstand for start, driftsparameter med hvor store avvik som tillates før forsøket avbrytes og hvilken tilstand riggen skal forlates.
Nødstopp-prosedyre beskriver hvordan en nødstopp skal skje, (utført av uinnvidde), hva som skjer, (strøm/gass tilførsel) og hvilke hendelser som skal aktivere nødstopp, (brannalarm, lekkasje).

Kapittel 9 Risikomatriise Tallfesting av restrisiko
For å synliggjøre samlet risiko, jevnfør skjema for risikovurdering, plottes hver enkelt aktivitets verdi for sannsynlighet og konsekvens inn i risikomatriisen. Bruk aktivitetens IDnr. Eksempel: Hvis aktivitet med IDnr. 1 har fått en risikoverdi D3 (sannsynlighet 3 x konsekvens D) settes aktivitetens IDnr i risikomatriisens felt for 3D. Slik settes alle aktivitetenes risikoverdier (IDnr) inn i risikomatriisen.
I risikomatriisen er ulike grader av risiko merket med rød, gul eller grønn. Når en aktivitets risiko havner på rød (= uakseptabel risiko), skal risikoreducerende tiltak gjennomføres. Ny vurdering gjennomføres etter at tiltak er iverksatt for å se om risikoverdien er kommet ned på akseptabelt nivå.

<table>
<thead>
<tr>
<th>KONSEKVENSES</th>
<th>Svært alvorlig</th>
<th>Alvorlig</th>
<th>Moderat</th>
<th>Liten</th>
<th>Svært liten</th>
</tr>
</thead>
<tbody>
<tr>
<td>SANSYNLIGHET</td>
<td>E1</td>
<td>E2</td>
<td>E3</td>
<td>E4</td>
<td>E5</td>
</tr>
<tr>
<td></td>
<td>D1</td>
<td>D2</td>
<td>D3</td>
<td>D4</td>
<td>D5</td>
</tr>
<tr>
<td></td>
<td>C1</td>
<td>C2</td>
<td>C3</td>
<td>C4</td>
<td>C5</td>
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<td></td>
<td>B1</td>
<td>B2</td>
<td>B3</td>
<td>B4</td>
<td>B5</td>
</tr>
<tr>
<td></td>
<td>A1</td>
<td>A2</td>
<td>A3</td>
<td>A4</td>
<td>A5</td>
</tr>
</tbody>
</table>

| Svært liten | Liten | Middels | Stor | Svært Stor |

| SANSYNLIGHET |
Prinsipp over akseptkriterium. Forklaring av fargene som er brukt i risikomatrisen.

<table>
<thead>
<tr>
<th>Farge</th>
<th>Beskrivelse</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rød</td>
<td>Uakseptabel risiko. Tiltak skal gjennomføres for å redusere risikoen.</td>
</tr>
<tr>
<td>Gul</td>
<td>Vurderingsområde. Tiltak skal vurderes.</td>
</tr>
<tr>
<td>Grønn</td>
<td>Akseptabel risiko. Tiltak kan vurderes ut fra andre hensyn.</td>
</tr>
</tbody>
</table>
### Kartlegging av risikofylt aktivitet

**Enhet:** Vannkraftlaboratoriet  
**Dato:** 22.01.18

**Linjefører:** Torbjørn Kristian Nilsen

**Deltakere ved kartleggingen (m/f, funksjon):** Ole Gunnar Dahlhaug, Christoffer Meek, Bjørn Solemslie, Chirag Trivedi  
(Ans. veileder, student, evtl. medveileder, evtl. andre m. kompetanse)

**Kort beskrivelse av hovedaktivitet/hovedprosess:** Masteroppgave Christoffer Meek. Design of a turbine which utilize the spill water of ships

**Er oppgaven rent teoretisk? (JA/NEI):** Nei


**Signaturer:**  
- Ansvarlig veileder:  
- Student:

<table>
<thead>
<tr>
<th>ID nr.</th>
<th>Aktivitet/prosess</th>
<th>Ansvarlig</th>
<th>Eksisterende dokumentasjon</th>
<th>Eksisterende sikringstiltak</th>
<th>Lov, forskrift o.l.</th>
<th>Kommentar</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Testing av vannturbin</td>
<td>Christoffer Meek, Ole Gunnar Dahlhaug</td>
<td>Tegninger (se vedlegg)</td>
<td>Vernesko, hørselsvern, vernebriller</td>
<td></td>
<td></td>
</tr>
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</table>

Vennesko, hørselsvern, vernebriller
## Risikovurdering

### Enhet:
Vannkraftlaboratoriet

### Linjeføder:
Torbjørn Kristian Nilsen

### Deltakere ved kartleggingen:
Ole Gunnar Dahlhaug, Christoffer Meek, Bjørn Solemslie, Chirag Trivedi

### Kort beskrivelse av hovedaktivitet/hovedprosess:
Masteroppgave Christoffer Meek. Design of a turbine which utilize the spill water of ships

### Signaturer:
Ansvarlig veileder: Student:

### Tabell med risikovurderinger:

<table>
<thead>
<tr>
<th>ID nr</th>
<th>Aktivitet fra kartleggings-skjemaet</th>
<th>Mulig uønsket hendelse/belastning</th>
<th>Vurdering av sannsynlighet (1-5)</th>
<th>Vurdering av konsekvens: Menneske (A-E), Ytre miljø (A-E), Øvemateriell (A-E), Om-dømme (A-E)</th>
<th>Risikoverdi (menneske)</th>
<th>Kommentarer/status Forslag til tiltak</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Testing av vannturbin</td>
<td>Vannleksasje og sprut</td>
<td>4</td>
<td>A A B A A4</td>
<td></td>
<td>Dobbeltsjekke at flenser er tilstrøkkelig strammet, bruke vernebriller</td>
</tr>
<tr>
<td>1</td>
<td>Testing av vannturbin</td>
<td>Strømgang fra elektriske koblinger</td>
<td>1</td>
<td>C A C C C1</td>
<td></td>
<td>Avskjerming av elektriske koblinger under kjøring</td>
</tr>
<tr>
<td>1</td>
<td>Testing av vannturbin</td>
<td>Varmegang i turbin</td>
<td>3</td>
<td>A A B A A3</td>
<td></td>
<td>Bruk hansker</td>
</tr>
<tr>
<td>1</td>
<td>Testing av vannturbin</td>
<td>Klemknede</td>
<td>4</td>
<td>A A A A A4</td>
<td></td>
<td>Bruk vernesko og være oppmerksom</td>
</tr>
</tbody>
</table>
Sannsynlighet vurderes etter følgende kriterier:

<table>
<thead>
<tr>
<th>Svært liten 1</th>
<th>Liten 2</th>
<th>Middels 3</th>
<th>Stor 4</th>
<th>Svært stor 5</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 gang pr 50 år eller sjeldnere</td>
<td>1 gang pr 10 år eller sjeldnere</td>
<td>1 gang pr år eller sjeldnere</td>
<td>1 gang pr måned eller sjeldnere</td>
<td>Skjer ukentlig</td>
</tr>
</tbody>
</table>

Konsekvens vurderes etter følgende kriterier:

<table>
<thead>
<tr>
<th>Gradering</th>
<th>Menneske</th>
<th>Ytre miljø Vann, jord og luft</th>
<th>Øk/materiell</th>
<th>Omdømme</th>
</tr>
</thead>
<tbody>
<tr>
<td>E</td>
<td>Død</td>
<td>Svært langvarig og ikke reversibel skade</td>
<td>Drifts- eller aktivitetsstans &gt;1 år</td>
<td>Trovertighet og respekt betydelig og varig svekket</td>
</tr>
<tr>
<td>D</td>
<td>Alvorlig personskade. Mulig utøvelse</td>
<td>Langvarig skade. Lang restitusjonstid</td>
<td>Driftsstans &gt; ½ år Aktivitetsstans i opp til 1 år</td>
<td>Trovertighet og respekt betydelig svekket</td>
</tr>
<tr>
<td>C</td>
<td>Moderat</td>
<td>Alvorlig personskade.</td>
<td>Mindre skade og lang restitusjonstid</td>
<td>Drifts- eller aktivitetsstans &lt; 1 mnd</td>
</tr>
<tr>
<td>B</td>
<td>Liten</td>
<td>Skade som krever medisinsk behandling</td>
<td>Mindre skade og kort restitusjonstid</td>
<td>Drifts- eller aktivitetsstans &lt; 1 uke</td>
</tr>
<tr>
<td>A</td>
<td>Svært liten</td>
<td>Skade som krever førstehjelp</td>
<td>Ubetydelig skade og kort restitusjonstid</td>
<td>Drifts- eller aktivitetsstans &lt; 1 dag</td>
</tr>
</tbody>
</table>

Risikoverdi = Sannsynlighet x Konsekvens

Beregn risikoverdi for Menneske. Enheten vurderer selv om de i tillegg vil beregne risikoverdi for Ytre miljø, Økonomi/materiell og Omdømme. I så fall beregnes disse hver for seg.

Til kolonnen "Kommentarer/status, forslag til forebyggende og korrigerende tiltak":

Tiltak kan påvirke både sannsynlighet og konsekvens. Prioriter tiltak som kan forhindre at hendelsen inntreffer, dvs. sannsynlighetsreduserende tiltak foran skjerpet beredskap, dvs. konsekvensreduserende tiltak.
MATRISE FOR RISIKOVURDERINGER ved NTNU

## Konsekvens

<table>
<thead>
<tr>
<th>Svært alvorlig</th>
<th>E1</th>
<th>E2</th>
<th>E3</th>
<th>E4</th>
<th>E5</th>
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<tbody>
<tr>
<td>Alvorlig</td>
<td>D1</td>
<td>D2</td>
<td>D3</td>
<td>D4</td>
<td>D5</td>
</tr>
<tr>
<td>Moderat</td>
<td>C1</td>
<td>C2</td>
<td>C3</td>
<td>C4</td>
<td>C5</td>
</tr>
<tr>
<td>Liten</td>
<td>B1</td>
<td>B2</td>
<td>B3</td>
<td>B4</td>
<td>B5</td>
</tr>
<tr>
<td>Svarter liten</td>
<td>A1</td>
<td>A2</td>
<td>A3</td>
<td>A4</td>
<td>A5</td>
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</table>

## Sanneynlighet

<table>
<thead>
<tr>
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<th>Liten</th>
<th>Middels</th>
<th>Stor</th>
<th>Svært stor</th>
</tr>
</thead>
</table>

**Prinsipp over akseptkriterium.** Forklaring av fargene som er brukt i risikomatrisen.

<table>
<thead>
<tr>
<th>Farge</th>
<th>Beskrivelse</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rød</td>
<td>Uakseptabel risiko. Tiltak skal gjennomføres for å redusere risikoen.</td>
</tr>
<tr>
<td>Gul</td>
<td>Vurderingsområde. Tiltak skal vurderes.</td>
</tr>
<tr>
<td>Grønn</td>
<td>Akseptabel risiko. Tiltak kan vurderes ut fra andre hensyn.</td>
</tr>
</tbody>
</table>
**Vedlegg til**  
**Risikovurderingsrapport**

**QRRNT-turbin**

<table>
<thead>
<tr>
<th>Prosjekttittel</th>
<th>Design av en turbin som utnytter overskuddsvann fra store skip</th>
</tr>
</thead>
<tbody>
<tr>
<td>Apparatur</td>
<td>QRRNT-turbin</td>
</tr>
<tr>
<td>Enhet</td>
<td>Vannkraftlaboratoriet</td>
</tr>
<tr>
<td>Apparaturansvarlig</td>
<td>Christoffer Meek</td>
</tr>
<tr>
<td>Prosjektleder</td>
<td>Ola Gunnar Dahlhaug</td>
</tr>
<tr>
<td>HMS-koordinator</td>
<td>Bård Aslak Brandåstrø</td>
</tr>
<tr>
<td>HMS-ansvarlig (linjeleder)</td>
<td>Torbjørn Kristian Nielsen</td>
</tr>
<tr>
<td>Plassering</td>
<td>Vannkraftlaboratoriet</td>
</tr>
<tr>
<td>Romnummer</td>
<td>-</td>
</tr>
<tr>
<td>Risikovurdering utført av</td>
<td>Christoffer Meek</td>
</tr>
</tbody>
</table>

**INNHOLDSFORTEGNELSE**

VEDLEGG A: PROSESS OG INSTRUMENTERINGSDIAGRAM......................................................... 1  
VEDLEGG B: HAZOP (MAL) ........................................................................................................... 2  
VEDLEGG C: PRØVESERTIFIKAT FOR LOKAL TRYKKTESTING......................................................... 4  
VEDLEGG D: HAZOP PROSEDYRE (MAL) .......................................................................................... 5  
VEDLEGG E: FORSØKSPROSEDYRE ............................................................................................... 6  
VEDLEGG F: OPPLÆRINGSPLAN FOR OPPERATØRER ........................................................................ 8  
VEDLEGG G: SKJEMA FOR SIKKER JOBBA楠LYSE ......................................................................... 9  
APPARATURKORT / UNITCARD .................................................................................................. 11  
FORSØK PÅGÅR /EXPERIMENT IN PROGRESS ............................................................................. 12
<table>
<thead>
<tr>
<th>Ref#</th>
<th>Guideword</th>
<th>Causes</th>
<th>Consequences</th>
<th>Safeguards</th>
<th>Recommendations</th>
<th>Action</th>
<th>Date/Sign</th>
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<td>Safety</td>
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</tbody>
</table>
VEDLEGG C: PRØVESERTIFIKAT FOR LOKAL TRYKKTESTING

Trykk testen skal utføres I følge NS-EN 13445 del 5 (Inspeksjon og prøving). Se også prosedyre for trykktesting gjeldende for VATL lab

<table>
<thead>
<tr>
<th>Trykkpåkjent utstyr:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Benyttes i rigg:</td>
</tr>
<tr>
<td>Design trykk for utstyr (bara):</td>
</tr>
<tr>
<td>Maksimum tillatt trykk (bara): (i.e. burst pressure om kjent)</td>
</tr>
<tr>
<td>Maksimum driftstrykk i denne rigg:</td>
</tr>
</tbody>
</table>

Prøvetrykket skal fastlegges i følge standarden og med hensyn til maksimum tillatt trykk.

<table>
<thead>
<tr>
<th>Prøvetrykk (bara):</th>
</tr>
</thead>
<tbody>
<tr>
<td>X maksimum driftstrykk:</td>
</tr>
<tr>
<td>I følge standard</td>
</tr>
<tr>
<td>Test medium:</td>
</tr>
<tr>
<td>Temperatur (°C)</td>
</tr>
<tr>
<td>Start tid:</td>
</tr>
<tr>
<td>Trykk (bara):</td>
</tr>
<tr>
<td>Slutt tid:</td>
</tr>
<tr>
<td>Trykk (bara):</td>
</tr>
<tr>
<td>Maksimum driftstrykk i denne rigg:</td>
</tr>
</tbody>
</table>

Eventuelle repetisjoner fra atm. trykk til maksimum prøvetrykk:.................

Test trykket, dato for testing og maksimum tillatt driftstrykk skal markers på (skilt eller innslått)

______________________________  __________________
Sted og dato  Signatur
### VEDLEGG D: HAZOP PROSEDYRE (MAL)

**Project:**
- **Node:** 1

<table>
<thead>
<tr>
<th>Ref#</th>
<th>Guideword</th>
<th>Causes</th>
<th>Consequences</th>
<th>Safeguards</th>
<th>Recommendations</th>
<th>Action</th>
<th>Date/Sign</th>
</tr>
</thead>
<tbody>
<tr>
<td>Uklar</td>
<td>Prosedyre er laget for ambisjøs eller preget av forvirring</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Trinn på feil plass</td>
<td>Prosedyren vil lede til at handleder blir gjennomført i feil mønster/rekkefølge</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Feil handling</td>
<td>Prosedyrens handling er feil spesifisert</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Uriktig informasjon</td>
<td>Informasjon som er gitt i forkant av handling er feil spesifisert</td>
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<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Trinn utelatt</td>
<td>Manglende trinn, eller trinn krever for mye av operatør</td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Trinn mislykket</td>
<td>Trinn har stor sannsynlighet for å mislykkes</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Påvirkning og effekter fra andre</td>
<td>Prosedyrens prestasjoner vil trolig bli påvirket av andre kilder</td>
<td></td>
<td></td>
<td></td>
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<td></td>
</tr>
</tbody>
</table>
## VEDLEGG E: FORSØKSPROSEDYRE

<table>
<thead>
<tr>
<th>Apparatur</th>
<th>Dato</th>
<th>Signatur</th>
</tr>
</thead>
<tbody>
<tr>
<td>QRRNT-turbin</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

### Conditions for the experiment:

- Experiments should be run in normal working hours, 08:00-16:00 during winter time and 08.00-15.00 during summer time. Experiments outside normal working hours shall be approved.  
- One person must always be present while running experiments, and should be approved as an experimental leader.  
- An early warning is given according to the lab rules, and accepted by authorized personnel.  
- Be sure that everyone taking part of the experiment is wearing the necessary protecting equipment and is aware of the shut down procedure and escape routes.

### Preparations

- Post the “Experiment in progress” sign.

### Start up procedure

1. Skru på frekvensomformeren i kjelleren.

2. Oppstart i kontrollrommet: Still inn ventilene til riktig posisjon via remote-desktop-PC koblet til en PCene i kontrollrommet.
   - a) Gå en runde i laboratoriet og kontroller at ventilene er riktig innstilt
   - b) Se til at pumpas setpunkt er 100 RPM.
   - c) Start Pumpa.

3. Sett opp turtallet på pumpen til ønsket vannhøyde i trykktank oppnåes (ca 500rpm)

4. Skru på generatoren til turbinen og la den kjøre på 100 RPM

5. Øk turtall til pumpe og turbin stegvis til ønsket driftspunkt oppnås.

### Control of temperature, pressure, flow and power

### End of experiment

### Shut down procedure

1. Skru ned turtall på pumpe og generator stegvis til du oppnår turtall 100 på b) pumpa og generator.
2. Stopp generator.
4. e) Gå ned i kjelleren og skru av frekvensomformeren.

---

Remove all obstructions/barriers/signs around the experiment. X
Tidy up and return all tools and equipment. X
Tidy and cleanup work areas. X
Return equipment and systems back to their normal operation settings (fire alarm) X

To reflect on before the next experiment and experience useful for others

Was the experiment completed as planned and on scheduled in professional terms? X
Was the competence which was needed for security and completion of the experiment available to you? X
Do you have any information/knowledge from the experiment that you should document and share with fellow colleagues? X

---

Operatører:

<table>
<thead>
<tr>
<th>Navn</th>
<th>Dato</th>
<th>Signatur</th>
</tr>
</thead>
<tbody>
<tr>
<td>Christoffer Meek</td>
<td>30.01.18</td>
<td><img src="signature.png" alt="Signature" /></td>
</tr>
<tr>
<td>Fredrik Linge</td>
<td>30.01.18</td>
<td><img src="signature.png" alt="Signature" /></td>
</tr>
</tbody>
</table>
VEDLEGG F: OPPLÆRINGSPLAN FOR OPPERATØRER

Prosjekt
Design av en turbin som utnytter overskuddsvann fra store skip

Apparatur
QRNRT-turbin

Prosjektleder
Ole Gunnar Dahlhaug

Dato
30.01.2018

Kjennskap til EPT LAB generelt
Lab
- adgang
- rutiner/regler
- arbeidstid

Kjenner til evakueringsprosedyrer

Aktivitetskalender
Innmelding av forsøk til: iep-experiments@ivt.ntnu.no

Kjennskap til forsøkene
Prosedyrer for forsøkene
Nødstopp
Nærmeste brann/førstehjelpsstasjon

Jeg erklærer herved at jeg har gjennomgått og forstått HMS-regelverket, har fått hensiktsmessig opplæring for å kjøre dette eksperimentet og er klar over mitt personlige ansvar ved å arbeide i EPT laboratorier.

Operatører

<table>
<thead>
<tr>
<th>Navn</th>
<th>Dato</th>
<th>Signatur</th>
</tr>
</thead>
<tbody>
<tr>
<td>Christoffer Meek</td>
<td>30.01.18</td>
<td>[Signatur]</td>
</tr>
<tr>
<td>Fredrik Linge</td>
<td>30.01.18</td>
<td>[Signatur]</td>
</tr>
</tbody>
</table>
VEDLEGG G: SKJEMA FOR SIKKER JOBB ANALYSE

**SJA tittel:**
Dato: 30.01.2018
Kryss av for utfylt sjekkliste: x
Sted: Vannkraftlaboratoriet

**Deltakere:**

<table>
<thead>
<tr>
<th>Christoffer Meek</th>
<th>Fredrik Linge</th>
</tr>
</thead>
</table>

**SJA-ansvarlig:**

| Christoffer Meek |

**Arbeidsbeskrivelse:**
(Hva og hvordan?)
Effektmålinger av aksial Kaplan turbin. Måling av trykk, volumstrøm og effekt fra sensorer.

**Risiko forbundet med arbeidet:**
Fare for vannsprut, klemfare, elektrisk støt og støy

**Beskyttelse/sikring:**
(tiltaksplan, se neste side)
Vernebriller, vernesko, hørselsvern og avskjermering

**Konklusjon/kommentar:**

**Anbefaling/godkjenning:**

<table>
<thead>
<tr>
<th>Anbefaling/godkjenning:</th>
<th>Dato/Signatur:</th>
<th>Anbefaling/godkjenning:</th>
<th>Dato/Signatur:</th>
</tr>
</thead>
<tbody>
<tr>
<td>SJA-ansvarlig: Christoffer</td>
<td>30.01.18</td>
<td>HMS coordinator Morten Grønli</td>
<td></td>
</tr>
<tr>
<td>Ansvarlig for utføring:</td>
<td>30.01.18</td>
<td>Annen (stilling):</td>
<td></td>
</tr>
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</table>

Ansvarlig for utføring: 30.01.18
HMS coordinator Morten Grønli

9
<table>
<thead>
<tr>
<th>HMS aspekt</th>
<th>Ja</th>
<th>Nei</th>
<th>NA</th>
<th>Kommentar / tiltak</th>
<th>Ansv.</th>
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</thead>
<tbody>
<tr>
<td>Dokumentasjon, erfaring, kompetanse</td>
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<td>Kjent arbeidsoperasjon?</td>
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<tr>
<td>Nødvendig personell?</td>
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<td></td>
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<tr>
<td>Kommunikasjon og koordinering</td>
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<tr>
<td>Mulig konflikt med andre operasjoner?</td>
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<tr>
<td>Håndtering av en evnt. hendelse (alarm, evakuering)?</td>
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<tr>
<td>Behov for ekstra vakt?</td>
<td>X</td>
<td></td>
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<tr>
<td>Arbeidsstedet</td>
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<tr>
<td>Uvante arbeidsstillinger?</td>
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<tr>
<td>Arbeid i tanker, kummer el.lignende?</td>
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<tr>
<td>Arbeid i grøfter eller sjakter?</td>
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<tr>
<td>Rent og ryddig?</td>
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<td>Verneutstyr ut over det personlige?</td>
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<td>Vær, vind, sikt, belysning, ventilasjon?</td>
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<tr>
<td>Bruk av stillas/lift/seler/stropper?</td>
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<tr>
<td>Arbeid i høyden?</td>
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<tr>
<td>Ioniserende stråling?</td>
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<tr>
<td>Rømningsveier OK?</td>
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<tr>
<td>Kjemiske farer</td>
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<tr>
<td>Bruk av helseskadelige/giftige/etsende kjemikalier?</td>
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<tr>
<td>Bruk av brannfarlige eller eksplosjonsfarlige kjemikalier?</td>
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<tr>
<td>Er broken risikovurdt?</td>
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<tr>
<td>Biologisk materiale?</td>
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<td>Støv/asbest/isolasjonsmateriale?</td>
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<tr>
<td>Mekaniske farer</td>
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<tr>
<td>Stabilitet/styrke/spenning?</td>
<td>X</td>
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<tr>
<td>Klem/kutt/slag?</td>
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<tr>
<td>Støy/trykk/temperatur?</td>
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<tr>
<td>Behandling av avfall?</td>
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<tr>
<td>Behov for spesialverktøy?</td>
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<tr>
<td>Elektriske farer</td>
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<tr>
<td>Strøm/spenning/over 1000V?</td>
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<tr>
<td>Støt/krypstrøm?</td>
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<td>Tap av strømtilførsel?</td>
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<td>Området</td>
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<td>Behov for befaring?</td>
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<td>Merking/skilting/avsperring?</td>
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<td>Miljømessige konsekvenser?</td>
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<tr>
<td>Sentrale fysiske sikkerhetssystemer</td>
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<tr>
<td>Arbeid på sikkerhetssystemer?</td>
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<tr>
<td>Frakobling av sikkerhetssystemer?</td>
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<tr>
<td>Annet</td>
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</tbody>
</table>
**Dette kortet SKAL henges godt synlig ved maskinen!**
_This card MUST be posted on a visible place on the unit!_

<table>
<thead>
<tr>
<th>Apparatur (Unit)</th>
<th>Dato Godkjent (Date Approved)</th>
</tr>
</thead>
<tbody>
<tr>
<td>QRRNT Turbin</td>
<td>torsdag 8. februar 2018</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Prosjektleder (Project Leader)</th>
<th>Telefon mobil/privat (Phone no. mobile/private)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ole Gunnar Dahlhaug</td>
<td>91897609</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Apparatursvarlig (Unit Responsible)</th>
<th>Telefon mobil/privat (Phone no. mobile/private)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ole Gunnar Dahlhaug</td>
<td>91897609</td>
</tr>
</tbody>
</table>

**Sikkerhetsrisikoer (Safety hazards)**
Støy, vannsprut, elektrisk støt, varmegang, klem

**Sikkerhetsregler (Safety rules)**
Bruk av vernesko, vernebriller, avskjerme elektriske koblinger

**Nødstopp prosedyre (Emergency shutdown)**
Slå av vanntilførsel og stenge av ventiler til turbin

<table>
<thead>
<tr>
<th>Her finner du (Here you will find):</th>
</tr>
</thead>
<tbody>
<tr>
<td>Prosedyrer (Procedures)</td>
</tr>
<tr>
<td>Bruksanvisning (User manual)</td>
</tr>
<tr>
<td>Brannslukningsapparat (Fire extinguisher)</td>
</tr>
<tr>
<td>Førsthjelpsskap (First aid cabinet)</td>
</tr>
</tbody>
</table>

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Ved rigg</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
</tr>
<tr>
<td>1. etasje v/trapp</td>
<td></td>
</tr>
<tr>
<td>1. etasje v/trapp</td>
<td></td>
</tr>
</tbody>
</table>

**NTNU**
Institutt for energi og prosessteknikk

<table>
<thead>
<tr>
<th>Dato</th>
<th>Signert</th>
</tr>
</thead>
<tbody>
<tr>
<td>13/2-2018</td>
<td>Jane Doe</td>
</tr>
</tbody>
</table>

**NTNU**  **SINTEF**
Dette kortet SKAL henges opp før forsøk kan starte! This card MUST be posted on the unit before the experiment startup!

<table>
<thead>
<tr>
<th>Apparatur (Unit)</th>
<th>Dato godkjent (Date Approved)</th>
</tr>
</thead>
<tbody>
<tr>
<td>QRRNT Turbin</td>
<td>torsdag 8. februar 2018</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Prosjektleder (Project Leader)</th>
<th>Telefon mobil/privat (Phone no. mobile/private)</th>
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<tbody>
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<td>Ole Gunnar Dahlhaug</td>
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</tbody>
</table>

<table>
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<tr>
<th>Apparatursvarlig (Unit Responsible)</th>
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</tr>
</thead>
<tbody>
<tr>
<td>Ole Gunnar Dahlhaug</td>
<td>91897609</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Godkjente operatører (Approved Operators)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Navn/Name</td>
</tr>
<tr>
<td>-----------</td>
</tr>
<tr>
<td>Meek, Christoffer</td>
</tr>
<tr>
<td>Linge, Fredrik</td>
</tr>
</tbody>
</table>

Prosjekt (Project)
Design av en turbin som utnytter overskuddsvann fra store skip

Forsøkstid / Experimental time (start - stop)
08.02.2018 - 08.02.2019

Kort beskrivelse av forsøket og relaterte farer (Short description of the experiment and related hazards)
Påtrykkning av vann og måling av produsert effekt. Fare for støy, vannsprut, elektrisk støt, varmegang, klem

NTNU
Institutt for energi og prosessteknikk

Dato: 13/2/2018

Signert: [Signature]

NTNU SINTEF
Appendix G

MATLAB Scripts

This section contains the MATLAB scripts and functions used to analyze data, sensor calibration and to find design parameters. Here is a summary of the codes and their usage:

<table>
<thead>
<tr>
<th>Script/Function Name</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>ContAnalysis</td>
<td>Creates time-serie plots of selected variables</td>
</tr>
<tr>
<td>PerformanceAnalysis</td>
<td>Analyses multiple data points and produce turbine characteristics</td>
</tr>
<tr>
<td>Calibration</td>
<td>Used for manual calibration and uncertainty calculation</td>
</tr>
<tr>
<td>hill</td>
<td>Creates a Hill chart from measurement data</td>
</tr>
<tr>
<td>HillFromXl</td>
<td>Creates a Hill chart from parametric simulation results</td>
</tr>
<tr>
<td>Initializer</td>
<td>Finds turbine design parameters</td>
</tr>
<tr>
<td>XSteam</td>
<td>Provides the thermodynamic properties of water</td>
</tr>
</tbody>
</table>

Table G.1: Summary of MATLAB scripts and functions

ContAnalysis.m

```matlab
clc;
clear all;
close all;

N=1;
r=100000; %Reduction factor (#points to average)

var= {'Pin', 'Pout', 'Pdiff', 'Q', 'n', 'P', 'T', 't', 'H'};

%index 1 2 3 4 5 6 7 8 9

% Calibration errors

calErr=[0.0001029,0.0001048,0.0002967,0.00542,0.0004,0.0006,0.0005,0];

% Constants

D=0.5;
g=9.82146516;
da=pi^2/16*(0.5^4+0.6^4)/(0.6^4-0.5^4); % Squared turbine area difference

% Read in data

file='q67h41-0706.csv';

[Pin,Pout,Pdiff,Q,n,T,t,Date,Time,Loggers,Plab]=readCSVData(file);

% From another source

T(1:length(P))=17.13;
P(1:length(P))=15000;
n(1:length(n))=50.6+7.5;
Q(1:length(Q))=0.67;

data=[Pin;Pout;Pdiff;Q;n;P;T;t];

% Analysis

rho=XSteam('rho_pT',Plab,mean(T)); % Density

% Head

H=Pdiff*10^5/(rho*g)+(Q.^2/(2*g*da));
H(H<0)=0;
data([9,:])=H;

% Reduce datasets and find error

for i=1:8
    [dataR(i,:),eMax(i),f(i,:)]=reducePointsX(data(i,:),r);
eMax(i)=sqrt(eMax(i)^2+calErr(i)^2);
    fprintf('Max error %5s = %0.5f
',var{i},eMax(i));
end

% Head error

k=rho*dataR(4).^2./(da*dataR(3));
f(9,:)=((1./(0.5+k+1)).^2.*f(3,:).^2+(1./((1./k+0.5)).^2.*f(4,:).^2).^0.5);
eMax(9)=max(f(9,:));
    fprintf('Max error %5s = %0.5f \n', sprintf('%H'),eMax(9));

% Qed error

Qed=data(4,:)./(D^2+(1+data(9,:)).^0.5);
[dataR(10,:),a,b]=reducePointsX(Qed,r);
f(10,:)=(f(4,:).^2+(0.5*f(9,:)).^2).^0.5;
eMax(10)=max(f(10,:)); % Max uncertainty
APPENDIX G. MATLAB SCRIPTS

APPENDIX G. MATLAB SCRIPTS

60 fprintf('Max error Qed = %0.5f \n', eMax(10));
61
62 Ned = data(5,:)/((g.*data(9,:)).^0.5);
63 [dataR(11,:), a, b] = reducePointsX(Ned, r);
64 f(11,:) = (f(5,:).^2 + (0.5*f(9,:)).^2).^(0.5);
65 eMax(11) = max(f(11,:)); % Max uncertainty
66 fprintf('Max tot error %5s = %0.5f \n', 'Ned', eMax(11));
67
68 [dataR(9,:), a, b] = reducePointsX(H, r);
69
70 % Ned error
71 k1 = rho/(da*2);
72 Eta = data(6,:)/((data(4,:).*(data(3,:)+100000+(data(4,:).^2.*rho.*da/2)))).^(0.5);
73 f(12,:) = (f(6,:).^2 + (3.*k1.*dataR(4,:).^2 + dataR(3,:))./(dataR(3,:)+k1.*dataR(4,:).^2.*2)).*f(4,:)
74 .^2 + (1./((1 + dataR(4,:).^2.*k1./dataR(3,:))).*f(3,:).^2.*2).^(0.5);
75 eMax(12) = f(12);
76 [dataR(12,:), a, b] = reducePointsX(Eta, r);
77 fprintf('Max tot error %5s = %0.5f \n', 'Eta', eMax(12));
78
79 clear k a b
80
81 % Time charts
82
83 varplot = [5, 6, 8, 12]; % variables to plot: Q, n, T, H
84 ylab = {'Rotational velocity, n [rpm]', 'Power, P [W]', 'Head, H [m]', 'Total Efficiency, \eta [-]'}; % label
85 tit = {'Rotational Velocity', 'Electrical Power Output', 'Head', 'Total Efficiency'};
86
87 for j = 1:length(varplot)
88 subplot(2,2,j)
89 errorbar(dataR(8,:), dataR(varplot(j,:), :), f(varplot(j,:), :));
90 hold on;
91 plot(dataR(8,:), dataR(varplot(j,:), :), 'r');
92 title(sprintf('%s', tit{j}));
93 xlabel('Time, t [s]');
94 ylabel(ylab{j});
95 % set(gca, 'FontSize', 16);
96 disp(tit{j});
97 end
98
99 s up t i t l e ( s p r i n t f ( ' Parameters at Q=%0.3f m^3/s', mean(Q)));
close all;

% Constants

Var = {'Pin', 'Pout', 'Pdiff', 'Q', 'n', 'P', 't', 'H', 'Qed', 'Ned', 'Eta', 'Plab', 'Date', 'Time', 'Loggers'};

% %index 1 2 3 4 5 6 7 8 9 10 11 12

dat = {'07.06.2018', '24.01.2018', '29.11.2017'};

% Calibration errors
if strcmp(dat, '24.01.2018') || strcmp(dat, '07.01.2018')
calErr = [0.0001029, 0.0001048, 0.0002967, 0.01, 0.005, 0.005, 0.005, 0];
else if strcmp(dat, '29.11.2017')
calErr = [0.00501, 0.00502, 0.00507, 0.00542, 0.0004, 0.0006, 0.0005, 0];
end

% Gravitational acceleration in laboratory
% Turbine inlet and outlet diameter
D = 0.5; % Squared turbine area difference

READ = 1; % Set to 1 if short Excel file exist, 0 if not

% Filenames

if strcmp(dat, '07.06.2018')
qs = [ ];
heads = [ ];
else if strcmp(dat, '24.01.2018')
qs = [120, 153, 192, 262, 285, 315, 364, 395, 454, 486, 520, 526, 580, 628];
heads = [5, 7, 86, 17, 15, 23, 27, 34, 46, 42, 35, 33, 44, 54];
else if strcmp(dat, '29.11.2017')
qs = [140, 209, 264, 304, 357, 393, 417, 419, 430, 445, 467, 478, 486, 493, 521, 521, 542, 562, 600];
heads = [6, 12, 17, 21, 28, 31, 32, 37, 36, 40, 41, 46, 46, 49, 50, 56, 58, 60, 66, 70];
end

N = length(qs);

if READ == 0 % If no short Excel file exist, calculate values from raw data in folder
for i = 1:N
% Read in data

filename = sprintf('%s/q%1.0fh%1.0f.csv', dat(i), qs(i), heads(i));

% Use cell arrays

[Pin, Pout, Pdiff, Q, n, P, t, Date, Time, Loggers, Plab] = readCSVData(filename);

APPENDIX G. MATLAB SCRIPTS
Q(Q<0) = 0;
Pdiff = abs(Pdiff); % Remove missing data values correction
T = Tcorr(T);
P = Tcorr(P);
n = Tcorr(n);

% Head
rho(i) = XSteam('rho_pT', Plab, mean(T));
H = Pdiff * 10^5 / (rho(i) * g) + (Q(i)^2 / (2 * g * da));
H(H<0) = 0;

temp = {Pin, Pout, Pdiff, Q, n, T, t, H, Plab, Date, Time, Loggers};
data{i} = temp; % data measurement # parameter # data index

clear Pin Pout Pdiff Q n T t H Plab Date Loggers temp Time file index a b filename

% Analysis
for i = 1:N
    heads(i) = heads(i) / 10;

    fprintf('--------------------------------------------------------
');
    fprintf('Measurement no %1.0f: H=%1.0f, Q=%1.0f 
', i, heads(i), Qs(i));
    fprintf('--------------------------------------------------------
');
    for j = 1:8
        [eMax(i, j), f1{i, j}] = xError(data{i}{j}, 0.05); % Max statistic error and error distr.
        f{i, j} = (f{i, j})^2 + calErr(j)^2.^(0.5);
        eMax(i, j) = sqrt(eMax(i, j)^2 + calErr(j)^2);
        avg{i}{j} = mean(data{i}{j}); % Averaged primary parameters
        if j == 4 % Estimating the flow rate error
            js(1:length(f{i, 1})) = 0.005;
            f{i, j} = js;
            eMax(i, j) = max(f{i, j});
        elseif j == 5 % Estimating the rpm error from the loadbank
            js(1:length(f{i, 1})) = 0.05;
            f{i, j} = js;
            eMax(i, j) = sqrt(max(f{i, j})^2 + calErr(j)^2);
        elseif j == 6 % Estimating the power error from the loadbank
            js(1:length(f{i, 1})) = 0.05;
            f{i, j} = js;
            eMax(i, j) = sqrt(max(f{i, j})^2 + calErr(j)^2);
        end

end
if j ~= 8
    fprintf('Max tot error %5s = %0.5f \n', Var{j}, eMax(i, j));
end

% Head error
avg{i}{9} = mean(data{i}{9});
kl = rho(i) * data{i}{4} .* data{i}{3} * da;
f{i, 9} = ((1 ./ (0.5 + k + 1)) .* 2 + f{i, 3} .* 2 + (1 ./ (1 ./ k + 0.5)) .* 2 + f{i, 4} .* 2) .^ (0.5);
fH(i) = max(f{i, 9}); % Max uncertainty
eMax(i, 9) = fH(i);
fprintf('Max tot error %5s = %0.5f \n', Var{9}, fH(i));

% Qed error
avg{i}{10} = mean(qed{i});
f{i, 10} = (f{i, 4} .* 2 + 0.5 * f{i, 9}) .^ (0.5);
fQed(i) = max(f{i, 10}); % Max uncertainty
eMax(i, 10) = fQed(i);
fprintf('Max tot error %5s = %0.5f \n', 'Qed', fQed(i));

% Ned error
ned{i} = data{i}{4} ./ (D^2 * (g * data{i}{9}) .^ (0.5));
avg{i}{11} = mean(ned{i});
f{i, 11} = (f{i, 5} .* 2 + 0.5 * f{i, 9}) .^ (0.5);
fNed(i) = max(f{i, 11}); % Max uncertainty
eMax(i, 11) = fNed(i);
fprintf('Max tot error %5s = %0.5f \n', 'Ned', fNed(i));

% Eta error
kl = rho(i) ./ (da + 2);
etata{i} = data{i}{6} ./ ((data{i}{4} ./ (data{i}{3} .* 100000 + (data{i}{4} .* da ./ 2))));
% etata{i} = data{i}{6} ./ (data{i}{4} .* data{i}{9} .* rho(i) + g);
avg{i}{12} = mean(eta{i});
f{i, 12} = (f{i, 6} .* 2 + (3 * kl .* data{i}{4} .* 2 + data{i}{3}) ./ ((data{i}{3} + kl .* data{i}{4} .* 2) .* f{i, 4} .* 2 + (1 + data{i}{4} .* 2 + kl .* data{i}{3}) .* f{i, 3} .* 2) .^ (0.5);
fEta(i) = max(f{i, 12}); % Max uncertainty
eMax(i, 12) = fEta(i);
fprintf('Max tot error %5s = %0.5f \n', 'Eta', fEta(i));
clear j i k ts s js kl
end

% Collecting values
% Creating matrices of all averaged points of each measurement
for i = 1:N
    H(i) = avg[i][9];
    Q(i) = avg[i][4];
    Eta(i) = avg[i][12];
    Qed(i) = avg[i][10];
    Ned(i) = avg[i][11];
    fQ(i) = eMax(1,4);
    P(i) = avg[i][6];
    n(i) = avg[i][5];
    fN(i) = eMax(1,5);
    fP(i) = eMax(1,6);
    for j = 1:8
        measErr1(i,j) = max(i1[i][j]);
    end
end

% Measurement uncertainty
fprintf('--------------------------------------------------
');
fprintf('Measurement and total uncertainty
');
fprintf('--------------------------------------------------
');

for j = 1:12
    TotErr(j) = mean(eMax(:,j));
    if j<=8
        measErr(j) = mean(measErr1(:,j));
    else
        measErr(j) = 0;
    end
    if j ~= 8
        fprintf('5 Measurement error = %.2f %%, Total error = %.2f %%
', Var[j], measErr(j)*100, TotErr(j)*100);
    end
end

elseif READ==1 % If short Excel file exist, open this instead
    [Q, n, P, H, Eta, Qed, Ned, rho, fQ, fN, fP, fH, fEta, fQed, fNed] = ReadExcel(dat);
end

% Affinity law estimation
j = 8; % Index number for calculation
n1 = n[j];
Q1 = Q[j];
P1 = P[j];
H1 = H[j];

for i = 1:length(Q)
n2Q(i) = n1 * Q(i) / (Q1);
P2(i) = P1 * (n(i)^3) / (n1^3);
H2(i) = H1 + n2Q(i)^2 / n1^2;
Eta2(i) = P2(i) / (rho(i) * g * Q(i) * H2(i));
n2H(i) = sqrt(n1^2 * H(i) / H1);
Q2(i) = Q1 * n(i) / n1;

% Estimating the hydraulic efficiency
Qm = 4 * Q(i) / ((pi * (D^2 - 0.3^2)));
% Cu2 = Qm / (tand(58.67));
d = linspace(0.3, D, 15);
for w = 1:15
  u(w) = n(i) * d(w) * pi / 60;
  cu2 = 0.9 * g * H(i) / u(w) + u(w) * (1 - (500 * Q(i)) / (n(i) * 0.7));
  b3(w) = tand(Qm / u(w));
  cu3 = u(w) - Qm / (tand(b3(w));
end
U = mean(u);
Cu2 = mean(cu2);
Cu3 = mean(cu3);
Etah(i) = U / (g * H(i)) * (Cu2 - Cu3);
Pt(i) = Etah(i) * 0.958 * rho(i) * Q(i) * H(i) * g;

% Plotting
figure();
errorbar(Q, H, fH.*H, fH.*H, fQ.*Q, fQ.*Q, ' . ');
hold on;
plot(Q, H, 'or');
title('Head vs Volume Flow');
xlabel('Volume flow, Q [m^3/s]');
ylabel('Head, H [m]');

figure();
hold on;
plot(H, sort(Pt), 'g--', 'Color', [0 0.6 0])
legend('Uncertainty', 'Measured value', 'Theoretical value', 'Location', 'Northwest');
% Qed - Ned
figure();
hold on;
plot(Ned, Qed, 'or')
title('Qed vs Ned');
xlabel('Speed Number, Ned [-]');
ylabel('Specific Flow Rate, Qed [-]');
plot(35.7,0.4,'sm');
legend('Uncertainty', 'Measured value', 'BEP', 'Location', 'Northwest');
ylim([0.25 0.42]);

% Eta - Ned
figure();
hold on;
plot(Ned, Eta, 'or')
title('Total Efficiency vs Ned');
xlabel('Speed factor, Ned [-]');
ylabel('Total efficiency, \eta [-]');
plot(35.7,0.861,'sm');
legend('Uncertainty', 'Measured value', 'Design point', 'Location', 'Northwest');

% Combined - Hill
figure();
G=[Ned' Qed' Eta' fEta' fNed'];
G=sort(G);
Ned1=(G(:,1))'
Eta1=(G(:,3))'
fEta1=(G(:,4))'
fNed1=(G(:,5))'

yyaxis left;
hold on;
plot(Ned,Qed,'or')
xlabel('Speed factor, Ned [-]');
ylabel('Flow factor, Qed [-]');
plot(35.7,0.4,'sm')

yyaxis right;
%errorbar(Ned1,Eta1,fEta1.*Eta1,fEta1.*Eta1,fNed1.*Ned1,fNed1.*Ned1,'or');
hold on;
plot(Ned1,Eta1)
xlabel('Speed factor, Ned [-]');
ylabel('Total efficiency, \eta [-]');
title('Performance Chart');
legend('Uncertainty', 'Qed', 'Design Qed', 'Measured Efficiency', 'Location', 'Northwest');
% Hill diagram
% hill(Ned,Qed,Eta,n,5,[0.82:0.05:0.87],0.002);

% Create a short Excel-file
if READ=0
    B=[Q' , n' , P' , H' , Eta' , Qed' , Ned' , rho ' ];
    C=A;
    for l=1:size(B,1)
        for y=1:size(B,2)
            C{l+1,y}=B{1,y};
        end
    end
    C{2,9} = data{1}{11};
    C{2,10}= data{1}{12};
    C{2,11}= data{1}{13};
    DD= {'Q' , 'n' , 'P' , 'H' , 'Eta' , 'Qed' , 'Ned' };
    E=[fQ' , fN' , fP' , fH' , fEta' , fQed' , fNed' ];
    for l=1:size(E,1)
        for y=1:size(E,2)
            DD{l+1,y}=E{1,y};
        end
    end
    xlswrite(sprintf('Tests\test%s.xls',dat),C,1); %Values
    xlswrite(sprintf('Tests\test%s.xls',dat),DD,2); %Errors
end

calibration.m

clc;
clear all;
close all;

% Open measurement files
[-, ~, raw] = xlsread('C:\Users\chrismee\OneDrive\Dokumenter\MATLAB\Master\QPKalibrering.xlsx',
    'QPKalibrering','A2:F23');
data = reshape([raw{:}],size(raw));

% Variables
leg={'Q' , 'P' , 'n' };
dim={'[m^3/s]', '[W]', '[rpm]'};

for i=1:3
    M(:,i)=data(:,i);
    V(:,i)=data(:,i+3);
end

M(:,3)=M(:,3)*7.5;

clearvars data raw;

for j=1:3
    fprintf('%s calibration:
', leg{j});

    %Regression
    [r,s,o]=regression(V(:,j)',M(:,j)');
    fprintf('%s = %1.7f * v + %1.7f
R=%1.4f
', leg{j}, s, o, r);

    %Regression error
    [err,f]=xyError(V(:,j)',M(:,j)',0.05,o);
    fprintf('Maximum regression error = %.4f
', err);

    x=linspace(0,max(V(:,j)),size(V,1));
    y=x*s+o;

    %Plots calibration line
    figure();
    plot(V(:,j)',M(:,j)', 'x',x,y,x+f,x,y-f);
    title(sprintf('%s calibration', leg{j}));
    xlabel('Voltage, [V]');
    ylabel(sprintf('%s, %s', leg{j}, dim{j}));
    legend('Measurements', 'Regression line', 'Lower bound 2.5 % conf', 'Upper bound 97.5 % conf');
    fprintf('

end

hill.m

function hill(Ned1,Qed1,nh1,n1,n,n1,r,off)

% NN=unique(n);
% nn=length(NN); %Number of ns
nhead=length(nh1)/nn;

APPENDIX G. MATLAB SCRIPTS

```matlab
clear NN;

for i=1:n
    for j=1:nhead
        Qed(j,i)=Qed1((j+(i-1)*nhead));
        Ned(j,i)=Ned1((j+(i-1)*nhead));
        nh(j,i)=nh1((j+(i-1)*nhead));
        n(j,i)=n1((j+(i-1)*nhead));
    end
    leg{i}=sprintf('n= %3.0f',n(1,i));
end

leg{length(leg)+1}='Design point';
leg{length(leg)+1}='True BEP';

Qed_d=0.4;
Ned_d=35.7;

[mx, ind]=max(nh1);
Qed_x=Qed1(ind);
Ned_x=Ned1(ind);

yyaxis left;
plot(Ned1, Qed1, '* ',Ned_d, Qed_d, 'mx ',Ned_x, Qed_x, 'c+ ');
xlabel('Ned');
ylabel('Qed');

yyaxis right;
plot(Ned1, nh1);
ylabel('nh');
title(sprintf('Parametric plot %1.0f, eta_{h,max}= %0.3f',res,mx));
legend('Qed', 'Design point', 'BEP', 'Efficiency', 'Location', 'Northwest ');

for i=1:length(nh1)-1
    nhlm(i)=(nh1(i)+nh1(i+1))/2;
    Qedlm1(i)=(Qed1(i)+Qed1(i+1))/2+off;
    Qedlm2(i)=(Qed1(i)+Qed1(i+1))/2-off;
    Nedlm(i)=(Ned1(i)+Ned1(i+1))/2;
end

figure()
plot(Ned,Qed,Ned_d,Qed_d,'mx ',Ned_x,Qed_x,'c+ ', 'Linewidth', 1.5);
hold on
[C, h]=contour([Ned1, Qed1, Ned_d, Qed_d, 'mx ',Ned_x, Qed_x, 'c+ ', 'ShowText', 'on', 'LabelSpacing', 300, 'HandleVisibility', 'off ']);

% title(C, 'manual', 'color', 'k', 'background', 'w', 'edgecolor', 'k');

legend(leg, 'Location ', 'Northwest')
```

HillFromXl.m

```matlab
cle; 
clear all; 
close all; 

% Constants 
g=9.82146516; 
D=0.3; 

res=8; % Choose simulation result 

% Read from Excel 
switch res 
case 1 
[-, -, raw] = xlsread('C:\Users\chrismee\OneDrive\Dokument\MATLAB\Master\Para.xlsx', 'Dr0.693', 'B8 :E31'); 
case 2 
[-, -, raw] = xlsread('C:\Users\chrismee\OneDrive\Dokument\MATLAB\Master\Para.xlsx', 'Dr0.729', 'B8 :G31'); 
case 3 
[-, -, raw] = xlsread('C:\Users\chrismee\OneDrive\Dokument\MATLAB\Master\Para.xlsx', 'Zgv19', 'B7 :G30'); 
case 4 
[-, -, raw] = xlsread('C:\Users\chrismee\OneDrive\Dokument\MATLAB\Master\Para.xlsx', 'n1600', 'B8 :D22'); 
case 5 
[-, -, raw] = xlsread('C:\Users\chrismee\OneDrive\Dokument\MATLAB\Master\Para.xlsx', 'H19', 'B8:D19'); 
case 6 
[-, -, raw] = xlsread('C:\Users\chrismee\OneDrive\Dokument\MATLAB\Master\Para.xlsx', 'Par1mass', 'A2:C6'); 
case 7 
[-, -, raw] = xlsread('C:\Users\chrismee\OneDrive\Dokument\MATLAB\Master\Para.xlsx', 'n1400', 'B8 :I19'); 
case 8 
[-, -, raw] = xlsread('C:\Users\chrismee\OneDrive\Dokument\MATLAB\Master\Para.xlsx', 'q168', 'B8 :F19'); 
end 

% Create output variables 
data = reshape([raw{:}], size(raw)); 

% Read from Excel 
switch res 
```
APPENDIX G. MATLAB SCRIPTS

```matlab
case 1
    H = data(:,1);
    n = data(:,3)/pi*30;
    nh1 = data(:,4);
    Q(1:length(H))=0.142; %0.142;
end

case 2
    H = data(:,1);
    n = data(:,2)/pi*30;
    nh1 = data(:,6);
    Q=data(:,3)/1000;
end

case 3
    H = data(:,1);
    n = data(:,2)/pi*30;
    nh1 = data(:,4);
    Q=data(:,3)/1000;
end

case 4
    H = data(:,1);
    n = data(:,2)/pi*30;
    nh1 = data(:,3);
    Q(1:length(H))=160/1000;
end

case 5
    H = data(:,2);
    n = data(:,1)/pi*30;
    nh1 = data(:,3);
    Q(1:length(H))=160/1000;
end

case 6
    H = data(:,1);
    n = data(:,2)/pi*30;
    nh1 = data(:,3);
    Q(1:length(H))=160/1000;
end

case 7
    H = data(:,1);
    n = data(:,2)/pi*30;
    nh1 = data(:,8);
    Q=data(:,5)/1000;
end

case 8
    H = data(:,1);
    n = data(:,2)/pi*30;
    nh1 = data(:,3);
    Q=data(:,4);
end

Hn=unique(H);
N=length(Hn); %Number of heads
nn=unique(n);
NN=length(nn); %Number of ns

clearvars data raw Hn nn;
```
% Plots

for i = 1:length(Q)
    Qed1(i) = Q(i) / (D^2 * sqrt(g * H(i)));
    Ned1(i) = n(i) * D / sqrt(g * H(i));
end

Qed_d = 0.160 / (D^2 * sqrt(g * 18));
Ned_d = 1500 * D / sqrt(g * 18);

A = [Ned1' Qed1' nh1];
A = sortrows(A);

[mx, ind] = max(nh1);
Qed_x = Qed1(ind);
Ned_x = Ned1(ind);

fprintf('Best Punkt:\n');
fprintf('H = %2.0f \n', H(ind));
fprintf('n= %4.0f \n', n(ind));
fprintf('nh= %1.4f \n', mx);
leg = {

switch res
    case 6
        Qed = Qed1;
        Ned = Ned1;
        nh = nh1;
        for i = 1:5
            leg{i} = sprintf('n= %3.0f', n(i));
        end
    case 8
        Ned1 = A(:, 1);
        Qed1 = A(:, 2);
        nh1 = A(:, 3);
        for i = 1:4
            for j = 1:3
                Qed(j, i) = Qed1(j + (i - 1) * 3);
                Ned(j, i) = Ned1(j + (i - 1) * 3);
                nh(j, i) = nh1(j + (i - 1) * 3);
                n1(j, i) = n(j + (i - 1) * 3);
            end
            leg{i} = sprintf('n= %3.0f', n1(1, i));
        end
    otherwise
        for i = 1:NN
            for j = 1:NH

APPENDIX G. MATLAB SCRIPTS
Qed(j,i)=Qed1(j+(i-1)*NH);
Ned(j,i)=Ned1(j+(i-1)*NH);
nh(j,i)=nh1(j+(i-1)*NH);
nl(j,i)=nl(j+(i-1)*NH);
end
leg{i}=sprintf('n= %3.0f',nl(1,i));
end
end

leg{length(leg)+1}='Design point';
leg{length(leg)+1}='True BEP';

if res==6
    yyaxis left;
    plot(Ned, Qed, 'x', Ned_d, Qed_d, 'mx');
    xlabel('Speed factor, Ned [-]');
    ylabel('Flow factor, Qed [-]');
    yyaxis right;
    plot(Ned, nh);
    ylabel('Hydraulic efficiency, \eta_h [-]');
    title(sprintf('Parametric Plot %1.0f', res));
    legend('Qed', 'Design point', 'Efficiency', 'Location', 'Northwest')
elseif res==8
    yyaxis left;
    plot(Ned1,Qed1,'x', Ned_d,Qed_d, 'mx', Ned_x,Qed_x, 'c+');
    xlabel('Speed factor, Ned [-]');
    ylabel('Flow factor, Qed [-]');
    yyaxis right;
    plot(Ned1,nh1);
    ylabel('\eta_h');
    title(sprintf('Parametric Plot %1.0f', res));
    legend('Qed', 'Design point', 'BEP', 'Efficiency', 'Location', 'Northwest')
figure()
off=0.002; %Offset
for i=1:length(nh1)-1
    nhlm(i)=(nh1(i)+nh1(i+1))/2;
    Qedlm1(i)=(Qed1(i)+Qed1(i+1))/2+off;
    Qedlm2(i)=(Qed1(i)+Qed1(i+1))/2-off;
    Nedlm(i)=(Ned1(i)+Ned1(i+1))/2;
end
figure()
plot(Ned,Qed,Ned_d, Qed_d, 'mx', Ned_x,Qed_x, 'c+', 'Linewidth',1.5);
hold on
```matlab
[C, h] = contour([Ned1m' (Ned1(1: length(Ned1)-1)) NedIm'], [Qed1m' (Qed1(1:length(Qed1)-1)) QedIm2'], [nh1m' ( nh1(1:length(Ned1)-1)) nh1m'], [0.86:0.004:0.885], 'ShowText', 'off', 'LabelSpacing', 300, 'HandleVisibility', 'off');

xlabel('Speed factor, Ned [-]');
ylabel('Flow factor, Qed [-]');
title(sprintf('Hill Chart Parametric %1.0f\%', res))
t = clabel(C, 'manual', 'color', 'k', 'background', 'w', 'edgecolor', 'k', 'FontSize', 8, 'Margin', 1);
plot(NaN, NaN, 'ks');

if length(leg) + 1

else

fig = figure();
plot(Ned, Qed, Ned_d, Qed_d, 'mx', Ned_x, Qed_x, 'c+', 'Linewidth', 1.5);
hold on;

[C, h] = contour(Ned, Qed, nh, 6, 'ShowText', 'off', 'LabelSpacing', 300, 'HandleVisibility', 'off');
plot(NaN, NaN, 'ks');

legend(leg, 'Location', 'Northwest');

end

% Extrapolation

figure();
plot(Ned, Qed, Ned_d,Qed_d, 'mx', Ned_x, Qed_x, 'c+', 'Linewidth', 1.5);
legend(leg, 'Location', 'Northeast');

hold on;
F = scatteredInterpolant(Ned1, Qed1, nh1);
[N, Q] = meshgrid(min(Ned1):0.002:max(Ned1), min(Qed1):0.0001:max(Qed1));
eta = F(N, Q);
contour(N, Q, eta, 8, 'ShowText', 'on', 'HandleVisibility', 'off');
xlabel('Speed factor, Ned [-]');
ylabel('Flow factor, Qed [-]');

Initializer.m

clc;
clear all;
```
close all;

% Parameters

D=0.3; %Outer diameter [m]
H=17; %Head [m]
Q=0.160; %Flow rate [m^3/s]

Hs=2; %Suction height [m] (height between turbine outlet and free water surface. Negative when submerged)

% Constants

g=9.82146516;

nh=0.9;
rho=998;

ns=[500:100:1600]; % rpms
N=20; %Radial blade elements

% From "Design of propeller turbines for pico hydro" => set=1
nq=[75 100 125 175 212.5];
zb=[8 7 6 5 4];
dd=[0.68 0.63 0.58 0.5 0.47];

% From Kaplan– NINJ => set=2
n_s=[300 400 500 600 700 800 900];
dd1=[0.65 0.575 0.51 0.45 0.42 0.42 0.45];

% From "Hydraulic Design of Hydraulic Machinery" => set=3
Nq=1059.2*H^(-0.625); %should be 140–400 for Bulb

n_in=Nq*H^0.75/sqrt(Q);
dD_in=0.443-2.2*10^(-4)*Nq;

set=1; %Select empirical d/D-equation

% Finds ideal rpm

fprintf('+++Design Parameters+++\n');
fprintf('H= %2.0f m, Q= %1.2f m^3/s\n',H,Q);
fprintf('Initial guesses: Nq= %3.1f, n= %3.1f, d/D= %1.3f\n',Nq,n_in,dD_in);
fprintf('Iterating...\n');
n=0;
nl=n_in;

%Iteration

while abs(nl-n)>0.01
    nl=n;

Ku=1+0.0038*Nq;
n=Ku*60*sqrt(2*g*H)/(pi*D);
Nq=n1*sqrt(Q)/(H^0.75);
end

%Finds d/D
if set==3
dD=0.443-2.2*10^(-4)*Nq;
elseif set==1
dD=interp1(nq,dd,Nq,'pchip','extrap');
elseif set==2
dD=interp1(n_s,dd1,Nq*3,'pchip','extrap');
else
disp('Error')
return
end

fprintf('Nq= %3.1f, n= %3.1f, d/D= %1.3f

');
fprintf('Correcting to closest even rpm...
');
diff=1000;
ni=n;

%Finds closest rpm
for i=1:length(ns)
    if abs(ni-ns(i))<diff
        n=ns(i);
        diff=abs(ni-ns(i));
    end
end

Nq=n*sqrt(Q)/(H^0.75);

%Recalculates d/D
if set==3
dD=0.443-2.2*10^(-4)*Nq;
elseif set==1
dD=interp1(nq,dd,Nq,'pchip','extrap');
elseif set==2
dD=interp1(n_s,dd1,Nq*3,'pchip','extrap');
else
disp('Error')
return
end

d=dD*D;
Cm=4*Q/(pi*(D^2-d^2));
dr=linspace(d,D,N);
Z=interp1(nq,zb,Nq,'linear','extrap');
Z=round(Z);
% Calculates angles and velocity components

for e = 1:N
    u(e) = n * dr(e) * pi / 60;
    cu2(e) = nh * g * H / u(e);
    b2(e) = atand (Gn / (u(e) - cu2(e)));
    if b2(e) < 0
        b2(e) = b2(e) + 180;
    end
    b3(e) = atand (Gn / u(e));
    if b3(e) < 0
        b3(e) = b3(e) + 180;
    end
    a(e) = atand (Cm / cu2(e));
    w2 = sqrt (Cm^2 + u(e)^2);
    bm(e) = mean([b2(e) b3(e)]);
end

Umax = max(u);

rhs = (1e5 - 2.34e3) / (rho * g) - (1.1 * Cm^2 / (2 * g) + 0.225 * Umax^2 / (2 * g)); % Cavitation

if Hs > rhs
    cav = 'fail';
else
    cav = 'ok';
end

Cu2 = mean(cu2);
U = mean(u);

C2 = (Cu2.^2 + Cm.^2).^0.5;
drawTriangle(U,Cm,Cu2,n); % Velocity triangle

P2 = rho * g * H - 0.5 * rho * C2.^2;

Pr = P2 / (rho * g * H); % Reaction ratio

% Pr = 2 * (mean(u) / sqrt(2 * g * H)) * (mean(cu2) / sqrt(2 * g * H)) - (mean(cu2) / (sqrt(2 * g * H)))^2;

UCu = U / C2;
W = mean(w2);

Um = mean(bm); % Mean angle of attack
B3 = mean(b3);
B2 = mean(b2);

fprintf('
% Alternatives:
');

fprintf('n= %4.0f, d/D= %0.3f, Zb= %2.0f, Nq= %3.2f, R= %3.2f%%, U/C2= %1.2f, U/W2= %1.2f, b2= %1.2f,
        b3= %1.2f, Cavitation: %s \n
', n, d/D, Zb, Nq, R, U/C2, U/W2, b2, b3, cav);

fprintf(' Alternatives:
');
n_set = n;

% Alternatives

for i = 1:length(ns)
    if ns(i) == n_set
        n = ns(i);
        Nq = n * sqrt(Q) / (H^0.75);
% d/D ratio
if set==3
dD=0.443−2.2*10^(-4)*Nq;
elseif set==1
dD=interp1(nq,dd,Nq,'pchip','extrap');
elseif set==2
dD=interp1(n_s,dd1,Nq+3,'pchip','extrap');
else
disp('Error')
return
end

if ns(i)==1400
    dD=0.693;
end

d=dD*D;
C_m=4*Q/(pi*(D^2−d^2));

dr=linspace(d,D,N);
Z=interp1(nq,zb,Nq,'linear','extrap');
Z=round(Z);

% Blade angles and velocities
for e=1:N
    u(e)=n*dr(e)*pi/60;
    cu2(e)=nh*g*H/u(e);
    b2(e)=atand(C_m/(u(e)−cu2(e)));  
    if b2(e)<0
        b2(e)=b2(e)+180;
    end
    b3(e)=atand(C_m/u(e));
    if b3(e)<0
        b3(e)=b3(e)+180;
    end
    w2=sqrt(C_m^2+u(e)^2);
end

Umax=max(u);

rhs=(1e5−2.34e3)/(rho*g)−(1.1*C_m^2/(2*g)+0.225*Umax^2/(2*g)); % Cavitation
if Hs > rhs
    cav='fail';
else
    cav='ok';
end

Cu2=mean(cu2);
U=mean(u);
drawTriangle(U,Qn,Cu2,ns{i});
W=mean(w2);
APPENDIX G. MATLAB SCRIPTS

U=W/ U;
UCu=U/C2;
C2=(Cu2.^2+Cm. ^ 0.5);
P2=rho* g*H−0.5*rho* C2. ^ 2;
Pr=P2/(rho* g*H);

Pr=2*(mean(u) / sqrt(2* g*H)) * (mean(cu2) / sqrt(2* g*H)) − (mean(cu2) / (sqrt(2*g*H))) ^ 2;
B3=mean(b3);
B2=mean(b2);

fprintf(‘n= %4.0f, d/D= %0.3f, Zb= %2.0f, Nq= %3.2f, R= %3.2f%%, U/C2= %1.2f, U/W 2= %1.2f, b2= %1.2f, b3= %1.2f, Cavitation: %s 
’,n,d/D,Z,Nq, Pr *100, U/C2, U/W 2, b2, b3, cav);
end

end

spreadfigures();