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Eve Cathrin Walseth

Dynamic Behavior of Reversible Pump-Turbines in Turbine Mode of Operation

Doctoral Thesis

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Faculty of Engineering Science and Technology
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Thesis for the degree of Philosophiae Doctor

Trondheim, July 2016

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If we knew what we were doing
it would not be called research,
would it?
Albert Einstein

Preface

The work has been conducted at the Waterpower Laboratory, Department of Energy and Process Engineering at the Norwegian University of Science and Technology (NTNU) in Trondheim. Professor Torbjørn K. Nielsen has been the main supervisor and Bjørnar Svingen from Rainpower the co-supervisor.

The work has been funded by the Centre for Environmental Design of Renewable Energy (CEDREN) through the Norwegian Research Council.

Abstract

Reversible pump-turbines, with their steep and s-shaped characteristics, have proven throughout history to be a challenge with regards to both stability and transients. The aim of this thesis is to increase our understanding of the dynamic behavior of these machines in turbine mode of operation. The work comprises of three objectives. The first is to investigate and verify stability criteria for hydraulic systems equipped with reversible pump-turbines. An investigation of the influence of the characteristics on the system transients forms the second objective. The third objective is to improve and verify an analytical one-dimensional reversible pump-turbine model for transient calculations.

The research methods utilized in this work includes laboratory and field measurements combined with transient calculations. Measurements from the laboratory and prototype are used to investigate and verify stability criteria, while the prototype measurements are also used to explore the characteristics impact on the transients. Transient calculations are used to better explain phenomena revealed in the measurements, and to further explore the dynamic behavior of the hydraulic systems. In addition, the laboratory measurements combined with transient calculations are used to verify the improved one-dimensional reversible pump-turbine model.

The findings from the research show that the theoretical stability criterion consists of two inequalities; one for a negative T_{ed}-N_{ed}-gradient at runaway and one for a positive. The inequality for a negative gradient gave a correct prediction for the laboratory system, while the criteria for stable behavior with a positive gradient could not be verified for the prototype. Transient calculations revealed that the system was at the stability limit.

The investigation of the impact of the characteristics on the system transients showed that, for the prototype, the turbine was the major contributor.

In the research presented the importance of transient analysis at an early stage of a project has been emphasized. This has motivated further development of a one-dimensional reversible pump-turbine model improving the correspondence with measured characteristics.

Based on these findings it is recommended that future work includes measurements of the dynamic characteristics on a prototype for comparison with predicted theoretical behavior. Further improvement of the one-dimensional model is recommended, suggested work includes both analytical studies and measurements.

Acknowlegdements

An old African proverb states that ''*it takes a village to raise a child*". My experience is that it also takes a village to write a PhD-thesis. I would like to express my gratitude to my supervisors, Professor Torbjørn Nielsen and Dr.Ing Bjørnar Svingen for invaluable discussions and motivation during the work with this thesis.

Thanks to CEDREN for financing my work and Professor Ånund Killingtveit for believing in my work and providing additional funding during the completion of this thesis.

A major contributor to my work has been the good working environment at the Waterpower Laboratory. My colleagues there have always been supportive and willing to help. I especially want to thank Wenche, Halvor, Joar, Trygve, Bård and Ole Gunnar, and my former PhD-collegues Einar, Håkon, Pål-Tore, Pål Henrik, Mette and Bjørn.

I would like to express my gratitude to Bjørn Åril, who has been my mentor at Rainpower and has taught me almost everything I know about practical challenges with regards to system dynamic and reversible pump-turbines. Without his catching enthusiasm and knowledge transfer this thesis would never have been completed.

My greatest appreciation to Rainpower, who has provided me with the necessary data and time to complete this thesis. A special thanks to my colleagues at the System Department, for their support and for interesting technical discussions.

Throughout the years of working with this thesis, my friends and family have been very supportive and extremely patient. I will always be grateful. A special thank you to Grunde Olimstad for interesting discussions at all hours of the day, cooperation and encouraging talks during the work with this thesis. Jørgen Ramdal, for always being positive and an oracle with regards to laboratory and field measurements. Bente Brunes for being the best and toughest proofreader I know and for being available 24/7 to cheer me up during the finishing stage of this thesis.

Last, but not least, my husband Bjørn Lage. Thank you for your patience, positive spirit and for marrying me before the thesis was completed. I don't think you had any idea of what you got yourself into, but you've never complained. That is nothing but impressive, and I promise that I will try my hardest to make it up to you.

List of publications

Selected papers – Main author

1. **Walseth, E.C.**, Nielsen T.K., Svingen B., "Measuring the Dynamic Characteristics of a Low Specific Speed Pump–Turbine Model", *Energies* 2016, 9, 199.
2. **Walseth, E.C.**, Nielsen T.K., Svingen B., "Investigation of Stability Criteria for Reversible Pump–Turbines with Laboratory and Prototype Measurements"', submitted.
3. **Walseth, E.C.**, Nielsen T.K., Svingen B., "Prototype Study on the Effect of Reversible Pump–Turbine Characteristics on System Transients", submitted.

Secondary papers – Main author

1. **Walseth, E.C.**, Svingen B., Nielsen T.K., "Investigating the effect of turbine characteristics on the pressure response of a system", *4-th International Meeting on Cavitation and Dynamic Problems in Hydraulic Machinery and Systems*, Belgrade, Serbia, October 26-28, 2011
2. **Walseth, E.C.**, Nielsen T.K., Svingen B., "Investigation of a 1D-model for simulating the characteristics of a high head Francis turbine", *14th International Symposium on Transport Phenomena and Dynamics of Rotating Machinery ISROMAC*, Honolulu, HI, USA, February 27th-March 2nd,2012.
3. **Walseth, E.C.**, Svingen B., Nielsen T.K., "Comprehensive experimental study of instability in a reversible pump-turbine model at no-load operation",*Hydro 2012 Innovative Approaches to Global Challenges*, Bilbao, Spain, October 29-31, 2012
4. **Walseth, E.C.**, Svingen B., "Upgrading high head power plants. Footprint measurements", *Hydro 2013 Promoting the Versatile Role of Hydro*, Innsbruck, Austria, October 7-9, 2013

Secondary papers – Co-author

1. **Svingen B.**, Luraas H., Walseth E.C., "Frequency response measurements and calculations with Water Column Compensation and Pressure Feedback", *4-th International Meeting on Cavitation and Dynamic Problems in Hydraulic Machinery and Systems*, Belgrade, Serbia, October 26-28, 2011

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List of Symbols

A	Area (m^2)
a	Acoustic velocity (m/s)
b_1	Linear slope of the dimensionless torque-flow characteristics(-)
D	Diameter (m)
f	Friction factor (-)
g	Gravitational acceleration (m/s^2)
H	Head (m)
h	Dimensionless head (-)
I	Inertia (m^2/s^2)
n	Rotational speed (rev/s)
N_{ed}	Dimensionless rotational speed (-)
Q_{ed}	Dimensionless flow (-)
Q	Flow rate (m^3/s)
q	Dimensionless flow rate (-)
r	Radius (m)
R_m	Mechanical loss coefficient (-)
T_a	Time constant rotating masses (s)
T_{ed}	Dimensionless torque (-)
T_{osc}	Period of oscillations (s)
T	Torque (Nm)
T_w	Time constant of water masses in penstock (s)
T_{wt}	Time constant of water masses in the turbine (s)
t	Time (s)
v	Velocity (m/s)
β	Runner blade angle (-)
η	Turbine efficiency (-)
η_g	Efficiency generator (-)
ρ	Density of water (kg/m^3)
τ_f	Ratio of time constant of the waterway on the time constant of the rotating masses (-)
Ω	Dimensionless rotational speed (-)
ω	Rotational speed (rad/s)

Sub- and superscript

dyn	Dynamic
max	Maximum
net	Net
n	Nominal value
opt	Best efficiency point
r	Rated value
0	Initial value
1	Runner inlet
2	Runner outlet
*	Best efficiency

Derived parameters one-dimensional turbine model

Δh	Dimensionless hydraulic loss
$\widetilde{m_s}$	Dimensionless start torque
R_p	$\frac{r_1}{A_1 \tan \beta_1} - \frac{r_2}{A_2 \tan \beta_2}$, machine constant
r_p	$\frac{R_p \omega_n Q_n}{g H_n}$, machine constant
γ	$1 - \widetilde{m_s} + \psi + r_p$, machine constant
κ	Guide vane opening degree
σ	$\frac{\eta_{max} - \psi}{\eta_{max} + \psi}$, machine constant
ψ	$\frac{r_2^2 \omega_n^2}{g H_n}$, machine constant

Part I

Chapter 1

Introduction

1.1 Background

Climate change has necessitated worldwide climate goals intensifying the development and installation of sustainable renewable energy. In 2015, the first universal legally binding climate deal, the Paris agreement, was signed by 195 countries. The European Union (EU) was the first major economy to submit its intended contribution to the agreement. The EU 2030 climate and energy framework aims to cut greenhouse gas emissions by at least 40% by 2030 [1], and to increase the portion of renewable energy in the energy production to 27% [2]. The 2030 framework is a continuation of the EU's Renewable Energy Directives goal of 20% renewables within 2020, which spurred on the installation of wind and solar power plants in Europe.

The increased amount of intermittent sources, like wind and solar power, in an energy system requires a greater reserve power capacity, both with regards to supply and storage of energy. A role that hydropower can fill, with its unique ability to store energy both short term and long term in reservoirs. Today hydropower already provide 99 % of the world's electricity storage for grid systems [3], with 220 TWh in Europe alone [4]. This includes the storage capacity from pumped hydro, which offers extra flexibility as these plants can generate, as well as use excess energy to pump water up to reservoirs for storage. The shift in the energy matrix, with a larger portion of intermittent sources, has increased the focus on pumped hydro and in Europe 8600 MW of pumped storage is planned or under construction.

Pumped storage plants can roughly be divided into two types; plants with separate pump and turbine installed and those equipped with reversible pump-turbines. Reversible pump–turbines have the ability to run both as a pump and a turbine, depending on the rotational direction of the runner. This presents certain challenges in the design process where the pump is prioritized. The flow in pump mode is more sensitive to separation, recirculation and losses [5], in addition, the head in pump mode is higher than what is available in turbine mode. As a consequence, the machine works off–design in the entire range of turbine operation and the efficiency in both turbine and pump mode is lower than for an installation with a separate pump and turbine. This loss in efficiency is compensated by a more cost-effective solution with only one machine.

The design compromises made to enable both a pump and a turbine in the same machine is the source to a phenomenon known as s-shaped characteristics in turbine mode of operation. The shape of these characteristics can, in combination with a hydraulic system, result in instability during start–up, load rejection and possible off–design operation [5]. In addition to increased impact on the system transients, compared to a regular Francis turbine. Computational Fluid Dynamic (CFD) tools and model tests in a laboratory combined with transient analysis are tools that can be used to predict and possibly avoid the above mentioned challenges. There are, however, no guarantee that the prototype will behave according to the prediction as there will always be a difference between calculations and real life behavior.

1.2 Objective

Dynamic behavior of reversible pump–turbines involves a wide range of fields from research on flow pattern in the runner, stability challenges on prototypes to runner geometry impact on the characteristics. Aside from these contributions, the physics of reversible pump-turbines are still not fully understood.

The aim of this thesis is to increase our understanding of the dynamic behavior of reversible pump-turbines in turbine mode of operation, by means of laboratory and prototype measurement and transient calculations. This will be achieved through the following objectives:

- O1: Investigate and verify stability criteria for hydraulic systems with reversible pump-turbines
- O2: Investigate the effect of pump-turbine characteristics on the transient behavior of a system
- O3: Improve and verify an analytical one-dimensional reversible pump-turbine model for transient calculations

1.3 Research overview

The research methods applied in this thesis comprises of laboratory and prototype measurements combined with transient calculations. Research contributions from this work are presented in three selected papers, full versions are enclosed in Part II of this thesis:

Paper 1 – Measuring the Dynamic Characteristics of a Low Specific Speed Pump–Turbine Model

E.C. Walseth, T.K. Nielsen and B. Svingen. Published in Energies 2016, 9, 199.
The paper presents measurements of the dynamic characteristics of a reversible pump-turbine model in an open test loop system. Transient calculations are used, together with the measurements, to verify an existing one-dimensional reversible pump-turbine model in accordance with objective O3.

Paper 2 – Investigation of Stability Criteria for Reversible Pump–Turbines with Laboratory and Prototype Measurements

E.C. Walseth, T.K. Nielsen and B. Svingen. Submitted.
The paper presents stability criteria for hydraulic systems with reversible pump-turbines. Prototype measurements are, together with the laboratory measurements presented in Paper 1 and transient calculations, applied for verification of the criteria. The presented research corresponds to objective O1.

Paper 3 – Prototype Study on the Effect of Reversible Pump–Turbine Characteristics on System Transients

E.C. Walseth, T.K. Nielsen and B. Svingen. Submitted.
An investigation of reversible pump-turbine characteristics on the system transients, in accordance with objective O2, is presented in Paper 3. The research methods applied in this paper comprises of prototype measurements and transient calculations.

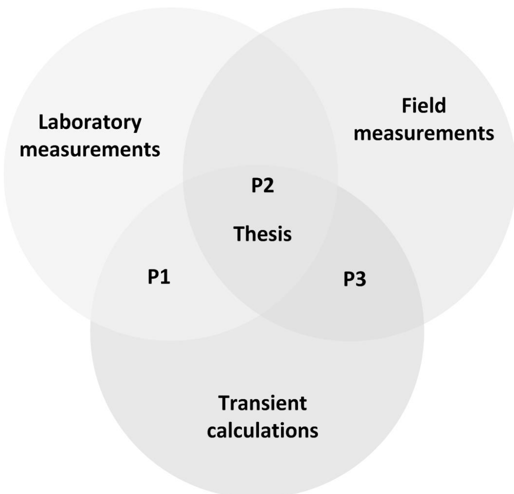


Figure 1.1: Coherence between the thesis, papers and applied research methods

The coherence between the thesis, papers (denoted as P) and the applied research methods are presented in Fig.1.1.

In addition to the three selected papers, a suggested improvement of the one-dimensional reversible pump-turbine model is presented in Chapter 3. The model is discussed and verified with the laboratory measurements in Chapter 5. This work is in accordance with objective O3.

1.4 Contributions

The research contributions in this thesis are presented in detail as described in the previous section, a summary of the main contributions are listed as follows:

- Presented measurements of the dynamic characteristics of a reversible pump-turbine model.
- Verification of stability criterion for hydraulic systems with reversible pump-turbines stating that the system is stable with a negative T_{ed}-N_{ed}-gradient at runaway.
- Stability criterion for a system with a positive T_{ed}-N_{ed}-gradient at runaway shown to not give an accurate prediction.
- Reversible pump–turbine characteristics are shown to be the major contributor to the system transients for the presented prototype.
- Verification of an existing one-dimensional reversible pump-turbine model for transient calculations.
- Presented and verified an improvement of the one-dimensional reversible pump-turbine model.

1.5 Outline of thesis

This thesis is divided into two parts. Part I gives an introduction to the background and objectives of this thesis. An overview of the research and the main contributions are presented, together with a summary of previous work and applied research methods. A short resumè of the three selected papers and a general discussion of the results are given, followed by conclusions and recommendations for future work. Part II consist of full length versions of the three selected papers.

Chapter 2

Previous work

In this section a summary of selected literature used in this work is presented.

2.1 Laboratory measurements

Experiments measuring the dynamic characteristics of a high head Francis turbine model at NTNU was performed by Nielsen in 1990 [6]. For these measurements an open test loop system was used, ensuring that the pressure upstream and downstream the turbine was approximately constant. The characteristics were obtained by disconnecting the generator when the turbine was running at low rotational speed allowing it to go towards runaway. Pressure upstream and downstream the turbine, torque, rotational speed and flow rate was measured, the latter by means of a method resembling the pressure-time method [7]. Nielsen's results showed that the dynamic characteristics deviated significantly from the measured steady-state characteristics. The cause of the deviation was explained by the inertia of the water masses inside the turbine, defined as the inlet of the spiral casing to the outlet of the draft tube. This effect can be removed by redefining the net head, also confirmed by Zeng [8]:

$$H_{dyn} = H_{net} - I\frac{dQ}{dt} \tag{2.1}$$

Where H_{dyn} is the redefined net head (m), H_{net} is the net head (m), I is the inertia of the water masses (s^2/m^2) and dQ/dt is the change in flow rate per time (m^3/s^2).

Nielsen's dynamic measurements was repeated by Stuksrud [9] on a low-head Francis turbine with two different draft tube designs. The results coincided with Nielsen's experiments; there was a significant deviation from the steady-state characteristics and this was most evident for the draft tube with the largest hydraulic inertia.

Olimstad [10] designed in 2010 a model pump-turbine runner for the Waterpower Laboratory at NTNU with purpose of investigating stability. Several methods for measuring the steady-state characteristics in the s-shaped area were tested [11]. These comprised of holding the turbine at a fixed rotational speed, throttling a valve downstream the turbine and using fixed torque as input. The results showed that the characteristics are pressure dependent for high non-dimensional speeds. The model runner was later used to investigate geometry impact by altering the leading edge profiles of the runner[12], concluding that small alterations can improve the stability without compromising the characteristics around the best efficiency point.

2.2 System dynamic analysis of hydropower plants

Within the field of system dynamic analysis in hydropower plants there have been numerous publications in the past decades. Both transient analysis in the time domain and stability analysis in the frequency domain have been covered in these publications, and they range from the purely theoretical to the more practical, related to actual power plants.

In 1984, Brekke [13] presented Norway's first doctoral thesis on system dynamics, focusing on stability calculations in the frequency domain. In the thesis, the theory of the Structural Matrix Method (SMM) for calculations of stability in hydropower plants with complex waterways, including the influence of the turbine characteristics, was presented. A new frequency dependent damping model was also introduced in this publication and verified by means of frequency response measurements on several large hydropower plants.

Pejovic et.al [14] published in 1987 a guideline for hydraulic transient analysis presenting important transient regimes to be aware of and how to perform a time domain analysis. There are several numerical methods for calculation of transients in a waterway, amongst these are the Method of Characteristics (MOC), presented by Wylie and Streeter [15]. The method is widely used for numerical solution of the equation of continuity and motion to determine changes in flow and head in a pipeline.

Modeling the waterway is important when performing transient calculations of hydropower plants, but it is also important to model the turbine correctly. Francis or reversible pump-turbines can be modeled by implementing measured steady-state characteristics from model tests into the calculation program. However, these are not always available. Nielsen presented in 1990 an analytical one-dimensional model for high head Francis turbines [6]. The purpose of the model was to better understand the physics of Francis turbines and to provide an option to measured model characteristics in transient calculations. The model has been further developed, and show good agreement with measured data in previous publications [6, 9, 16, 17, 18], and it was successfully used to better understand the physics of high head Francis turbines when it was presented by Nielsen in 1990. The model has in the later years also been developed further to be valid for reversible pump-turbines[19, 20].

2.3 Stability in reversible pump–turbines

The flow pattern in low specific speed pump–turbines operating in the s-shaped area has been investigated in several publications. High energy dissipation is the cause of the s-shaped characteristics and numerical calculations by both Staubli [5, 21] and Olimstad [22] show that local vortices starts to build up close to the leading edge of the runner blade, and at turbine brake mode the channel is almost completely blocked. The vortices cause an unsteady in- and outflow from the runner, thereby being the source of the instability. Hasmatuchi [23] showed with high speed visualization experiments that the flow at runaway is disturbed by a stall cell rotating in the vaneless space between the guide vanes and runner. The cell rotated at sub-synchronous speed and is likely a result of the backflow regions. The shape of the characteristics can to a certain degree be altered by changing the geometry of the runner. Yamabe [24] investigated by means of laboratory experiments the influence of the shape of the runner blades on the characteristics indicating that there is a relation between the blade angle and the characteristics. Olimstad [22] showed that giving the leading edges of the runner blades a more rounded off profile resulted in a positive contribution to the stability. The same is valid for decreasing the inlet diameter or nominal speed.

One of the early publications on dynamic behavior in pumped storage plants was presented by Pejovic [25] in 1976. He reported on unstable behavior at Bajina Basta Pumped Storage Power Plant. The plant consists of two reversible pump-turbines sharing the same penstock and tailrace. At the power house there is an asymmetric bifurcation to the two units. Pejovic concluded that the units influenced each other when entering the s-curved area and that asymmetry between the units caused phase shifting and increased the amplitudes of the pressure fluctuations significantly.

Martin [26, 27] showed that unstable behavior does not have to be related to an asymmetric system with two or more units. By performing a linear stability analysis on a system with short conduits and no friction he demonstrated that the stability is dependent on the slope of the unit torque-unit speed at runaway:

$$\frac{dT_{ed}}{dN_{ed}} < 0 \text{ Stable} \tag{2.2}$$

The unit properties are defined as:

$$T_{ed} = \frac{T}{\rho g H D_2^3} \tag{2.3}$$

$$N_{ed} = \frac{n D_2}{\sqrt{gH}} \tag{2.4}$$

Where T is torque (Nm), ρ is the density of water (kg/m^3), g is the gravitational acceleration (m/s^2), H is head (m), D_2 is the outlet diameter of the runner (m), Q is the flow rate (m^3/s) and n is the speed of rotation (rev/s).

The derivation of the criterion also indicates that the system can be stable with a positive gradient [22]:

$$\frac{dT_{ed}}{dN_{ed}} > \frac{n^0 2T_{ed}^* Q^*}{\tau_f n^* N_{ed}^0 \left|Q^0\right|} \text{ Stable} \tag{2.5}$$

Martin's analysis showed that the period of the inelastic oscillations, T_{osc}, was dependent on the time constant for the penstock and rotational masses, and the turbine characteristics:

$$T_{osc} = \pi\sqrt{\frac{2 \cdot T_a \cdot T_w}{b_1}} \tag{2.6}$$

Where:

T_w – time constant penstock, $[s]$

T_a – time constant rotational masses, $[s]$

b_1 – linear slope of the dimensionless torque-flow characteristics defined by:

$$\hat{t} = \frac{T}{n^2}\frac{(n^*)^2}{T^*} \tag{2.7}$$

$$v = \frac{Q}{n}\frac{n^*}{Q^*} \tag{2.8}$$

Where T is torque (Nm), n is rotational speed (rev/s), Q is flow rate (m^3/s) and superscript $*$ is best efficiency point.

Dörfler [28] reported that Bhira Pumped Storage Plant experienced sustained oscillations of speed, guide vane opening and head at speed no–load. The period of the low frequency oscillations was 15 seconds and confirmed the analytical prediction of the period of oscillation at runaway provided by Martin, see Eqn. (2.6). At 105% of rated speed additional medium frequency oscillations with a period of 3.75 seconds was observed. The low frequency oscillations made synchronization challenging. Pressure feedback was implemented in the governor algorithm and although this improved stability, it was of no help if the system was at or beyond the hydraulic stability limit. In order to render the system stable during synchronization, the start-up sequence was altered by throttling the main inlet valve. This introduced an artificial head loss pushing the stability limit to a higher value of n_{11}.

Medium frequency oscillations have later been investigated by Nicolet [29, 30]. He performed numerical calculations on a pumped storage plant equipped with 2x320 MW reversible pump–turbines experiencing load rejection where one unit close the guide vanes linearly and the other have guide vanes locked in open position and goes to runaway. The results showed oscillations with a period corresponding to the theoretical period predicted by Martin, see Eqn.(2.6). After approximately 250 seconds a shift over to elastic oscillations occurred, rendering a decrease in the period of the oscillations. The new period corresponded with the first natural frequency of the penstock, and the amplitude of torque and flow rate fluctuations increased with time. A parameter study showed that this shift, both existence and when it occurs, depended on the inertia of the rotating masses and the ratio of period of rigid to elastic water column oscillations. An eigenvalue and eigenmode analysis confirmed that the observed instability was induced by the positive slope of the pump–turbine at runaway, the analysis also showed that the elastic mode was more unstable than the rigid mode.

Prototype instability challenges have been reported numerous times and methods for avoiding the unstable behavior has been presented. Common for all methods is that they do not remove the source of the problem; they merely just avoid the operational area where the unstable behavior occurs. The method presented by Dörfler [28] at Bhira, throttling the main inlet valve during start-up is one option. Another method that has proved to work is Misaligned Guide Vanes (MGV), as described by Billdal and Wedmark [31], where a few guide vanes are operated independently. The MGV was successfully implemented to overcome synchronization challenges at low head at Tianhuangping Pumped-Storage Plant in China. A similar method, introduced by Klemm [32], solved the stability challenge at COO II.

Chapter 3

Research methods

The research methods applied in this work comprises of transient calculations, laboratory measurements and field measurements. An introduction to the different methods with their benefits and challenges are given in the following.

3.1 Transient calculations

One-dimensional numerical calculations are a time- and cost-effective method for analyzing the transient behavior of hydropower plants. The calculations are used both for dimensioning the waterway and to determine if the transient values are within the system requirements during the early stages of a project, as well as for further analysis of measured data for better understanding of physical phenomena.

Transient calculations are proven reliable, if executed correctly, but there are certain challenges to be aware of. There will always be a difference between real-life behavior and calculations as the transient phenomena are highly three-dimensional, hence with a one-dimensional analysis effects like draft tube vortex will not be included. Calculation of friction when dealing with transient flow is another challenge, as accurate friction models are typically in more than one dimension [33]. Simplified models are often used in one-dimensional calculations e.g. the Darcy-Weisbach-equation [34]. This model assumes a fully developed turbulent velocity profile, while in real life e.g. when the flow turns this assumption is not met. Another important parameter, frequency, is not taken into account in this model. This can cause an underestimation of friction loss as this loss increases with increasing friction.

A reliable calculation model is dependent on detailed and accurate input data, in addition to correct implementation. Large waterway systems contain numerous components in the waterway and it is often necessary to simplify parts of the system by making use of equivalents for certain elements. These simplifications must be made without removing elements with significant physical impact on the system transients.

Two programs have been used for transient calculations in this work; a custom made Matlab-program and in-house software from Rainpower. The calculation models have been verified with measurements. The Matlab-program has been used for calculations of the laboratory system, while the Rainpower program is used for prototype calculations. The latter was chosen based on the complexity of the prototype power plant, while the Matlab-program was developed to enable calculations with a one-dimensional turbine model.

The unsteady flow in the pipe elements are in both programs calculated with the Method of Characteristics (MOC), as described by Wylie and Streeter [15]. The method combine and transform the partial differential equations of continuity and motion, as presented in Eqn.(3.1) and (3.2), to finite differential equations enabling solutions from one time step (t) to the next (t+1).

$$\frac{\partial H}{\partial t} + \frac{a^2}{g}\frac{\partial v}{\partial x} = 0 \tag{3.1}$$

$$g\frac{\partial H}{\partial x} + \frac{\partial v}{\partial t} + f\frac{v\left|v\right|}{2D} = 0 \tag{3.2}$$

Where H is the head (m), t is time (s), a is the acoustic velocity (m/s), g is the gravitational acceleration (m/s^2),v is the velocity (m/s), f is the friction factor (-) and D is the pipe diameter (m).

The main difference between the two programs is the methods used for modeling the turbine. Measured model characteristics are implemented in the program from Rainpower, while the Matlab program uses a one-dimensional turbine model.

Measured model characteristics are a well-known method for turbine modeling in transient calculations. There will always be a deviation between the model and prototype characteristics, but with the true behavior of a prototype only available with field measurements, model characteristics are a good substitute, if available. Another option, as implemented in the Matlab-program, is to use an analytical turbine model.

The one-dimensional turbine model for high head Francis turbines consists of two equations, the hydraulic and the torque equation [6] :

$$T_{wt}\frac{dq}{dt} = h - \frac{q^2}{\kappa^2} - \sigma\left(\Omega^2 - 1\right) \tag{3.3}$$

$$T_a\frac{d\Omega}{dt} = q\left(\widetilde{m_s} - \psi\Omega\right)\left(1 - \frac{\Delta h}{h}\right) - R_m\Omega^2 - \eta_g \tag{3.4}$$

Description of the parameters are given in Tab.3.1 for further explanation of the model see Appendix A.

T_{wt}	Time constant of water masses in the turbine (s)
q	Dimensionless flow rate, $\frac{Q}{Q_n}$
h	Dimensionless head, $\frac{H}{H_n}$
κ	Guide vane opening degree (-)
σ	Machine constant, see Eqn.(A.18) in Appendix A
Ω	Dimensionless rotational speed, $\frac{\omega}{\omega_n}$
T_a	Time constant rotational masses (s)
$\widetilde{m_s}$	Dimensionless start torque, see Eqn.(A.10) in Appendix A
ψ	Machine constant, see Eqn.(A.12)
Δh	Dimensionless hydraulic loss, see Eqn.(A.13) and (A.15) in Appendix A
R_m	Mechanical loss factor (-)
η_g	Generator efficiency (-)

Table 3.1: Parameters turbine model

The model is implemented in the transient calculation model as a MOC boundary condition at the node located at the downstream end of the penstock. At time step Δt the values for Q and H at this node is found by solving the two differential equations representing the turbine.

The main benefit with this model is its ability to describe the physics of a Francis turbine with a minimum of input data. The required data can be obtained either from hydraulic designers in an early stage of a project, or by empirical data and calculations of initial design parameters as described by Brekke [35], amongst others. The model has been successfully implemented in the transient software LVTrans and has been verified with calculations and measurements on numerous power plants [36, 37]. A challenge with the model is, however, the implementation of losses. Loss factors for hydraulic and mechanical loss are required, and empirical data on the size of these factors are limited.

The turbine model was further developed by Nielsen [19, 20] to be valid for reversible pump-turbines by including a pumping effect in the hydraulic equation. This model was successfully implemented and verified with the transient calculations presented in Paper 1. During the work with this thesis, the model for pump–turbines has been further evolved by including the pumping effect in the torque equation, details are given in Appendix A.

The reversible pump-turbine model is now described with pumping effect, denoted as P, in both the hydraulic and torque equation:

$$T_{wt}\frac{dq}{dt} = h - \frac{q\,|q|}{\kappa^2} - \sigma\left|\Omega^2 - 1\right| - \overbrace{r_p\Omega^2 + r_p\Omega q}^{P} \tag{3.5}$$

$$T_a \frac{d\Omega}{dt} = |q| \left(\widetilde{m_s} - \psi\Omega + \overbrace{\gamma\Omega - r_p q}^{P} \right) \left(1 - \frac{\Delta h}{h} \right) - R_m \Omega^2 - \eta_g \tag{3.6}$$

The new parameters r_p and γ are defined in Eqn.(A.25) and (A.32), respectively. An explanation of the deduction of the model is given in Appendix A.

The validity of this model is discussed in Chapter 5.

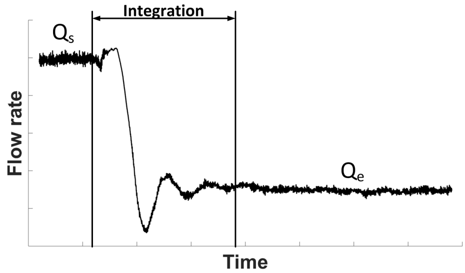
3.2 Laboratory measurements

Laboratory measurements are a common way of examining the qualities of a turbine design. The largest turbine manufacturers often perform model test to optimize the design and for verification of given guarantees. With these measurements the steady-state characteristics and efficiency curves of the runner are identified, cavitation properties are investigated, axial force at different loads is measured and so on. There are almost endless possibilities to what can be measured in a turbine test rig, but there are limitations to what can be scaled up to a prototype. The International Code for Model Acceptance Tests IEC 60193 [38] provides guidance on what and how to measure, and how to scale up the results to prototype.

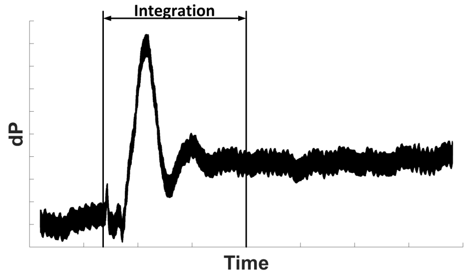
Model test are also often used for academic purposes and a few universities are equipped with a hydraulic laboratory. In addition to tests similar to the commercial tests, more elaborate tests and measurements with purpose of learning more on the physics of the machines are also often included. For these tests the up-scaling laws are not always applicable.

The laboratory measurements presented in this thesis is regarded as academic, see Paper 1 for a detailed description. The goal of the experiment was to obtain the dynamic characteristics of a model reversible pump–turbine installed in an open test loop to further explore the physics of these machines.

The instrumentation was as far as possible in accordance with the model test requirements in IEC 60193 [38], with the exception of the measured flow rate during the dynamic sequence. Due to the rapid changes in flow during the sequence, this parameter was measured with a method resembling the pressure-time method described in IEC 60041 [7]. The IEC method described is used for efficiency measurements on prototypes, where the flow is calculated based on the pressure rise between two measurement points during a deceleration of flow due to a rapid closure of the guide vanes. In the laboratory, flow measurements with a electromagnetic flow meter at steady-state before, Q_s, and after ,Q_e, the dynamic sequence is used as reference, see Fig. 3.1(a). This enables the flow during the sequence to be found by integration of the pressure difference between two sections upstream the turbine with respect to time, see Fig.3.1(b).



(a) Flow



(b) Pressure

Figure 3.1: Dynamic flow measurement

3.3 Field measurements

Numerical calculation models and laboratory measurements can, if performed correctly, give a good indication of the dynamic behavior or a system, but verification of the true behavior of a system can only be achieved through field measurements.

It is a challenge to obtain data from field measurements, both with regards to access and execution. Field measurements often require taking the unit out of normal operation and adding a certain amount of down-time to prepare for the measurements. Also, access to perform certain tests is challenging, especially as the most interesting cases to measure often are the most extreme. For the work presented here, the most interesting case would be to perform a trip-electrical-failure and blocking the servomotor to keep the guide vanes fully open allowing the turbine to go to runaway. Although the machine and hydraulic system is dimensioned to handle this scenario, it is often not included in a commissioning test program due to the amount of stress the unit, generator in particular, is exposed to.

Field measurements from a prototype reversible pump-turbine are presented in Paper 2 and 3, where they are used to investigate both the stability criterion for the system and the effect of the turbine characteristics on the transients. The measurements presented origins from the commissioning of the prototype where the above mentioned test was not part of the test program. However, as trip-electrical- and trip-mechanical-failure, see Table 3.2 for definitions, from full load was included, the measurements are highly relevant for the topic of this thesis. The tests were performed by the Rainpower commissioning team, where the author was a member and responsible for the instrumentation. The measurement setup was decided based on input from several disciplines; hydraulic, mechanical, system and governor. The execution of these tests were challenging both with regards to accessing the components of interest, selecting the most suitable sensors, signal noise and software.

From the list of parameters measured, the most relevant for this work are:

- Penstock pressure

- Spiral casing pressure
- Draft tube pressure
- Rotational speed
- Guide vane servo motor stroke

The test program was extensive as the suppliers of turbine, inlet valve, generator and control system each had separate tests that needed to be performed. The most significant turbine tests during the commissioning with regards to data presented in this thesis were:

- Trip-mechanical-failure 100% load
- Trip-electrical-failure 100% load

Trip-electrical-failure	The unit is running at a given load connected to the grid. The main circuit breaker opens and the unit accelerates towards runaway speed while the guide vanes are closing. The unit goes to stop.
Trip-mechanical-failure	The unit is running at a given load connected to the grid. A trip signal indicating a mechanical fault is given, and the guide vanes starts to close. When the guide vanes are almost closed the main circuit breaker opens.

Table 3.2: Definitions

Chapter 4

Summary of papers

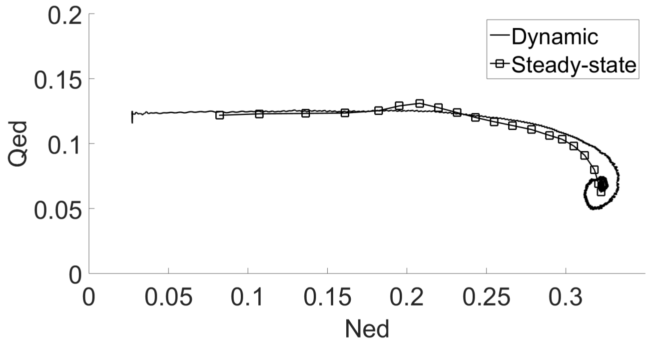
This chapter presents short summaries of the content in the three selected papers. Full length papers are found in Part II of this thesis.

4.1 Paper 1

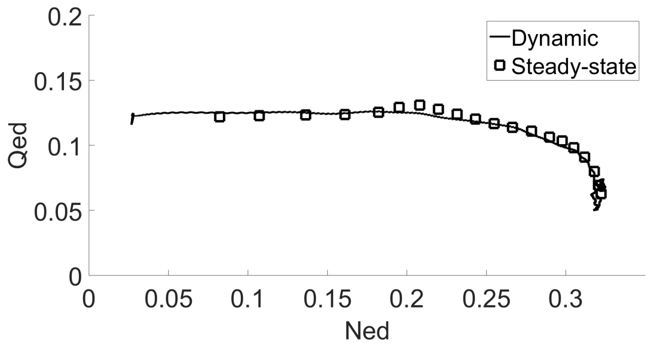
Measuring the Dynamic Characteristics of a Low Specific Speed Pump–Turbine Model
E.C. Walseth, T.K Nielsen and B. Svingen
Published in Energies 2016, 9, 199.

Paper 1 presents results from laboratory experiments performed to obtain the dynamic characteristics of a reversible pump–turbine model. The measurements were performed by initially operating the turbine at low rotational speed before disconnecting the generator allowing the turbine to go towards runaway. The open test loop system in the laboratory, with tanks located up- and downstream the turbine, ensured an approximate constant pressure. Pressure upstream and downstream the turbine, torque, rotational speed and flow was measured during the sequence. The latter was measured with a method resembling the pressure-time method. The results showed that the dynamic characteristics deviated from the measured steady-state characteristics after exceeding nominal speed, and it ended up with damped oscillations around runaway until steady-state operation was regained, see Fig.4.1(a) Hence, the turbine and system was stable and in accordance with the theoretical stability criterion from Martin [26].

A deviation between the dynamic and steady-state characteristics was detected and explained by the inertia of the water masses in the turbine, defined from the inlet of the spiral casing to the outlet of the draft tube. This inertia is taken care of by redefining the net head by subtracting the inertia multiplied with the change in flow per time, see Fig.4.1(b).



(a) Measured dynamic and steady-state characteristics



(b) Dynamic characteristics with redefined net head

Figure 4.1: Laboratory measurements

The results from the experiments were reproduced with transient calculations using an analytical one-dimensional turbine model for representation of the pump-turbine. The results showed good agreement with the measured data, proving that for transient analysis a simple one-dimensional model is a good alternative if measured model characteristics are not available. However, it should be used with caution as the stability of reversible pump-turbines are dependent on the unit torque-unit speed gradient at runaway and the calculations in this paper shows that the analytical model does not necessarily reproduce the correct steepness in this area.

4.2 Paper 2

Investigation of Stability Criteria for Reversible Pump–Turbines with Laboratory and Prototype Measurements
E.C. Walseth, T.K Nielsen and B. Svingen
Submitted

In Paper 2 an investigation of derived stability criteria is conducted. Measurements from the laboratory, consisting of steady-state and dynamic characteristics, as presented in Paper 1, is used together with prototype measurements for verification of the criteria. Stability for a system at constant speed is dependent on the gradient in the H-Q diagram; a negative gradient renders the possibility of an unstable system. The H-Q curves for the model pump–turbine in the laboratory, from here on referred to as the NTNU-runner, demonstrated that unstable behavior could occur at low flow rate. This was recognized during the measurements of the steady-state characteristics below runaway.

For a system with variable speed stability is dependent on gradient of the T_{ed}-N_{ed}– characteristics at runaway. The NTNU-runner has a negative gradient at runaway for all guide vane openings and the observed damped oscillations in the laboratory measurements are according to the predicted behavior, see Fig.4.1(a). The theoretical stability criterion for a positive gradient is fulfilled at guide vane opening degree k=1.6 for the prototype, where the opening degree, k, is defined as the normalized flow at $N_{ed} = 1$. In order to verify the true nature of the oscillations, measurements of a trip-electrical-failure while blocking the servo motor forcing the guide vanes to remain at opening k=1.6 should have been performed. This test would inflict the machine with a high level of stress and was not part of the test program. A transient calculation model, verified with prototype measurements, was used to investigate the nature of the oscillations for the described scenario. The results showed that the system was at the stability limit with periodic oscillations, see Fig.4.2, hence, the theoretical prediction did not correspond with the calculations.

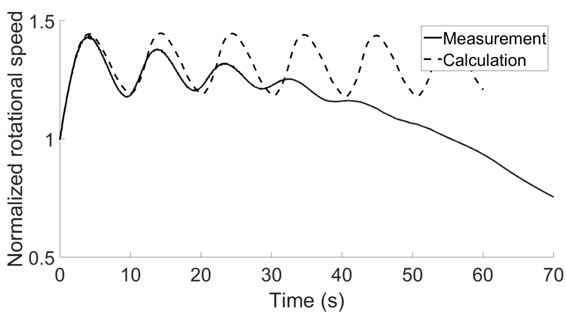


Figure 4.2: Measured speed at trip-electrical-failure and calculated speed at trip with servo motor blocked

Inclusion of friction in the stability criterion improves the margin towards stability. Calculations of a trip with the servomotor blocked with increased friction show that the amplitude of the oscillations decreases. However, the calculations also show that elastic effects become visible, demonstrated by the decrease in period of oscillations after approximately 60 seconds.

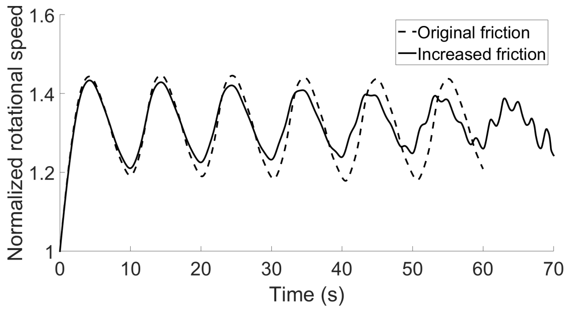


Figure 4.3: Calculation of speed at trip with servomotor blocked with original and increased friction loss

4.3 Paper 3

Prototype Study on the Effect of Reversible Pump–Turbine Characteristics on System Transients
E.C. Walseth, T.K Nielsen and B. Svingen
Submitted

The impact of turbine characteristics on system transients is studied in Paper 3. Depending on what operating point the turbine is at when a trip occurs, the steep part of the turbine characteristics have potentially a large influence on the system transients as a small increase in speed can give a large reduction in flow. This can result in a significant pressure increase upstream and decrease downstream the turbine.

Prototype measurements, as presented in Paper 2, are used for investigation of the characteristics impact on the transients. The prototype has a two-step closing law of the guide vanes, closing slowly during the first 30 seconds of the full load trip, before increasing speed during the last part of the sequence. This closing law is chosen based on the transient requirements of the system. An investigation of the guide vane closing scheme with transient calculations shows that an opposite closing law could be beneficial with regards to a trip on the prototype alone. Here the guide vanes close significantly during the first seconds of the trip, thereby avoiding entering the steep part of the characteristics at a large guide vane opening. However, as the pump-turbine shares penstock with several others, the worst case scenario is defined as a simultaneous trip-electrical-failure on all units, where one unit experience servomotor failure keeping the guide vanes locked in fully opened position. Due to this scenario the guide vanes had to close slowly in the beginning for all units to ensure that the transient requirements were met.

An indication of the effect of the turbine characteristics on the system transients is given by comparing measurements of a trip-electrical- and -mechanical failure from the same load, see Fig. 4.4. During the trip-mechanical-failure the unit remains at synchronous speed

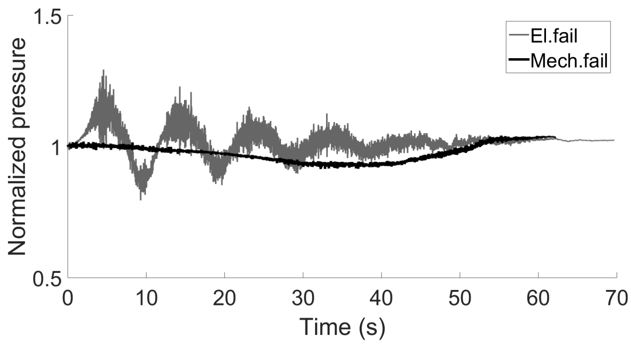


Figure 4.4: Measured pressure at trip-electrical- and -mechanical-failure

until the guide vanes are almost or completely closed, moving close to vertical when shown with Q_{ed}–N_{ed} properties and compared to the trip-electrical-failure, see Fig.4.5. Hence, the effect of the turbine characteristics on the transients are negligible. Measurements from the prototype show that the pressure increase during the trip-mechanical-failure is minimal, hence, the turbine is the major contributor to the pressure increase during the trip-electrical-failure.

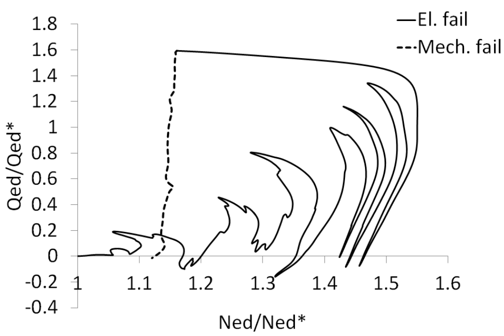


Figure 4.5: Calculated Q_{ed}-N_{ed} at trip-electrical- and -mechanical-failure

Chapter 5

General discussion

The research presented in the three selected papers focus on reversible pump-turbines in turbine mode of operation. Their influence on stability and transient behavior have been discussed and measurements from laboratory experiments and field tests on a prototype, combined with transient calculations, have been presented and compared to existing theory. The following presents a general discussion on the dynamic behavior of reversible pump-turbines, together with a discussion and verification of the improved one-dimensional reversible pump–turbine model, presented in Chapter 3.

5.1 Dynamic behavior of reversible pump–turbines

Derived stability criterion for hydraulic systems with reversible pump-turbines show that the gradient of the T_{ed}-N_{ed}–characteristics at runaway is decisive for the stability of the system. Laboratory and field measurements, as presented in Paper 1 and 2, show that a stable system is achieved with a negative gradient at runaway. A common approach to stability is to determine if the plant is stable during start-up in turbine mode. There are, however, other scenarios that should also be considered. A worst case scenario, that the plant must be dimensioned to withstand, is a trip from full load where a servo motor failure prevents the guide vanes from closing. The turbine will go towards runaway, and depending on the characteristics of the turbine, the system can experience damped, periodic or unstable oscillations.

Depending on the operating pattern of the plant, a quick re-connection to the grid following a load rejection can be a demand. In this case, the closing scheme of the guide vanes is an important parameter together with the turbine characteristics. The time frame for the re-synchronization is dependent on when the turbine approaches nominal speed after the disconnection from the grid. The moment the unit is disconnected from the grid it will start to accelerate and the guide vanes starts closing. The following speed rise depends on the characteristics and the closing time of the guide vanes. The quicker it closes, the smaller the speed increase will be, shortening the time it takes to re-connect to the grid. However, the guide vane closing scheme must be determined from the transient requirements of the plant, both with regards to pressure and speed, not for a quick re-connection to the grid.

Reversible pump-turbine characteristics can have a major influence on the system transients, as presented in Paper 3. Measurements of trip-electrical- and -mechanical-failure show that for the presented prototype, the majority of the pressure increase in the system is caused by the turbine. Transient analysis at an early stage of a project is important to ensure that the transient requirements are met, and for optimizing the guide vane closing scheme if necessary. The closing scheme is dependent on the turbine characteristics, the layout of the plant and the transient requirements, together with what is defined as the dimensioning worst case scenario. The latter can, as shown in Paper 3, make the difference between a quick and a slow closing in the beginning of a two-step closing scheme.

5.2 One-dimensional reversible pump–turbine model

Correct representation of reversible pump-turbine characteristics in a transient calculation model is important with regards to both stability and transients. This has motivated a development of the one-dimensional reversible pump–turbine model in this work, as presented in Chapter 3. The improved model, with added pumping effect in the torque equation, see Eqn.(3.6), show good correspondence with the measured dynamic characteristics, as presented in Fig.5.1. Hence, the inclusion of the pumping effect has not deteriorated the models ability to reproduce the physics in the measurements.

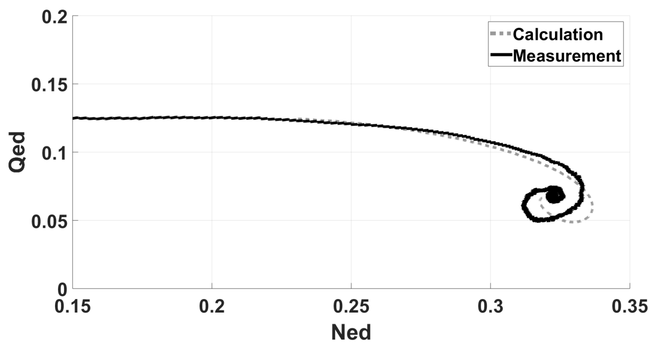
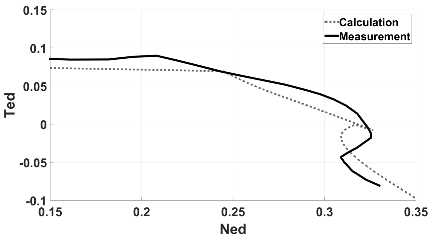
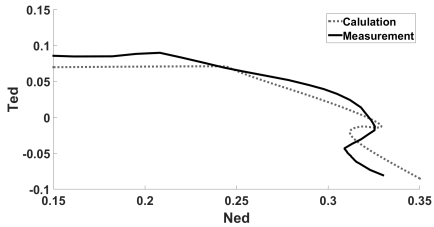


Figure 5.1: Dynamic $Q_{ed} - N_{ed}$ Characteristic

In Paper 1 the largest discrepancy between calculations and measurements was detected for the steady-state characteristics, specifically the T_{ed}-N_{ed}-characteristics. The added pumping effect in the torque equation gives a significant improvement with regards to the s-shape, as shown in Fig. 5.2 where a comparison of calculated T_{ed}-N_{ed}-characteristics with and without the added pumping effect in the model is presented.



(a) Model without pumping effect



(b) Model with pumping effect

Figure 5.2: Measured and calculated steady state $T_{ed}-N_{ed}$ characteristics with and without added pumping effect in torque equation

The improved model has introduced a clear s-shape in the T_{ed}-N_{ed}-characteristic, but there is still a significant discrepancy as the s-curve is too narrow. The added pumping effect has given a better inclusion of the rotational speeds influence on the flow. There are, however, effects significant to the shape of the curves that is not accounted for in the model. In the s-shaped area the flow through the runner is highly three-dimensional, and as stated in several publications [5, 22, 23, 39], the shape is caused by vortex formations and backflow regions in the turbine. For a one-dimensional model these effects are not possible to reproduce with high level of accuracy. Further analytical studies of the model might lead to improvements. However, a possible solution could be to find and include an empirical relation in the model based on measurements on several pump-turbines.

Chapter 6

General conclusions

The work in this thesis was divided into three objectives. The first comprised of an investigation and verification of the stability criteria for hydraulic systems with reversible pump-turbines by means of measurements and transient calculations.

Stability for a system at constant speed is dependent on the gradient in the H-Q-diagram. A negative gradient can render the system unstable. For the pump-turbine model in the laboratory, the gradient was negative at low flow rates and unstable behavior was detected during measurements of the steady-state characteristics below runaway.

For a system with variable speed stability is dependent on the gradient of the T_{ed}-N_{ed}–characteristics at runaway. The criterion consists of two inequalities; one for a negative gradient at runaway and one for a positive. Results from the investigation showed that the first part, stating stable behavior with a negative gradient, gave a correct prediction with regards to the laboratory measurements. The system experienced damped oscillations, before regaining steady-state behavior at runaway.

Damped oscillations were also predicted for the prototype with the criterion for a positive gradient. Results from calculations with a verified transient calculation model revealed periodic oscillations; hence, this part of the stability criterion could not be verified.

The effect of pump-turbine characteristics on the transients in the system was investigated in the second objective. The results show that the characteristics have a major effect on the transients, and that the guide vane closing law is an important parameter to ensure that the transient requirements of the plant are fulfilled.

The third objective was to improve a one-dimensional reversible pump-turbine model used in transient calculations. The existing model was verified with laboratory measurements revealing a potential for improvement with regards to the steady-state T_{ed}-N_{ed}-characteristics. The presented improvement of the model, with an added pumping effect in the torque equation, shows a clear s-shape and better correspondence with measured steady-state characteristics.

Chapter 7

Future work

The aim of this thesis has been to increase our understanding on the dynamic behavior of reversible pump-turbines in turbine mode of operation. During the work with this thesis, some suggestions for future work have been identified.

The measurements of the dynamic characteristics in the laboratory gave valuable information on the behavior of the model pump-turbine. It also made it possible to verify the theoretically predicted behavior by the stability criterion. Performing similar measurements on a prototype would be beneficial, especially on a prototype that fulfill the criterion for stable behavior with a positive gradient. Results from this type of measurement would reveal the true behavior of the system.

The improved one-dimensional reversible pump-turbine model has shown good agreement with measured characteristics. It does, however, have potential for further improvement as the s-shape of the steady-state characteristics is still not reproduced with sufficient accuracy. Part of this deviation is due to the lack of representation of three-dimensional effects, which is not possible to fully represent with a one-dimensional model. A further analytical study is suggested, but also an investigation of empirical relations by means of measurements should be performed. An inclusion of results from these measurements in the model can potentially solve the challenge of correct calculation of the s-shape.

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Appendix A

The One-Dimensional Turbine Model

The one-dimensional model for a reversible pump–turbine is developed from the original model for high head Francis turbine [6]. In the following the original Francis turbine model and the reversible pump-turbine model by Nielsen is presented together with a suggested further development of the latter. The last section contains a list of symbols.

A.1 The High Head Francis Turbine Model

The high head Francis turbine model consists of two equations, hydraulic and torque. In the following a presentation of the two equations and their parameters based on Nielsen [6] is given.

A.1.1 Hydraulic equation

The hydraulic equation is here presented on dimensionless form:

$$T_{wt}\frac{dq}{dt} = h - \frac{q^2}{\kappa^2} - \sigma\left(\Omega^2 - 1\right) \tag{A.1}$$

The dimensionless values in the equation are reduced on the nominal values:

$$\text{Flow } q = \frac{Q}{Q_n} \tag{A.2}$$

$$\text{Head } h = \frac{H}{H_n} \tag{A.3}$$

$$\text{Speed } \Omega = \frac{\omega}{\omega_n} \tag{A.4}$$

In the hydraulic equation, Eqn.(A.1), the term T_{wt} is introduced. This is the inertia of the water masses inside the turbine, defined from the inlet of the spiral casing to the outlet of the draft tube:

$$T_{wt} = I\frac{Q_n}{H_n} \tag{A.5}$$

The opening degree of the turbine, κ:

$$\kappa = \frac{\sin\alpha_1}{\sin\alpha_{1n}} \tag{A.6}$$

The geometry of Francis turbines gives a throttling effect that is dependent on the speed of rotation. This effect is included in the hydraulic equation with the parameter, σ:

$$\sigma = \frac{1}{2}r_1^2\left(1 - \frac{r_2^2}{r_1^2}\right)\frac{\omega_n}{gH_n} \tag{A.7}$$

A.1.2 Torque equation

The dimensionless torque equation:

$$T_a\frac{d\Omega}{dt} = q\left(\widetilde{m_s} - \psi\Omega\right)\left(1 - \frac{\Delta h}{h}\right) - R_m\Omega^2 - \eta_g \tag{A.8}$$

The time constant of the rotational masses:

$$T_a = \frac{J2\pi n^*}{T^*} \tag{A.9}$$

$\widetilde{m_s}$ is defined as the starting torque, i.e at $\Omega = 0$:

$$\widetilde{m_s} = \xi\frac{q}{\kappa}\left(\cos\alpha_1 + \tan\alpha_{1n}\sin\alpha_1\right) \tag{A.10}$$

Where:

$$\xi = \frac{u_{1n}c_{1n}}{gH_n} \tag{A.11}$$

The machine constant, ψ:

$$\psi = \frac{r_2^2\omega_n^2}{gH_n} \tag{A.12}$$

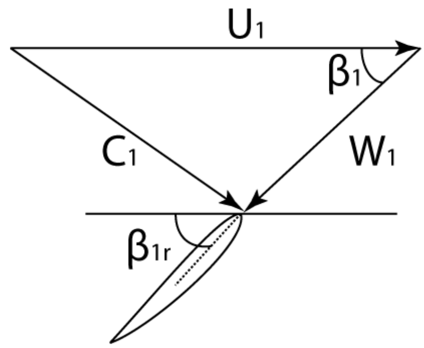

Figure A.1: Flow inlet angle and inlet blade angle

The torque equation also includes both mechanical and hydraulic loss. The mechanical loss, $R_m\Omega^2$, consist of loss in bearings and disc friction loss and is dependent on the rotational speed, Ω, and a loss factor, R_m.

The hydraulic losses are defined as:

$$\Delta H = \overbrace{R_1 Q^2}^{Viscous\ loss} + \overbrace{R_2 \left(Q - Q_c\right)^2}^{Incident\ and\ draft\ tube\ loss} \tag{A.13}$$

Where R_1 and R_2 are loss factors, and Q_c is the flow rate when the incident angle is zero; when the flow inlet angle, β_1, is equal to the inlet angle of the blade, β_{1n}, see Fig. A.1:

$$Q_c = \frac{A_1 \omega r_1 \tan\beta_{1n}}{1 + \tan\beta_{1n}\cot\alpha_1} \tag{A.14}$$

The hydraulic losses are presented on dimensionless form in Eqn.(A.8)

$$\text{Hydraulic losses } \Delta h = \frac{\Delta H}{H_n} \tag{A.15}$$

Efficiency

The hydraulic efficiency is defined as:

$$\eta = \frac{T\omega}{\rho g Q H} \tag{A.16}$$

Introducing dimensionless terms:

$$\eta = \frac{1}{gH}\left(\widetilde{m_s} - r_2^2\omega\right)\omega \tag{A.17}$$

Assuming that best efficiency point occurs when $q = 1$, $\Omega = 1$, $\kappa = 1$ and $h = 1$, and that the derivative $d\eta/d\Omega = 0$, the following relationships between σ, ψ, ξ and best hydraulic efficiency, η_{max}, emerges:

$$\sigma = \frac{\eta_{max} - \psi}{\eta_{max} + \psi} \tag{A.18}$$

$$\xi = (\eta_{max} + \psi)\cos\alpha_{1r} \tag{A.19}$$

A.2 The Reversible Pump–Turbine Model

A.2.1 Hydraulic equation

The model described above is valid for a high head Francis turbine, however, for a reversible pump-turbine the pumping effect must be taken into consideration [19]. This effect is a consequence of the prolonged blades of a pump–turbine intensifying the throttling in the turbine as the speed increases.

Nielsen[19] showed that the Euler equation for a pump using turbine notation is:

$$gH_p = u_1\left(u_1 - \frac{Q}{A_1\tan\beta_1}\right) - u_2\left(u_2 - \frac{Q}{A_2\tan\beta_2}\right) \tag{A.20}$$

Defining:

$$R_p = \frac{r_1}{A_1\tan\beta_1} - \frac{r_2}{A_2\tan\beta_2} \tag{A.21}$$

$$s_p = r_1^2\left(1 - \left(\frac{r_2}{r_1}\right)^2\right) \tag{A.22}$$

Combining Eqn.(A.21)-(A.22) and the relation $u = \omega r$ with Eqn.(A.20) yields the pumping head:

$$gH_p = s_p\omega^2 - \omega R_p Q \tag{A.23}$$

Introducing dimensionless terms and implementing the pumping head in the hydraulic equation, Eqn.(A.1):

$$T_{wt}\frac{dq}{dt} = h - \frac{q\,|q|}{\kappa^2} - \sigma\left(\Omega^2 - 1\right) - \sigma_p\Omega^2 + r_p\Omega q \tag{A.24}$$

$$r_p = \frac{R_p \omega_n Q_n}{g H_n} \tag{A.25}$$

Assuming steady-state operation, $\frac{dq}{dt} = 0$, at nominal condition, $q = 1$, $\Omega = 1$, $\kappa = 1$ and $h = 1$, the following relation is found:

$$\sigma_p = r_p \tag{A.26}$$

A.2.2 Torque equation

In the development from a Francis to a reversible pump-turbine model, Nielsen only changed the hydraulic equation. In the following a suggested change in the torque equation is presented. A discussion and verification of this equation is presented in Chapter 5.

The torque, M, is defined by the Euler equation:

$$M = \rho Q \left(r_1 c_{1x} - r_2 c_{2x} \right) \tag{A.27}$$

Combining this equation with the velocity triangles, see Fig.A.2, and the relation $u = \omega r$ gives an expression for the pump torque, M_p:

$$M_p = \rho Q \left\{ r_1 \left(\omega r_1 - \frac{Q}{A_1 \tan \beta_1} \right) - r_2 \left(\omega r_2 - \frac{Q}{A_2 \tan \beta_2} \right) \right\} \tag{A.28}$$

Introducing dimensionless values and defining the constant γ in Eqn.(A.28) yields an expression for the reduced pump torque, t_p:

$$t_p = \gamma \Omega - r_p q \tag{A.29}$$

$$\gamma = \frac{\omega_n^2 \left(r_1^2 - r_2^2 \right)}{g H_n} \tag{A.30}$$

The pump torque is added to the torque equation, Eqn.(A.8), for inclusion of the increased pumping effect in a reversible pump-turbine compared to a Francis turbine:

$$T_a \frac{d\Omega}{dt} = q \left(\widetilde{m_s} - \psi \Omega + \gamma \Omega - r_p q \right) \left(1 - \frac{\Delta h}{h} \right) - R_m \Omega^2 - \eta_g \tag{A.31}$$

Demanding $\eta = 1$ at zero loss with $q, \omega, h = 1$:

$$\gamma = \left(1 - \widetilde{m_s} + \psi + r_p \right) \tag{A.32}$$

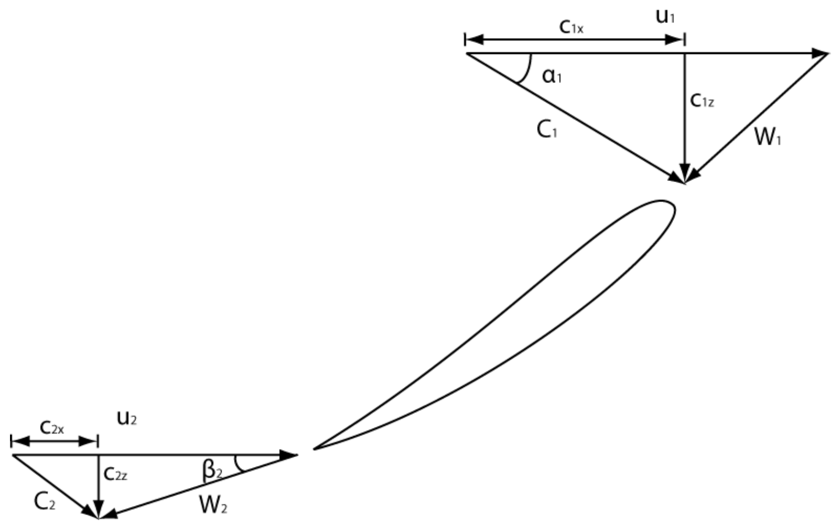

Figure A.2: Velocity triangles

A.3 List of Symbols

A	Area (m^2)
c	Absolute velocity (m/s)
g	Gravitational constant (m/s^2)
H	Head (m)
h	Reduced head, $\frac{H}{H_n}$ (-)
I	Hydraulic inertia (m^2/s^2)
J	Polar moment of inertia (kgm^2)
M	Torque (Nm)
n	Rotational speed (rev/s)
Q	Flow rate (m^3/s)
q	Reduced flow rate, $\frac{Q}{Q_n}$ (-)
R_1	Draft tube and incident loss coefficient (-)
R_2	Viscous loss coefficient (-)
R_m	Mechanical loss coefficient (-)
r	Radius (m)
T_a	Time constant rotating masses (s)
T	Period of oscillations/Torque (s)/(Nm)
t	Time, (s)
T_{wt}	Time constant of water masses in the turbine (s)
u	Peripheral speed (m/s)
α	Guide vane angle (-)
β	Runner blade angle (-)
η	Turbine efficiency (-)
η_g	Efficiency generator (-)
κ	Guide vane opening degree (-)
ρ	Density of water (kg/m^3)
Ω	Reduced rotational speed, $\frac{\omega}{\omega_n}$ (-)
ω	Rotational speed (rad/s)

Sub- and superscript

max	Maximum value
n	Nominal value
p	Pump
1	Runner inlet
2	Runner outlet
*	Best efficiency

Part II – Selected papers

Paper 1

Measuring the Dynamic Characteristics of a Low Specific Speed Pump–Turbine Model
E.C. Walseth, T.K. Nielsen and B. Svingen
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Article

Measuring the Dynamic Characteristics of a Low Specific Speed Pump—Turbine Model

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Abstract: This paper presents results from an experiment performed to obtain the dynamic characteristics of a reversible pump-turbine model. The characteristics were measured in an open loop system where the turbine initially was run on low rotational speed before the generator was disconnected allowing the turbine to go towards runaway. The measurements show that the turbine experience damped oscillations in pressure, speed and flow rate around runaway corresponding with presented stability criterion in published literature. Results from the experiment is reproduced by means of transient simulations. A one dimensional analytical turbine model for representation of the pump-turbine is used in the calculations. The simulations show that it is possible to reproduce the physics in the measurement by using a simple analytical model for the pump-turbine as long as the inertia of the water masses in the turbine are modeled correctly.

Keywords: reversible pump-turbine; stability; transient calculations

1. Introduction

For new or upgraded high head power plants, transient analysis is often an important part of the scope. This analysis gives information regarding maximum pressure in the penstock, minimum pressure in the draft tube and maximum overspeed during load rejection. Also, the system's response to ramping of load, transient response of the governor and stability can be investigated with appropriate software.

Today's software covers most of the needs regarding analysis of the dynamic system, but lack of information regarding the turbine in an early stage of a project will necessitate an analytical approach with respect to turbine modeling.

Nielsen [1,2] presented a simple analytical model for high head Francis turbines in 1991 and an improved version suitable for reversible pump-turbines in 2010. The model requires a minimum of input data and has proven good accuracy with measured data.

The analytical model has a one-dimensional approach making it challenging to model the highly three dimensional flow through a reversible pump-turbine. These machines have steep flow-speed and torque-speed characteristics, commonly known as S-shaped curves. Vortex formations and backflow regions in the turbine are shown both numerically and experimentally to be the cause of these curves [3–6], and when combining these characteristics with a hydraulic system unstable behavior can occur [7].

Pump-turbine instabilities often occur during start-up in turbine mode and possible off-design operation and several solutions have been proposed to avoid operational problems. Klemm [8] suggested moving two guide vanes separately at COO II, while Dörfler [9] introduced a solution by throttling the main inlet valve to introduce an artificial head loss at Bhira. Both solutions avoids operation in the area of hydraulic instability without removing the cause of the problem.

Martin [10,11] derived a stability criterion for reversible pump-turbines at runaway by using linearized stability analysis on a system with short conduits and no friction. He showed that the stability was dependent on the slope of the unit torque-unit speed curve and that there was a correlation between the ratio of time constant for the water masses in the penstock and the time constant of the rotating masses. Depending on these parameters the system can experience unstable, limit cycles, neutrally stable or periodic damped oscillations. Inelastic calculations of load rejection with the guide vanes remaining fully opened showed that with a positive slope of the unit torque-unit speed at runaway the machine experienced limit cycle or unstable oscillations at runaway. Machines with negative slope of the unit torque-unit speed and unit flow-unit speed at runaway, but still with S-shaped characteristics, showed damped oscillations around runaway.

The dynamic behavior of reversible pump turbines is still not fully understood even though it has been the subject of several publications. This paper contributes to the discussion with a representation of the transient behavior in reversible pump turbines based on laboratory measurements of the dynamic characteristics, which are reproduced in numerical transient analysis utilizing Nielsen's turbine model. This comparison demonstrates a good correlation between the results from the experiments and the results from the transient simulations. The measurements were performed by initially running a model pump turbine on low rotational speed. The generator was then disconnected allowing the turbine to go towards runaway while guide vane opening and head remained constant. During this sequence the flow rate, torque, pressure and rotational speed was measured. Results from the experiment are presented and discussed.

2. Experimental Setup

2.1. Test Rig

The Francis test rig at the The Waterpower Laboratory at the Norwegian University of Science and Technology (NTNU) has the possiblity to be run in a so called open loop configuration. The laboratory is equipped with two large tanks connected by a weir located upstream the turbine, see Figure 1. A draft tube tank with an overflow system is installed downstram the turbin. These installations work as an upper and lower reservoir, respectively, with an approximate constant level.

The experiments described in this paper are performed in the open loop system together with a reversible pump-turbine runner installed in an existing Francis turbine rig. The runner is designed by Olimstad [12] for NTNU with the purpose of investigating stability of pump-turbines. Table 1 describes the properties of the model turbine.

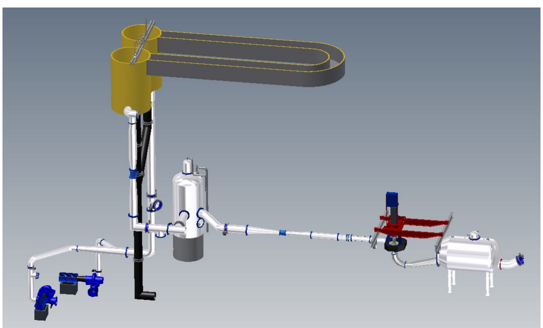

Figure 1. Open loop test rig.

Table 1. Turbine properties.

Param.	Value	Param.	Value	Param.	Value
D_1	0.631 m	N_{ED}^*	0.133	β_1	12°
D_2	0.349 m	Q_{ED}^*	0.223	β_2	12.8°
B_1	0.059 m	H^*	29.3 m	α^*	10°

2.2. Instrumentation

Pressure at the turbine inlet and outlet, rotational speed, torque and flow rate was measured during the experiment. For measuring the pressure at the turbine inlet and outlet, a Fuji Electronics FHCW 36 Ackay transducer was used. The rotational speed was measured with an optical sensor, Jaquet DSR 18200, and the pulse from this instrument was converted to a volt signal with a Jaquet FT 1400 D. A free-floating generator is mounted on the turbine shaft and a level arm connected to a force cell prevents the generator from rotating. The torque is obtained by multiplying the measured force with the length of the arm. The flow rate during steady-state operation was measured with an electromagnetic flow meter of the type Alto SC 100 AS from Krohne.

During the dynamic sequence the change in flow rate was obtained with the pressure-time method. Two absolute pressure transducers, PTX 1400, with a range 0–2.5 bar was mounted upstream the turbine with 3.94 m distance. The change in flow rate at each time step is obtained by integrating the pressure difference, Δp, between these two transducers with respect to time:

$$\Delta Q = \frac{A}{\rho L} \int_{t_1}^{t_2} (\Delta p + \zeta)\, dt \tag{1}$$

$$\zeta = kQ^2 \tag{2}$$

ΔQ —change in flow rate, $[m^3/s]$
A —cross-sectional area of pipe, $[m^2]$
L —length of pipe, $[m]$
ρ —density of water, $[kg/m^3]$
dt —time step, $[s]$
ζ —steady-state friction loss, $[Pa]$
k —friction loss coefficient, $[\frac{kg}{m^7}]$

A data acquisition system (NI cDAQ-9172) from National Instruments was used during the measurements. The sample frequency was set to 1613 Hz and the sampling was performed without filtering.

2.3. Test Program

Tests was performed at 4°, 7°, 10° and 13° guide vane opening. Each test were performed by initially running the turbine on low rotational speed. The generator was then disconnected allowing the turbine to go towards runaway speed with the guide vane opening and head remaining constant.

3. Data Processing

3.1. Filtering of Data

A significant amount of noise was registered in the pressure measurements. This is not of relevance for the measurements used for the pressure-time method; however, for presenting the dynamic characteristics with unit properties reduced by net head, the data for pressure at the turbine inlet and outlet had to be filtered. To avoid any issue regarding phase distortion all measurements were run through a filter with the same parameters. A fifth order Butterworth filter with a cut off frequency at 50 Hz was used. FFT-analysis showed that a cut off at 50 Hz would not remove any of the dominating frequencies in the pressure-time measurements as these were located at approximately 0.25 Hz.

3.2. Flow Rate

Deviations from IEC 60041 [13] for pressure-time measurements was necessary due to constraints in the laboratory, however, as the objective of this experiment was to obtain the shape of the dynamic characteristics the uncertainty of the curves are of secondary importance. The IEC 60041 states that for pressure-time measurements the pressure sensors must be placed at least 10 m apart and the cross sectional area should be constant between the measurement sections. Due to geometrical constraints in the Francis test rig the sensors had to be placed 3.94 m apart and the cross-sectional area in between was not constant. The upstream sensor was mounted at a pipe diameter of 400 mm and was placed directly after a conical pipe. The downstream sensor was placed on a pipe with 350 mm diameter; hence the flow conditions were not optimal. Thus, the value for the effective area, A, in Equation (1) had to be determined by iteration of the measured data and, due to the flow conditions, this value varied for each measurement. The relative change in flow rate was found by integrating the differential pressure between the two sensors upstream the turbine according to Equation (1). The friction before and after the dynamic sequence together with the steady-state flow rate were known values. Iteration together with the value for the cross sectional area between the pressure measurement was used to determine the friction during the dynamic sequence.

4. Results

4.1. Dynamic Characteristic

Measured dynamic and steady-state characteristics with reduced values for 10° guide vane opening are shown in Figure 2. The values on the y- and x-axis are reduced flow, Q_{ed}, and rotational speed, N_{ed}, respectively. These parameters are defined by Equations (3) and (4).

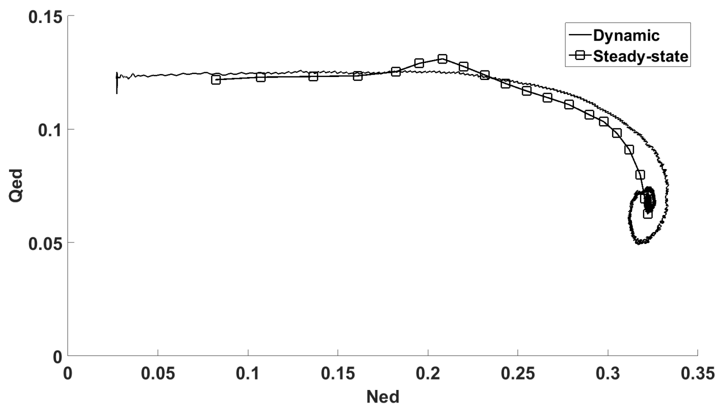

Figure 2. Q_{ed}-N_{ed} characteristics at 10° guide vane opening.

$$Q_{ed} = \frac{Q}{D_2^2\sqrt{gH}} \tag{3}$$

$$N_{ed} = \frac{nD_2^2}{\sqrt{gH}} \tag{4}$$

where D_2 is outlet diameter of the runner, (m), H is net head, (m), and g is the gravitational constant, (m/s^2). The change in flow rate over time is shown in Figure 3 for 10° guide vane opening.

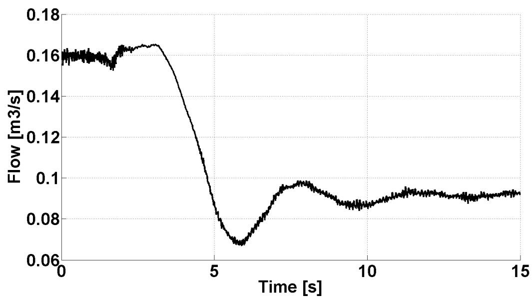

Figure 3. Flow rate during dynamic sequence at 10° guide vane opening.

4.2. Uncertainty

The measured characteristics have a calculated relative uncertainty of 2.61% for the Q_{ed}-value and 0.21% for the N_{ed}-value [14]. The main contributors to the uncertainty in Q_{ed} is the iteration of the area between the measurement points for the pressure-time measurement and the uncertainty of the electromagnetic flow meter. The loss is modeled as for steady-state flow, common for this type of measurement, however, it is known to be one of the causes of inaccuracy when using pressure-time method for flow measurements [15]. The repeatability of the measurement is good, shown in Figure 4 where three series at 10° guide vane opening are plotted. The objective with the experiment was to obtain the shape of the dynamic characteristics and the repeatability indicates that this was achieved with success.

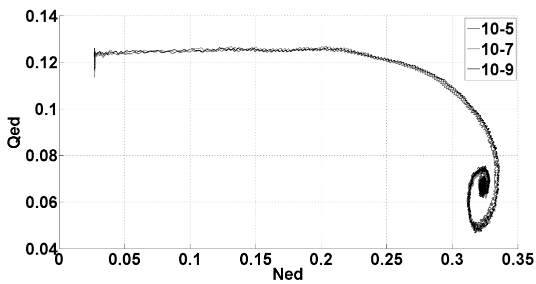

Figure 4. Repeatability at 10° guide vane opening.

4.3. Discussion

The results from the experiments show that the pressure-time method is well-suited for measuring the relative change in flow during a dynamic sequence. Constraints in the Francis test rig forced deviations from the IEC 60041 standard resulting in a significant uncertainty in the flow rate measurements. However, the repeatability in the measurements showed that the shape of the curves is consistent, thereby fulfilling the main objective of the experiment; obtaining the shape of the curves.

The dynamic characteristics in Figure 2 show that the combination of machine and hydraulic system is stable. The oscillations, with a period of approximately T = 4 s, are damped and kept within the turbine quadrant. The system regains steady-state operation at runaway after a short period of time. This result corresponds with Martins stability criterion [10,11] at runaway, see Figures 5 and 6:

$$\frac{dT_{ed}}{dN_{ed}} < 0 \Longrightarrow Stable \tag{5}$$

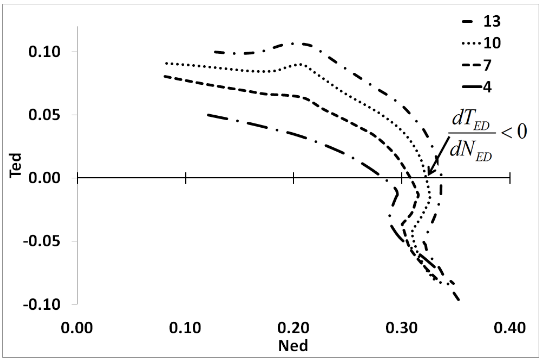

Figure 5. Steady state T_{ed}-N_{ed} characteristics.

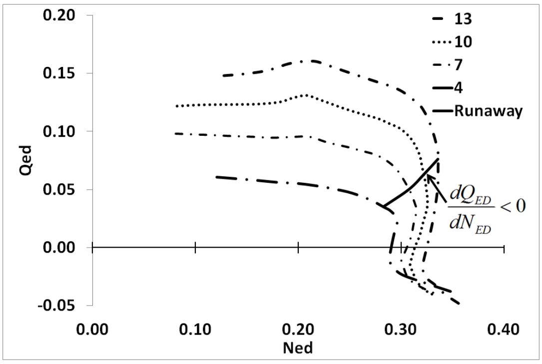

Figure 6. Steady state Q_{ed}-N_{ed} characteristics.

Figure 2 show a significant deviation from the measured steady-state characteristic at values higher than N_{ed} = 0.25. Nielsen performed an similar experiment on a high head Francis turbine in 1990 with corresponding results [1]; the characteristic starts to deviate from the steady-state at high values of N_{ed} and ends up with damped oscillations around the runaway point.

A proposed explanation is that the inertia of the water masses inside the turbine, defined as the inlet of the spiral casing to the outlet of the draft tube, is causing the deviation. The effect of the inertia, I, can be removed by redefining the net head, H_{net}, used for calculating the unit properties in Equations (3) and (4):

$$H_{dyn} = H_{net} - I\frac{dQ}{dt} \tag{6}$$

For the runner the inertia is found by integration along a streamline, as shown in Figure 7 and Equation (7). The same principle can be applied for the spiral casing and draft tube, but for the these the well-known $I = \frac{L}{gA}$ can be used with sufficient accuracy.

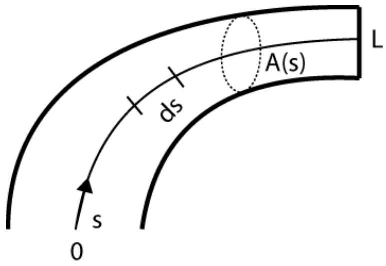

Figure 7. Integration of streamline.

$$I = \frac{1}{g} \int_0^L \frac{A(s)}{ds} \quad (7)$$

Calculating the inertia with Equation (7) is a complex task requiring complete knowledge of the geometry. For this case, measured data is available, hence the value for the inertia is found by iteration to be approximately $I = 20\ s^2/m^2$. Figure 8 show the dynamic characteristic with redefined net head together with the measured steady-state characteristics. The majority of the deviation is removed and the turbine follows a steady-state line up and down until the system has regained steady-state operation.

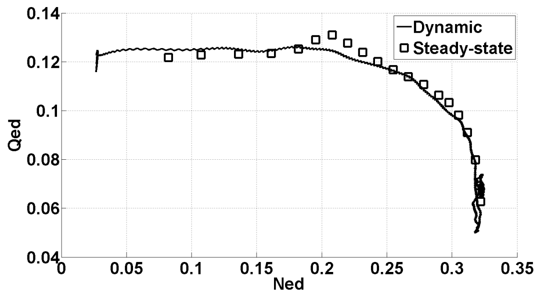

Figure 8. Dynamic characteristics with redefined net head.

5. Transient Simulations

Transient calculations were performed by using the the method of characteristics [16] to model the pipe elements and Nielsen's turbine model [1] for the representation of the pump-turbine. The turbine model has previously shown good accuracy compared with measured steady-state characteristics [2]; in addition it also takes into consideration the inertia of the water masses inside the turbine. It is cruicial to model the turbine correct in order to reproduce the physics of the measurements, hence the inclusion of the inertia is important, as shown by Walseth *et al.* [17].

The analytical model is derived from the Euler equation and consists of two equations; the hydraulic and torque equation, here shown on dimensionless form with the parameters reduced on the nominal values:

$$T_{wt} \frac{dq}{dt} = h - \frac{q\,|q|}{\kappa^2} - \sigma \left| \Omega^2 - 1 \right| - \sigma \Omega^2 + R_q \Omega q \quad (8)$$

$$T_a \frac{d\Omega}{dt} = |q| \left(\widetilde{m_s} - \psi_n \Omega \right) \left(1 - \frac{\Delta h}{h} \right) - R_m \Omega^2 - \eta_g \quad (9)$$

$$T_{wt} = I \frac{Q_n}{H_n} \quad (10)$$

$$\kappa = \frac{\sin\alpha_1}{\sin\alpha_{1opt}} \tag{11}$$

$$R_q = \frac{\omega_n Q_n}{gH_n}\left(\frac{r_1}{A_1\tan\beta_{1r}} - \frac{r_2}{A_2\tan\beta_{2r}}\right) \tag{12}$$

$$\widetilde{m_s} = \xi\frac{q}{\kappa}\left(\cos\alpha_1 + \tan\alpha_{1opt}\sin\alpha_1\right) \tag{13}$$

$$\Delta h = R_f q^2 + R_d\left(q - q_c\right)^2 \tag{14}$$

$$q_c = \Omega\left(\frac{1+\cot\alpha_{1opt}\tan\beta_{1r}}{1+\cot\alpha_1\tan\beta_{1r}}\right) \tag{15}$$

$$\psi_n = \frac{u_{2n}^2}{gH_n} \tag{16}$$

$$\sigma = \frac{\eta_n - \psi_n}{\eta_n + \psi_n} \tag{17}$$

$$\xi = \left(\eta_n + \psi_n\right)\cos\alpha_{1opt} \tag{18}$$

The inputs to the turbine model are known design parameters, except for the loss factors, R_m, R_f and R_d, that are tuned to obtain best possible fit. The value for the acceleration time for the rotating masses, T_a, and time constant for the water masses in the turbine, T_{wt}, was in this case also adjusted to improve the accuracy between simulations and measurements. With a low value of T_{wt} the turbine will follow a steady-state line and by increasing this value the characteristics goes further to the right compared to the steady-state measurements and the oscillations around runaway becomes more visible. Hence, giving a better fit with the measurements in addition to increasing the period of the oscillations in rotational speed and flow rate. The value for T_a was adjusted to obtain the correct gradient for increase in speed with respect to time, $\frac{dn}{dt}$.

Transient simulations of the dynamic sequence compared with measured data at 10° guide vane opening is shown in Figures 9–11. The simulations are started at nominal speed due to an underestimation of flow rate in the turbine model at lower speed making it difficult to tune the simulations according to the measurements.

The results show that the physics in the dynamic sequence is reproduced with good accuracy. Calculations show that the turbine experience damped oscillations around runaway before regaining steady-state operation, as shown in the measurements, see Figure 9 where the timescale for the plot is 128 s.

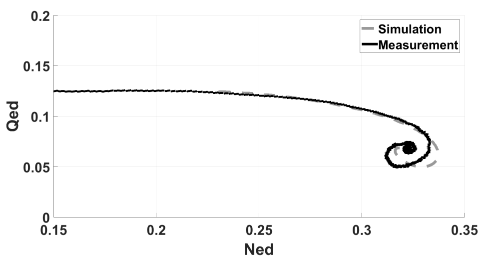

Figure 9. Q_{ed}-N_{ed} Characteristic.

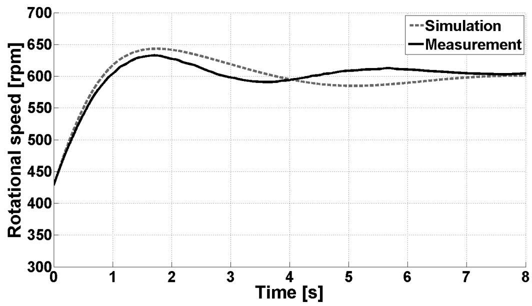

Figure 10. Rotational speed.

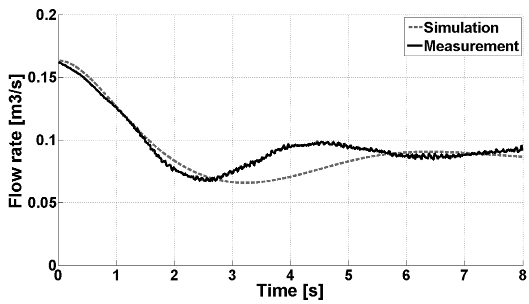

Figure 11. Flow rate.

The dampening of the oscillations in the calculations is larger than measured, see Figures 10 and 11. This can be explained by looking at the modeling of the turbine. Figures 12 and 13 show the measured steady-state characteristics, unit torque-unit speed and unit flow-unit speed, respectively, together with the simulated steady-state characteristics from the turbine model. From Figure 12 it is clear that the slope of the unit torque-unit speed at runaway $(T = 0)$ is less steep than measured, and the unit flow-unit speed slope at runaway (marked with stars) have a lower absolute value compared to the measured data. This leads to highly damped oscillations, as also reported by Martin [11]. The oscillations can be provoked by changing the time constant for the rotational masses to a lower value; however, this will alter the steepness of the $\frac{dn}{dt}$-gradient and increase the maximum transient speed.

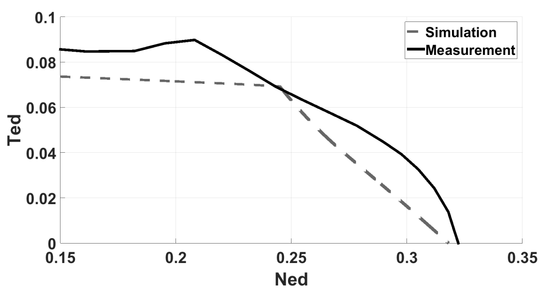

Figure 12. Steady state T_{ed}-N_{ed} characteristics.

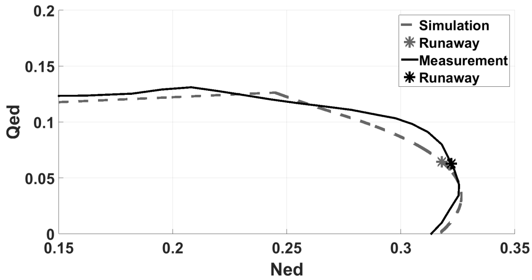

Figure 13. Steady state Q_{ed}-N_{ed} characteristics.

6. Conclusions

The measured dynamic characteristics showed behavior according to the stability criterion predicted by Martin. The oscillations were damped and the system regained steady-state operation at runaway after a short period of time. A significant deviation between the dynamic and steady-state characteristics was observed at values above $N_{ed} = 0.25$. The same result was obtained for a high head Francis turbine model by Nielsen in 1990 and explained by means of inertia in the water masses inside the turbine. By redefining the net head removing the inertia, the dynamic and steady-state characteristics coincide.

Transient simulations show that it is possible to reproduce the physics in the measurement by using an analytical model for the pump-turbine in the system. The simulations show good accuracy with regards to flow and rotational speed during the sequence. Oscillations with a period of approximately T = 4 s are observed in the measured data. In the simulations, the oscillations have a longer period and show larger dampening than the measured data. This deviation is mainly caused by the modeling of the turbine; the slope of the unit torque-unit speed and unit flow-unit speed are not steep enough and have a lower absolute value than measured. Simulations can only be as accurate as the input data and for the modeling of the turbine there will be deviations. The model used is a a modified high head Francis turbine model and it is not able to reproduce the steepness or S-shape of the characteristics around runaway for the pump-turbine, hence compromises must be made during the tuning of the transient model. In this paper the main focus was to achieve the best possible fit with the measured dynamic sequence. The rotational speed and flow rate at steady-state runaway show a good match to the measured values together with the gradient of the increase in speed, $\frac{dn}{dt}$.

Transient calculations of a pump-turbine prototype with the turbine model must be carried out with great caution. The stability criterion given by Martin depends on the gradient of the unit torque-unit speed at runaway. As shown in this paper the analytical model is not able to reproduce the correct steepness of the characteristics around runaway for the model pump-turbine. Hence, adjustments of the model according to the measurements was necessary. This tuning is often not possible with prototypes due to lack of measured data. As an initial calculation in the early stage of a project when no measured data are available the turbine model can be used, however measured data from model tests are recommended for analysis with purpose to reveal instability.

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Conflicts of Interest: The authors declare no conflict of interest.

Abbreviations

A	Area (m^2)
b_1	Linear coefficient of relative torque on relative flow (-)
B_1	Inlet height (m)
D	Diameter (m)
g	Gravitational constant (m/s^2)
H	Head (m)
h	Reduced head , $\frac{H}{H_n}$ (-)
I	Inertia (m^2/s^2)
n	Rotational speed (rpm)
p	Pressure (Pa)
Q	Flow rate (m^3/s)
q	Reduced flow rate, $\frac{Q}{Q_n}$ (-)
R_d	Draft tube loss coefficient (-)
R_f	Viscous loss coefficient (-)
R_m	Mechanical loss coefficient (-)
r	Radius (m)
T_a	Time constant rotating masses (s)
T	Period of oscillations/Torque (s)/(Nm)
T_w	Time constant of water masses in penstock (s)
T_{wt}	Time constant of water masses in the turbine (s)
u	Peripheral speed (m/s)
α	Guide vane angle (rad)
β	Runner blade angle (rad)
η	Turbine efficiency (-)
η_g	Efficiency generator (-)
κ	Guide vane opening degree (-)
ρ	Density of water (kg/m^3)
Ω	Reduced rotational speed, $\frac{\omega}{\omega_n}$ (-)
ω	Rotational speed (1/s)

Sub- and superscript

dyn	Dynamic
net	Net
n	Nominal value
opt	Best efficiency point
r	Rated value
1	Runner inlet
2	Runner outlet
*	Best efficiency

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Paper 2

Investigation of Stability Criteria for Reversible Pump–Turbines with Laboratory and Prototype Measurements
E.C. Walseth, T.K. Nielsen and B. Svingen
Submitted

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Paper 3

Prototype Study on the Effect of Reversible Pump–Turbine Characteristics on System Transients
E.C. Walseth, T.K. Nielsen and B. Svingen
Submitted

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