Master's thesis

NTNU
Norwegian University of Science and Technology
Faculty of Engineering
Department of Energy and Process Engineering

Lena Rostad

Implementation of a turbine governor on the Francis test rig at The Waterpower Laboratory

Master's thesis in Mechanical Engineering
Supervisor: Pål-Tore Storli
Co-supervisor: Truls Edvardsen Aarønes
June 2022

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MASTER WORK

for
student Lena Rostad

Spring 2022

"Implementation of a turbine governor on the Francis test rig at The Waterpower Laboratory"
Implementering av en turbinregulator på Francis-testriggen ved Vannkraftlaboreatoriet

Background

Turbines in operation are almost never experiencing steady state conditions. The rotational speed is changing due to the state of the electrical grid, and the turbines are required to alter their power to stabilize the rotational speed. This induce oscillations in the hydraulic domain as well, and the consequence is that the turbine governor is always in motion.
The Francis turbine test rig at the Waterpower laboratory has undergone modifications as a result of new desires for experimental testing. Further modifications will include a proper turbine governor so that the test rig can be operated as an actual turbine is operated. Signals between the Smart grid Laboratory and the Waterpower Laboratory will make it possible to perform tests where the turbine is subjected to realistic changes in operating conditions.

Objective

The student shall take part in the work needed to implement the turbine governor. The student must be involved in all stages in the implementation work, from the initial design to the physical assembly of the system.

The following tasks are to be considered:

1. Literature review of governing stability of hydropower plants
2. Stability analysis of the Francis test rig at the waterpower laboratory leading to a set of governor parameters
3. Take part in dimensioning and decisions on components and their assembly and commissioning
4. If time allows for it, take part in relevant measurement campaigns, and analyse the results

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The master work comprises 30 ECTS credits.

The work shall be edited as a scientific report, including a table of contents, a summary in Norwegian, conclusion, an index of literature etc. When writing the report, the candidate must emphasise a clearly arranged and well-written text. To facilitate the reading of the report, it is important that references for corresponding text, tables and figures are clearly stated both places.
By the evaluation of the work the following will be greatly emphasised: The results should be thoroughly treated, presented in clearly arranged tables and/or graphics and discussed in detail.

The candidate is responsible for keeping contact with the subject teacher and teaching supervisors.

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

According to “Utfyllende regler til studieforskriften for teknologistudiet/sivilingeniørstudiet ved NTNU” § 20, the Department of Energy and Process Engineering reserves all rights to use the results and data for lectures, research and future publications.

<u>Submission deadline:</u> To be found in Inspera.

☒ Work to be done in lab (Waterpower lab, Fluids engineering lab, Thermal engineering lab)
☐ Field work

Department for Energy and Process Engineering 12/1 2022

Pål-Tore Storli
Supervisor

Co-Supervisor(s): Truls Edvardsen Aarønes, Hymatec/NTNU

Abstract

The aim of this master thesis is to implement a turbine governor on the Francis turbine test rig at the Waterpower Laboratory at NTNU, so it can be operated as a Francis turbine in an actual hydropower plant. To be able to operate the turbine in the Waterpower Laboratory as a turbine in an actual hydropower plant, a set of governor parameters is set. These are found based on theory about hydropower and control engineering. Block diagrams and APF diagrams are analysed for both rigid and elastic systems. The governing system is tested according to requirements set by Statnett SF, as for all real hydropower plants. These tests include closing time, frequency step test and isolated grid test.

The frequency on the electrical grid must maintain stable at 50 Hz and not deviate by more than 0.1 Hz. The governor's most important task is to keep the frequency stable, which means that the power production must match the power demand. The power production is controlled by changing the guide vane angle to control the water flow rate through the turbine. Usually, the turbine unit is connected to a grid with many other units, sharing the job of keeping the frequency stable, which is called parallel operation. However, sometimes one power plant can be isolated from the others on an isolated grid, called island mode. Island mode is rare, but is used for sizing hydropower plants, as the worst case scenario is that the production unit is the only power production unit on the grid, and hence must keep the frequency stable all by itself.

Hydropower stability can be investigated using an Amplitude-Phase-Frequency (APF) diagram. The transfer functions for the most important parts of the system are put into a block diagram, and reduced to one transfer function for the total system. The transfer function can consider the elasticity of the system if necessary, and the APF diagram presents the stability margins. The Nyquist criterion is used to establish whether a hydropower system is stable or not. Time constants for the rotating masses, water masses and reflection time, as well as PID parameters, decide if the system is stable.

Even though the APF diagrams for the Waterpower Laboratory show stable systems, the results from the tests show unstable governing, and do not satisfy the requirements. This is because of backlash in the mechanical parts between the servomotor and the guide vanes. This means that the guide vanes are not able to follow the servomotor's movement, and can therefore not find the correct angle to stabilize the frequency.

Sammendrag

Målet for denne masteroppgaven er å implementere en turbinregulator på Francis-testriggen ved Vannkraftlaboratoriet ved NTNU, så den kan bli driftet som en francisturbin i et ekte kraftverk driftes. For å kunne drifte turbinen i Vannkraftlaboratoriet som en turbin i et ekte kraftverk, må det settes passende reguleringsparametere. Disse er funnet basert på teori om vannkraft og reguleringsteknikk. Blokkdiagram og APF-diagram er analysert for både stivt og elastisk system. Regulatoren er testet i henhold til krav fra Statnett SF, slik som for alle ekte vannkraftverk. Disse testene inkluderer lukketid, frekvensstegtest og isolert nett-test.

Frekvensen på strømnettet må holdes stabilt på 50 Hz, og må ikke variere mer enn 0.1 Hz. Regulatorens jobb er å holde frekvensen stabil, som betyr at produsert effekt og effektbehov må være av samme størrelse. Effektproduksjonen bestemmes av åpningsgraden på ledeapparatet, som bestemmer mengden vann som strømmer gjennom turbinen. Vanligvis er turbinaggregatet koblet til strømnettet sammen med mange andre aggregat, der jobben med å holde frekvensen stabil deles mellom aggregatene. Likevel kan ett kraftverk isoleres fra de andre på et isolert nett, kalt øydrift. Øydrift er sjelden, men blir brukt for å dimensjonere vannkraftverk, siden det verste som kan skje er at ett aggregat er den eneste effektprodusenten på nettet, og dermed må holde frekvensen stabil selv.

Stabiliteten i et vannkraftsystem kan undersøkes ved hjelp av et Amplitude-Fase-Frekvens-diagram, kalt APF-diagram. Transferfunksjonene for de viktigste delene av systemet settes inn i et blokkdiagram, og reduseres til én transferfunksjon for hele systemet. Transferfunksjonen kan ta hensyn til elastisiteten i systemet om nødvendig, og APF-diagrammet viser stabilitetsmarginene. Nyquist-kriteriet er brukt for å finne ut om systemet er stabilt eller ikke. Tidskonstanter for de roterende massene, vannmassene og refleksjonstiden, i tillegg til PID-parametere, bestemmer om systemet er stabilt.

Selv om APF-diagrammene for Vannkraftlaboratoriet viser stabile systemer, viser resultatene fra testene ustabil regulering, og tilfredsstiller dermed ikke kravene. Dette er på grunn av slark i mekaniske deler mellom servomotoren og ledeskovlene. Det betyr at ledeskovlene ikke er i stand til å følge bevegelsen til servomotoren, og finner derfor ikke riktig vinkel for å kunne stabilisere frekvensen.

Acknowledgments

When I finished my bachelor's degree two years ago, I wasn't sure I wanted to continue with a master's degree. But a certain pandemic led to few job opportunities, so I decided to apply just to have something to do for the next two years. A master's degree couldn't hurt. The first two semesters Covid-19 was the only thing everyone talked about, and due to restrictions and I didn't see any of my fellow students. The first year was actually quite lonely even though I had everyone I loved near. It wasn't better when I received the grade F for the first time, and the only classmate I knew quit. When the second F came, I really wanted to give up. But the summer came and I passed both the failed courses, which was a real motivational boost!

After the summer the campus was beginning to open up, and I got a place at the Waterpower Laboratory for the next year. Here I met some of the people I now call friends, and we have spent many hours in the sofa group during the breaks talking, laughing and doing quizzes. I have also spent hours and hours reading about hydropower, Francis turbines and governors, which this thesis is the result of. I am really proud of my achievement, but the thesis would not be as good if it weren't for my supervisor Pål-Tore and co-supervisor Truls. Thank you so much for making time for me to ask questions and complain, for always being honest, for answering the same questions over and over again, for explaining things thoroughly, and for being so positive. I couldn't have asked for better guidance.

The thing I have enjoyed most about this thesis is the work done in the lab. Thank you again, Truls, for teaching me all I know about turbine governors and how things doesn't work most of the time, but also for being patient and helpful. However, this project could not have been done without the laboratory technicians. I am sincerely thankful to Joar, Halvor and Trygve for preparing the test rig and being interested during the experiments, and for fixing everything that wasn't working right away without hesitation. I would also like to thank Øyvind for lots of great advice.

Thank you also to my incredible family for always being so encouraging and proud of me, even when I'm doubting myself. And Even, you are the most inspiring, helpful, understanding and positive person I have in my life. I'm lucky to have you!

I am so thankful for my year at the Waterpower Laboratory. Here I met so many wonderful people, and I feel extremely lucky to have been included in the environment hidden inside this old building. I don't know how many hours I have spent in this building, or how many stairs I have walked, but they have all been worth it. Without a doubt, it has been the best year of my life.

Symbols

P	Power	[W]
η	Turbine efficiency	[%]
ρ	Water density	[m^3/kg]
g	Gravitational acceleration	[m/s^2]
Q	Volume flow	[m^3/s]
H	Head	[m]
t	Time	[s]
a	Pressure propagation velocity	[m/s]
v	Water velocity	[m/s]
λ	Friction factor	[-]
D	Diameter	[m^2]
T_w	Time constant for water masses	[s]
L	Length	[m]
A	Cross section area	[m^2]
z	Water level height	[m]
Δh	Head loss	[m]
f	Frequency	[Hz]
T	Time period	[s]
α	Friction factor	[s^2/m]
T_r	Reflection time	[s]
T_c	Closing time	[s]
c	Water velocity	[m/s]
Δh_c	Pressure rise	[m]
T_a	Accelerating time for the rotating masses	[s]
J	Polar moment of inertia	[kg m^2]
ω	Angular velocity	[s^{-1}]
M	Moment of force	[Nm]
b_p	Droop	[%]
R	Strength of regulation	[W/Hz]
K_p	Proportional constant	[-]
T_i	Integral time	[s]
T_D	Derivative time	[s]
h_w	Allievi's constant	[-]
y	Guide vane opening	[%]
ψ	phase margin	[°]
Δk	gain margin	[dB]
n	Rotational speed	[rpm]

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Chapter 1

Introduction

The total power consumption in the world increases every year, and because of the critical environmental situation, the demand for renewable energy increases even more. Hydropower is a renewable energy source which has been, and is, a very important energy source for Norway and the world. It is a very controllable form of power production, which means that the water is easy to control and can be stored for long periods of time if necessary. Also, pumped-storage power plants have made it possible to reuse the water by pumping it back into the reservoir. Hydropower is a very efficient form of power production, with efficiencies reaching up to 96 %. The controllability of hydropower production makes it very reliable, and the start and stop sequences are completed in a short time. This is an advantage when less controllable power production, such as wind or solar power, contributes to a greater extent.

However, hydro turbines in operation are rarely exposed to steady state conditions. As consumers turn off and on electrical equipment connected to the electrical grid, the power demand from the grid changes. The frequency of the electrical grid must stay stable at 50 Hz, which requires that the power produced is equal to the power demanded. The produced power from a hydropower plant is controlled by a governor, that changes the water flow through the turbine. The state of the electrical grid is never constant, which means that the turbine governor is always in motion.

The Waterpower Laboratory at NTNU has been involved in development and experimental tests of hydro turbines for over 100 years. The Francis test rig has been rebuilt during the autumn of 2021 due to new desires for experimental tests. These modifications will include a full-fledged turbine governor, which will make it possible to operate the Francis test rig as an actual turbine is operated.

Previously, the Francis turbine at the Waterpower Laboratory has not been controlled by a proper turbine governor. The situation up til now has been an electric linear actuator which was controlled manually. A signal from the control room was sent to the actuator and the piston moved the guide vanes. Frequency or power were not considered when the electric actuator changed the guide vane opening. Also, the signal from the control room only controlled the back and forward motion of the piston, not a set point. To get a specific guide vane opening could therefore sometimes be a challenge. In a real hydropower plant, this would not be sufficient.

A proper turbine governor is necessary to obtain realistic governing. A full-fledged governor changes the guide vane opening based on both a power signal and a frequency signal. These signals decide power output automatically based on a setpoint for both the power and the frequency. The frequency setpoint is constant at 50 Hz, and the power setpoint changes due to the change in the power demand. Since these signal values can change quite quickly, the time to change the guide vane opening must be short to maintain the turbine's rotational speed constant.

As the title implies, the most important part of the thesis is to actually implement the governing system on the Francis test rig. This means that the thesis must be solved both analytically and practically. The analytical part of the thesis is to find suitable governor parameters to obtain a stable system, and the practical part includes testing the governor's ability to stabilize the hydropower system based on several requirements.

1.1 Scope and limitation

The scope of this master thesis is listed below:

- Implement a full-fledged governor on the Francis test rig at the Waterpower Laboratory.
- Find governor parameters ensuring stable operation.
- Carry out experiments where the turbine is subjected to realistic changes in operating conditions.

The existing mechanical parts of the Francis rig, such as the guide vanes or the runner, will not be changed or improved for this project. Aspects regarding the power market, such as power reserves, power trade and economy will not be focused on in this thesis.

Chapter 2

The governed system; a hydropower plant

2.1 System layout

Norway has an average land area height of about 400 m above sea level, and therefore has particularly good natural conditions for hydropower production. In addition, the amount of precipitation in many areas is relatively large, which can be utilized for power production [8]. Norwegian hydropower plants produce about 136.7 TWh in an average year [6].

Hydropower is based on a very simple principle: rainwater is stored in a lake or available in a river, and this water flows into a channel that leads to a turbine. The water causes the turbine to spin, and the generator converts the rotational energy to electricity. Even though the amount of precipitation varies from year to year, and season to season, the water cycle is endless. In other words: a hydropower plant produces electricity from conversion of rotational energy in the turbine and generator caused by potential energy in the water [4]. The main parts of a hydropower plant in general are one or several turbines with generators, water channels, surge shafts and reservoirs, as illustrated in figure 2.1. Water flows from the upper reservoir through the water channels and the turbine, and then out of the system to the tailwater. The turbine and generator rotate with the same rotational speed, and the amount of water that flows through the turbine decide the amount of power that is produced [11].

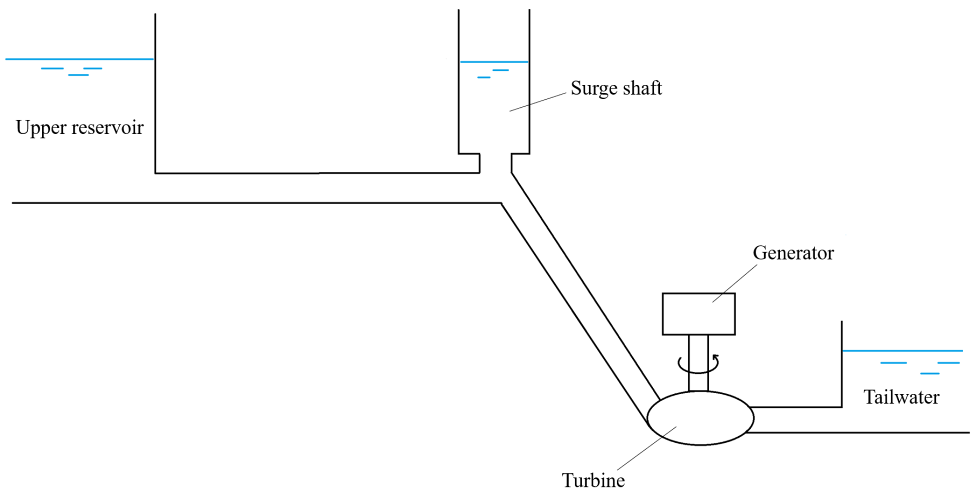


Figure 2.1: Simple illustration of a hydropower system

All hydropower plants must be designed and built based on the local conditions of the geographical area, such as hydrology, geology and topography. However, there are several similarities for the components in most hydropower plants, shown in figure 2.2.

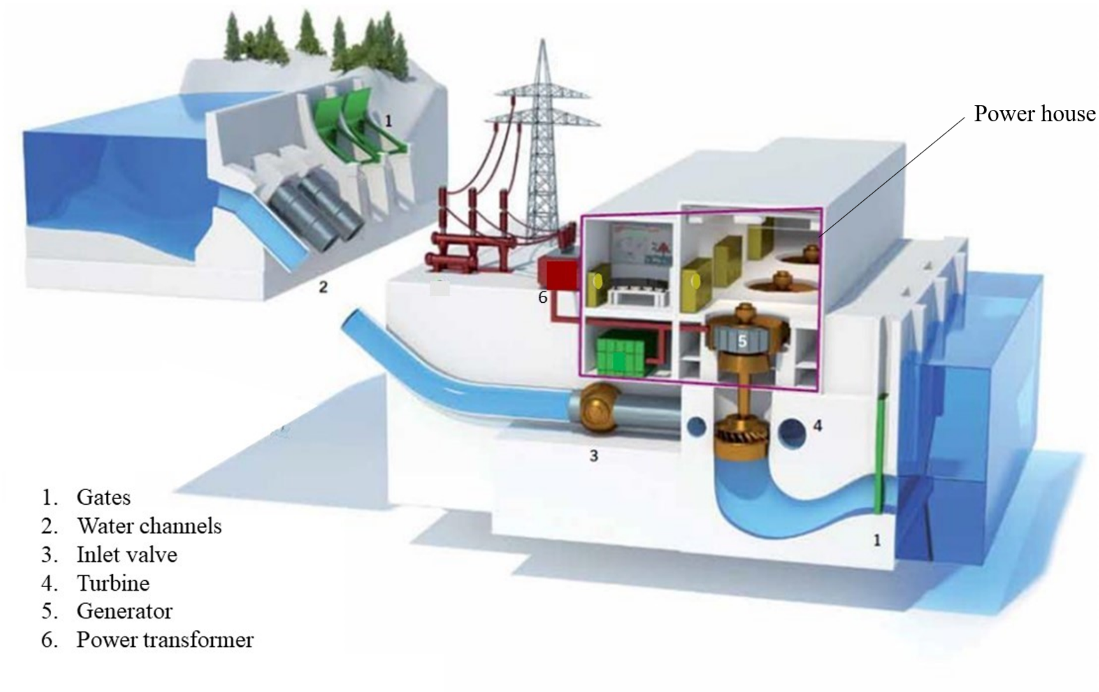


Figure 2.2: Model of a hydropower plant [4]

The gates can come in many different forms, but are there as water intakes, outlets, closing mechanisms or flood control. Many watercourses have requirements for minimum water flow. Power plants in these watercourses use the gates to release the required amount of water past the power plant. The water that flows into the power plant runs through the water channels, past the inlet valve and into the turbine. The turbine is connected to the generator with a shaft. This is where the energy in the water is converted to electrical energy. The generator is located inside of the power house with other equipment above the water channel. The power transformer converts the alternating current from the generator to a higher voltage before it is delivered to the electrical grid [4].

There are large differences in the amount of produced power from different hydropower plants. The amount of power produced, called hydraulic power, in a power plant is decided by the hydraulic head and the volume flow rate. The hydraulic head is the difference in height between the water levels of the upper reservoir and the tailwater. The hydraulic power, $\mathrm{P_h}$, is given by equation 2.1:

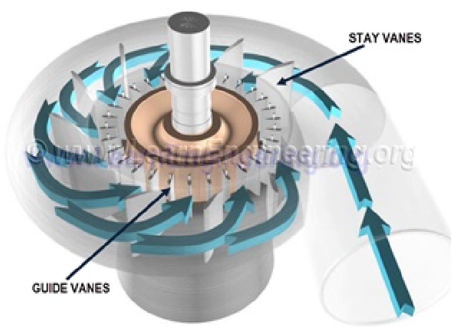
$$P_h = \eta \rho g Q H \tag{2.1}$$

where η is the efficiency of the turbine, ρ is the density of water, Q is the water flow rate and H is the head. This means that high head power plants with large volume flow give a large amount of power produced. In addition to hydraulic head, hydropower plants are often categorized based on operation terms, types of flow and production size. The two main types of hydropower plants are with reservoir, such as storage power plants and pumped-storage power plants, and run-of-river power plants [4].

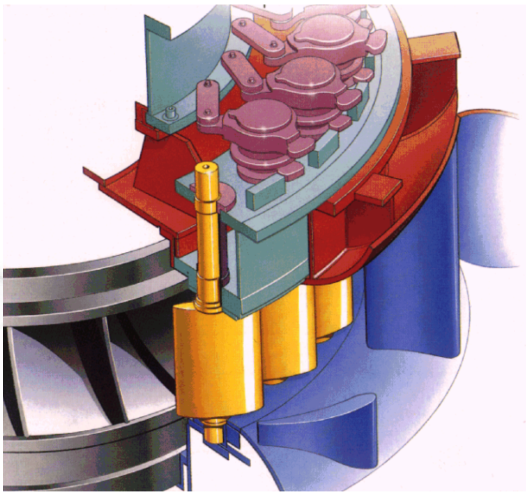
Hydropower plants produce power utilizing only water, a renewable resource, which makes it environmentally friendly as opposed to for example coal power. About 98 % of the Norwegian power production is renewable, mainly consisting of hydropower and wind power. This gives Norway the largest share of renewable power production and the lowest emissions from power production in Europe. Norwegian hydropower production is dominated by large hydropower plants with reservoirs that can store large amounts of energy in the form of water, which gives very flexible power production. Flexible hydropower production is important to balance the power market. The transmission network in Norway and trade between the Nordic and European countries ensure better stability in the power supply. Trade happens by large consumers and producers reporting power volumes for different power levels. The spot market price is determined by the power exchange, this provides a balance between power consumption and anticipated production [13].

2.2 Hydro dynamics

The dynamics in a hydropower plant are generally related to two factors; governing the hydraulic power to match the power demand and large load changes causing pressure oscillations and U-tube oscillations. The hydraulic power must at all times match the power demand from the electrical grid if the grid frequency is to be kept constant. The amount of power produced is changed by changing the guide vane angle. As illustrated in figure 2.3a, the water flows around the runner, and past the stay vanes and the guide vanes. The guide vanes, which are illustrated in figure 2.3b, control the water flow rate through the turbine by changing their angle.



(a) Water flow in a Francis turbine



(b) Guide vanes

Figure 2.3: Water flow and guide vanes in Francis turbines

This way, the frequency of the electrical grid maintains constant at 50 Hz. When a large load change happens, or if a total load rejection happens, the guide vane opening changes quickly, which will cause pressure oscillations in front of the turbine. The dynamic process can be described by figure 2.4 and equations for the water channels, surge shafts, turbine and governor [11]. These equations are presented below.

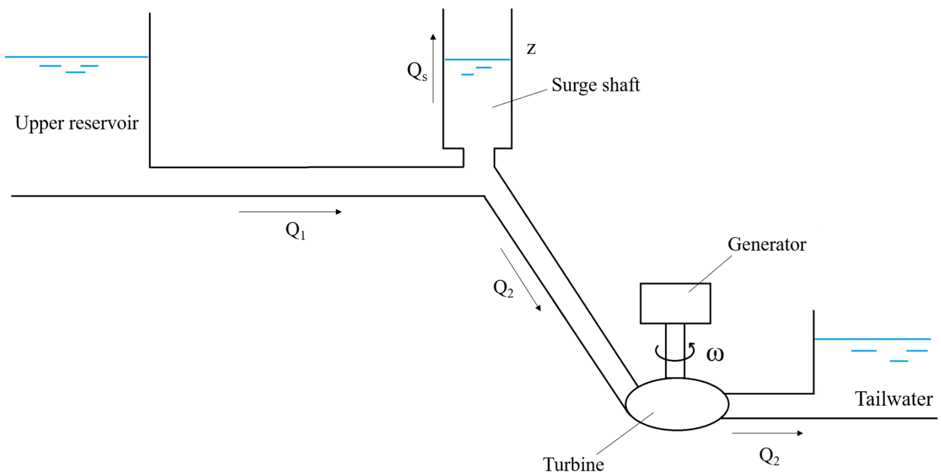


Figure 2.4: Simple hydropower system showing flow direction

The continuity equation for the water channel looks like equation 2.2:

$$\frac{\partial H}{\partial t} + \frac{a^2}{g}\frac{\partial v}{\partial x} = 0 \tag{2.2}$$

where H is the head, t is the time, a is the pressure propagation velocity, g is the gravitational acceleration and v is the velocity of the water [11]. Power balance of the fluid is:

$$g\frac{\partial H}{\partial x} + \frac{\partial v}{\partial t} + \lambda\frac{v|v|}{2D} = 0 \tag{2.3}$$

where λ is a friction factor and D is the diameter of the water channel [11].

The pressure propagation velocity, a, is given by equation 2.4:

$$a = \sqrt{\frac{K}{\rho}} \tag{2.4}$$

K is the compressibility factor and ρ is the density. The pressure propagation velocity occurs due to the elastic properties of the water and the water channels. The pressure propagation velocity is the speed of sound, which is different for different properties. In this thesis water in channels is considered, which is in practice a = 1200 m/s [11].

The elastic properties of the water channels depend on the pressure deviation in front of the turbine. When the water flow through the turbine must be reduced, the guide vane opening is reduced. Changing the guide vane angle creates a dynamic pressure change in front of the turbine, due to the inertia in the water masses. To prevent a too large pressure wave inside the water channel, surge shafts can be implemented to reduce the distance between the free surfaces, and hence reduce the acceleration time for the water masses, T_w [11].

The acceleration time for the water masses, T_w, is defined as the time it takes to accelerate the water masses from the nearest free surface upstream of the turbine to the nearest free surface downstream of the turbine, from zero to nominal volume flow, Q, under the influence of the hydraulic head, H. T_w is defined as:

$$T_w = \frac{Q}{gH}\sum\frac{L}{A} \tag{2.5}$$

where Q is the volume flow, H is the head, L is the distance between the free surfaces and A is the cross section area of the water channels.

In high head power plants the water channels can be extremely long, often several kilometers, which gives a long acceleration time for the water masses. To reduce T_w, a surge shaft is placed near the turbine. A surge shaft, as illustrated in figure 2.4, is a tank open to the atmosphere near the turbine where water can flow into when the pressure in the pressure shaft[1] increases. Hence, the pressure in front of the turbine is reduced. The best pressure-equalizing solution is often one surge shaft upstream and one downstream of the turbine [11].

The dynamics of a hydropower system can be described by the continuity equation for the water flow, which is given by equation 2.6:

$$Q_1 = Q_s + Q_2 \tag{2.6}$$

where Q_1 is the volume flow in the channel before the surge shaft, Q_s is the volume flow in the surge shaft and Q_2 is the volume flow in the channel after the surge shaft, as illustrated in figure 2.4. The volume flow in the surge shaft, Q_s, is found from equation 2.7:

$$Q_s = A_s\frac{dz}{dt} \tag{2.7}$$

where A_s is the cross section area of the surge shaft and z is the water level height in the surge shaft.

The equation of motion gives the change in water level height, Δz, as seen in equation 2.8:

$$\frac{L}{gA}\frac{dQ}{dt} = -\Delta z \tag{2.8}$$

where L is the distance between the reservoir and surge shaft and A is the cross section area of the water channel.

[1] The pressure shaft is the last part of the water channel which is very steep and leads the water into the turbine [4].

Rearranging equation 2.7 gives:

$$A_s \frac{dz}{dt} = Q_1 - Q_2 \tag{2.9}$$

where

$$Q_1 = vA \tag{2.10}$$

Choosing:

$$dz \approx \Delta z$$
$$dv \approx \Delta v$$
$$dt \approx \Delta t$$

This leads to the equation for change in water level height in the surge shaft, Δz, without losses:

$$\Delta z = \pm \Delta Q \sqrt{\frac{L}{gA_s A}} \tag{2.11}$$

The real change in water level height, Δz, can be found from equations 2.12 and 2.13 for load decrease and increase, respectively:

$$\Delta z = \Delta Q \sqrt{\frac{L}{gA_s A}} + \frac{1}{3}\Delta h \tag{2.12}$$

$$\Delta z = -\Delta Q \sqrt{\frac{L}{gA_s A}} - \frac{1}{9}\Delta h \tag{2.13}$$

where ΔQ is the change in volume flow and L is the distance between the turbine and the surge shaft. A_s is the cross section area of the surge shaft, A is the cross section area of the water channel and Δh is the head loss. The cross section area of the surge shaft, A_s should be calculated to avoid overflow [11].

Even though surge shafts reduce the pressure deviation in hydropower plants, they also introduce a challenge called U-tube oscillations, which is illustrated in figure 2.5. If the guide vanes close, the water masses upstream of the turbine will be stopped at the turbine inlet. The water masses upstream of the surge shaft cannot flow through the pressure shaft, it must flow into the surge shaft. The inertia in the water masses will cause the water level in the surge shaft to rise above the water level in the reservoir. The water will then flow towards the reservoir, the opposite direction of the main flow. Because of the inertia in the water masses the water level in the surge shaft will now sink below the water level in the reservoir, which will cause the water to flow back towards the surge shaft. The inertia in the water masses will make the water level in the surge shaft rise above the reservoir level again. These oscillations between the surge shaft and the reservoir, which are called U-tube oscillations, will continue until they are damped due to friction losses. U-tube oscillations are problematic if the water level rises above the walls of the surge shaft. The spilled water is lost energy, but can also cause damage in the nature if the amount is large. If the water level drops under the surge shaft walls, air can come into the system and cause cavitation and harm the turbine and pressure shaft [11].

The frequency of the U-tube oscillations can be found using equation 2.11:

$$\Delta z = \pm \Delta Q \sqrt{\frac{L}{gA_s A}} \tag{2.14}$$

The continuity equation is given in equation 2.15.

$$\frac{dz}{dt} = \frac{Q}{A_s} \tag{2.15}$$

Rearranging and using derivation to obtain equation 2.16:

$$\frac{dQ}{dt} = \frac{1}{A_s}\frac{d^2 z}{dt^2} \tag{2.16}$$

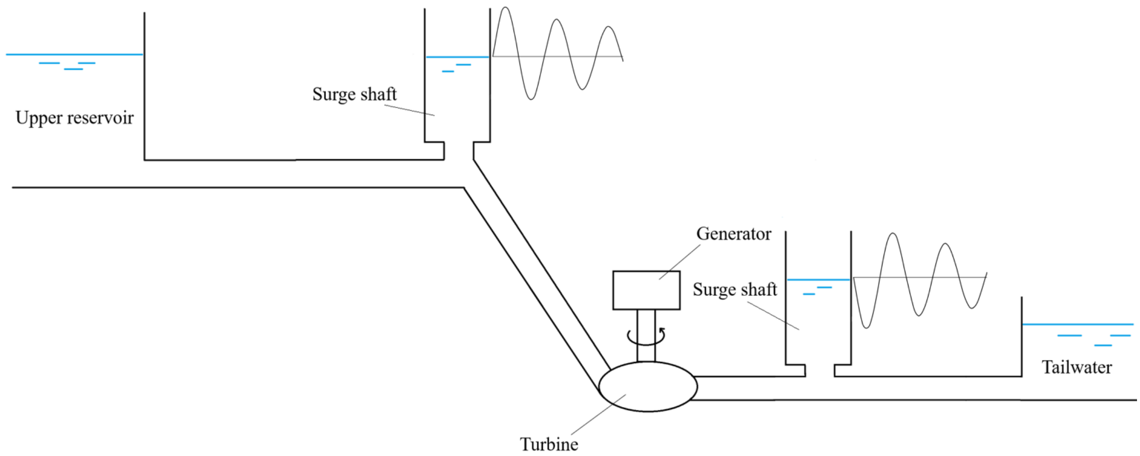


Figure 2.5: U-tube oscillations

Combining equations 2.16 and 2.8 gives equation 2.17:

$$\frac{L}{A_s}\frac{d^2z}{dt^2} \times \frac{1}{gAz} = 0 \tag{2.17}$$

which is now on the form of mx"/kx = 0 and can be put into the equation 2.18:

$$f = \sqrt{\frac{k}{m}} \tag{2.18}$$

The frequency of the U-tube oscillations is then:

$$f = \sqrt{\frac{g}{A_s(L/A)}} \tag{2.19}$$

which gives the time period:

$$T = \frac{2\pi}{f} \tag{2.20}$$

The smallest cross section area of the surge shaft that still gives steady state U-tube oscillations is called the Thoma cross section area. The Thoma cross section area, A_T, is found from equation 2.21:

$$A_T \geq \frac{LA}{2g\alpha\Delta h} \tag{2.21}$$

where α is the friction factor in the pipes [11]. This is the theoretical cross section area, so a safety factor is necessary to ensure steady state U-tube oscillations. For shorter water channels a safety factor of 1.5 is often used, and for longer channels 1.3 [3].

If the distance between the surge shaft and the turbine is long, the elastic properties must be considered. The elasticity causes pressure waves, or water hammer, in the pipes. If the power demand should suddenly drop, the governor must close the guide vanes and reduce the water flow. This is called a load rejection. When this happens the pressure in the water channel will rise. The velocity of the water at a point near the turbine will be reduced, meanwhile the velocity of the water at a point further upstream of the turbine will remain the same as before the guide vane closing, due to the inertia of the water masses. This causes a water hammer to move upstream, towards the reservoir. The pressure reflects on this constant pressure boundary, and the reflection propagates back towards the turbine. Accompanying this pressure propagation is the reversed flow, and reaching the turbine this gives negative pressure near the turbine and results in a water hammer back towards the turbine. The pressure rises in the water channel again, and oscillates until the friction loss dampens the water hammer. The time it takes for the water hammer to travel from the turbine to the nearest free surface and back is called the reflection time, T_r, and can be calculated using equation 2.22:

$$T_r = \frac{2L}{a} \tag{2.22}$$

where L is the distance between the turbine and the nearest free surface and a is the pressure propagation velocity, or the speed of sound in water through pipes, which is approximately 1200 m/s in practice.

In a real system it takes some time to close the guide vanes. This is called the closing time, T_c, which is the shortest time it takes to close the guide vanes from a fully open position, without creating a damaging pressure increase in the water channel. The closing time is calculated based on the dimensioning pressure in the water channel. To prevent damage caused by the pressure increase, Δh_c, it is often chosen to a value of about 10 % of the dimensioning pressure, due to the limitations of the water channels. The closing time can be found from equation 2.23:

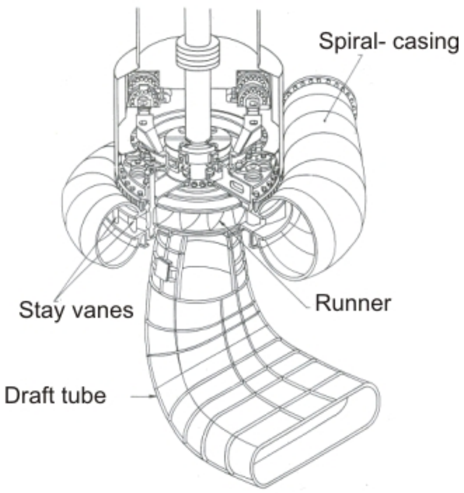
$$T_c = T_r \frac{a\Delta c}{g\Delta h_c} \tag{2.23}$$

where Δh_c is the pressure increase in the water channel and Δc is the change in velocity of the water [11].

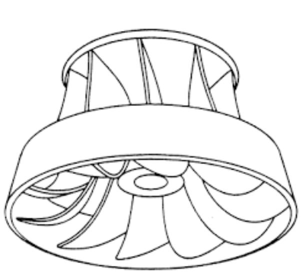
The closing time must be calculated to ensure that elastic oscillations do not harm the system, in other words the shortest time it takes to close the guide vanes without causing any damage. When a load rejection happens it is critical that the guide vanes close quickly, so acceleration of the turbine unit[2] rotation is prevented, but not so quickly that the water channel is damaged [11].

2.3 The Francis turbine

The Francis turbine is the most common type of turbine in hydropower plants today. The main parts of the Francis turbine are the runner, guide vanes, stay vanes, spiral casing, governor and draft tube, as seen in figure 2.6a. The water flows through the water channels and is led into the spiral casing. The water flows through the spiral casing where the stay vanes and guide vanes convert some of the pressure energy in the water to kinetic energy, and lead the water into the rotating runner, which is illustrated in figure 2.6b.



(a) Francis turbine [17]



(b) Runner [14]

Figure 2.6: Illustrations of a Francis turbine and runner

The stay vanes are fixed and placed at a certain angle to contribute to achieving high efficiency. The angle of the guide vanes is adjustable to control the volume flow through the runner. Figure 2.7 illustrates the Francis turbine test rig at the Waterpower Laboratory showing guide vanes, stay vanes, spiral casing and runner. The runner is usually a welded or cast stainless steel construction.

[2]The turbine unit is the generator and the runner connected with a shaft [11]

A shaft connects the runner and the generator, where the rotational energy is converted to electrical energy. The water flows through the runner outlet and out through the draft tube, which is shaped to convert the kinetic energy in the water to pressure energy [9].

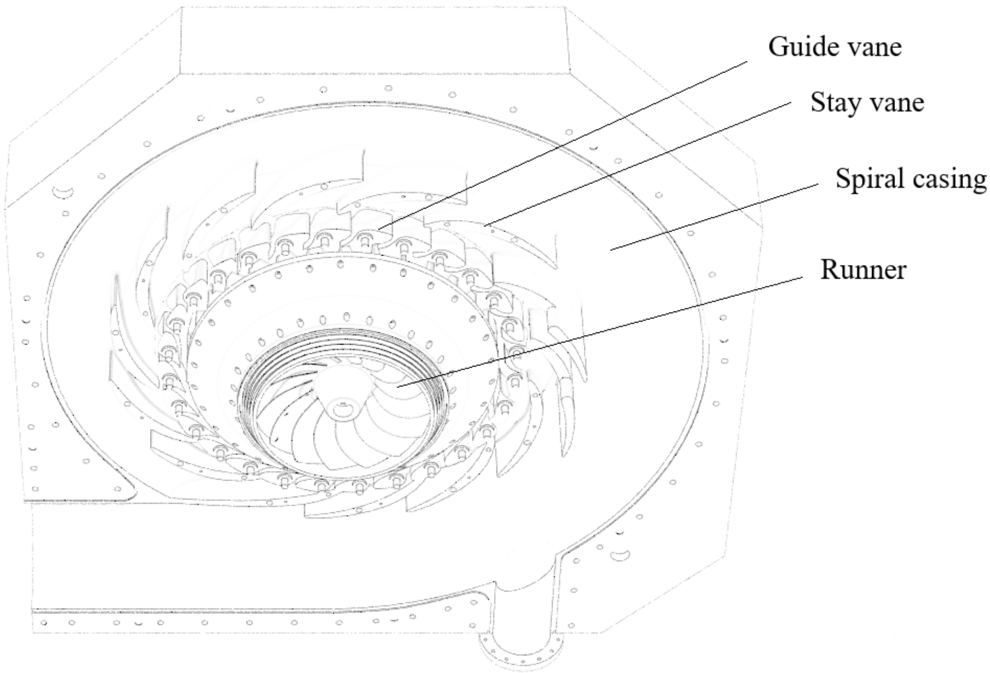


Figure 2.7: Francis turbine from frog perspective

The turbine unit, or the rotating masses in the system, are accelerated by the deviation between the power production and the power consumption. Because the masses of the runner and the generator often are relatively large, it takes some time for the water to accelerate them. This means that the turbine unit has a stabilizing effect on the system because the governor gets time to limit the acceleration before damage is done. The acceleration time for the rotating masses, T_a, which is the time it takes to accelerate the turbine unit from an angular velocity of zero to the nominal angular velocity, ω, at maximum volume flow, Q, will vary based on the size of the turbine unit.

T_a is found from equation 2.24:

$$T_a = J\frac{\omega^2}{P_{max}} \tag{2.24}$$

where J is the polar moment of inertia for the turbine unit, ω is the nominal angular velocity, and P_{max} is the maximum power output. For larger turbine units the T_a is in the range of 5 - 7 s. To obtain stable operation and prevent large pressure deviations in the water channels, the T_a/T_w ratio should be at least 6, meaning T_w should not be larger than 1 s when the T_a is approximately 6 s [11].

2.4 The generator

The generator converts the rotational energy from the turbine into electrical energy. The rotational power is given from equation 2.25:

$$\eta \rho g Q H = M\omega \tag{2.25}$$

where η is the efficiency of the turbine, H is the head, Q is the volume flow, M is the moment of force and ω is the angular velocity of the turbine unit. The acceleration law for rotating masses is shown in equation 2.26:

$$\sum M = J\frac{d\omega}{dt} \tag{2.26}$$

where J is the turbine unit's polar moment of inertia. The moment of force is found from the expression for rotating power:

$$P = M\omega \tag{2.27}$$

Power balance of the turbine and the generator gives the hydraulic power, P_h, as described in chapter 2.2, which creates an acceleration of the rotating masses, electric power to the grid (P_N) and the losses (P_L) [11], as seen in equation 2.28 and figure 2.8.

$$P_h = J\omega\frac{d\omega}{dt} + P_N + P_L = \eta \rho g Q H \tag{2.28}$$

Rearranging equation 2.28 gives the following equation:

$$J\omega\frac{d\omega}{dt} = P_h - P_N - P_L \tag{2.29}$$

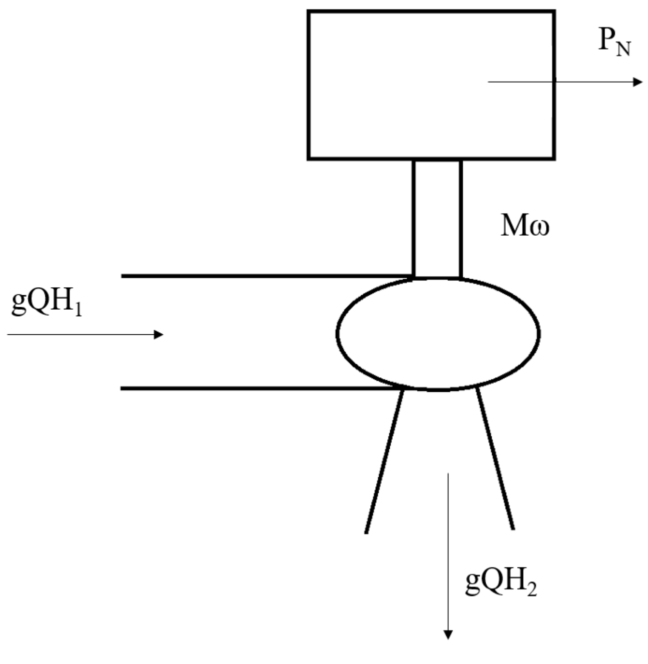


Figure 2.8: Power balance of the turbine and generator

As explained in chapter 2.1, the hydraulic power is decided by the head, H, and the water flow rate, Q. The head is constant, but the water flow rate is controlled by the guide vane opening. It is the governor's job to decide the guide vane movement. This is discussed further in the next chapter.

Chapter 3

Governing the system

3.1 The governor

The main purpose of the governor is to keep the grid frequency stable at 50 Hz. Since the frequency and rotational speed are coupled, the turbine unit's rotational speed must maintain constant. If the amount of produced power does not match the amount of power demanded from the electrical grid, the frequency will change. The governor changes the power output from the turbine by changing the guide vane opening. This means that when the power demand decreases, the governor makes the guide vane opening smaller to decrease the amount of water running through the turbine and hence reduce the power produced. It takes some time from the power demand decreases to the power production is decreased to the desired level. During this time more water than desired is flowing through the turbine, creating a power surplus that will accelerate the turbine unit [11].

Electricity from power production such as hydropower or wind power production is in the form of alternating current (AC), which cannot be stored efficiently at today's point[1]. Therefore, there has to be a balance between power demand and produced power for the frequency to be kept stable.

The water channels and the turbine are the parts in a hydropower plant that have the greatest influence on the governing. The water channel utilizes gravity to lead the water, often several kilometers, to the turbine. This creates strong forces and the governor only has limited opportunities to control the system. The greatest challenge for the governor is the inertia in the water masses, leading to a pressure deviation in the water channels. This pressure deviation can harm the water channels and the turbine, and must therefore be limited [11].

A modern governor is a system consisting mainly of a PID or PI controller, an oil pressure system and a servomotor. The most common type of governor today is an electro-hydraulic governor, which is a governor with a digital controlling system and a hydraulic servomotor [3]. The servomotor is illustrated in figures 3.1 and 3.2.

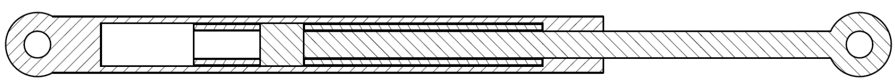


Figure 3.1: Principle sketch of a servomotor

The governor changes the angle based on two signals, a power signal and a frequency signal [10]. This is called power-frequency control, and is further explained in chapter 3.2.

[1] The produced power can be transformed from AC to DC for battery storage, but because of losses in the process it is not common.

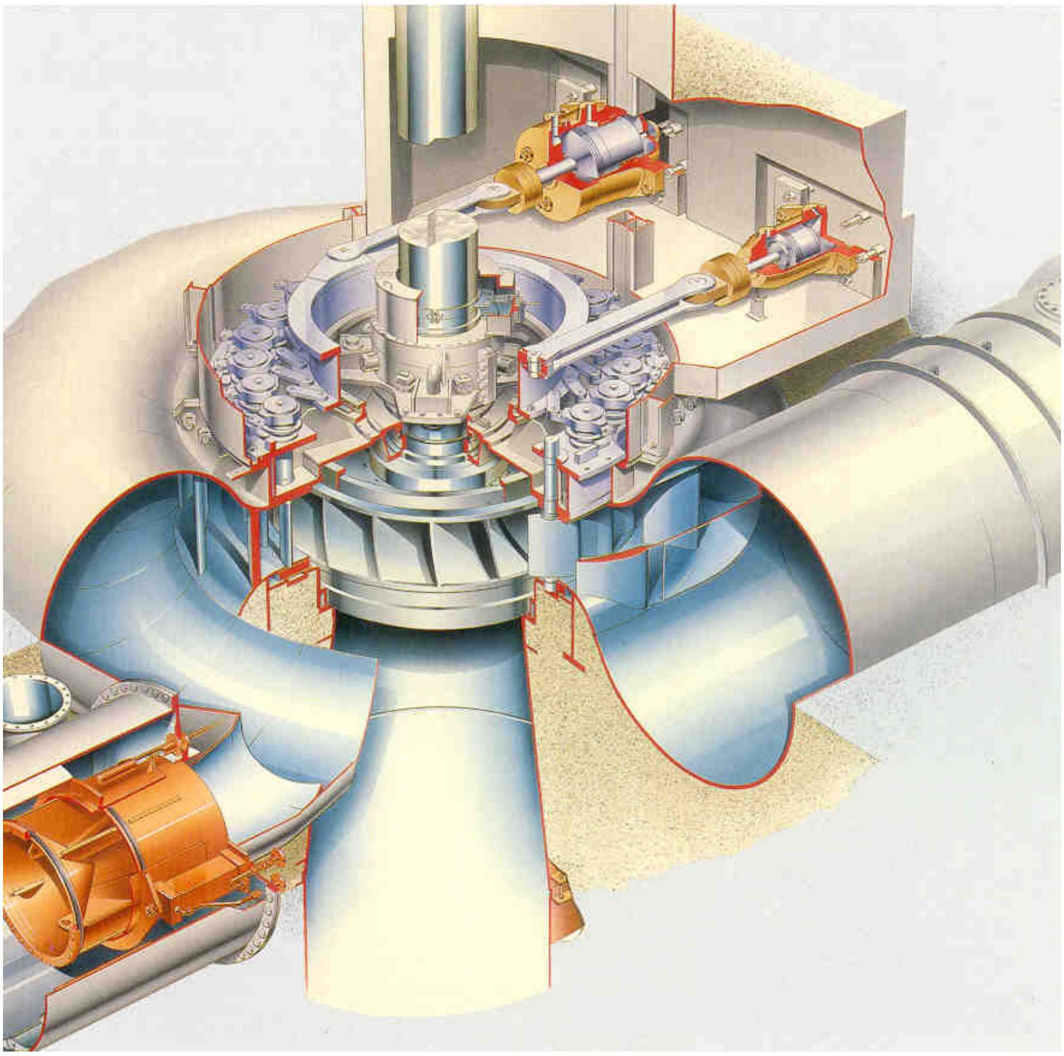


Figure 3.2: Francis turbine with two servo motors [2]

3.2 Modes of operation

Reliable power supply is critical in a modern society. This requires stable power production from the power plants. There are two different modes of operation that a power plant might be facing; island mode and parallel operation mode [10].

For the vast majority of operating time the production unit is connected to a grid with many other units, sharing the job of keeping the system stable. However, sometimes one power plant can be isolated from the others on an isolated grid [11]. In the following subchapters the two modes of operation are presented.

3.2.1 Parallel operation mode

Under normal conditions where all the power plants are connected via the electrical grid, the generators work in parallel mode. This means that if the power demand increases all the power plants will share the power production increase, meaning that all turbines will increase the power output less than in island mode, so the total power output will match the power demand. This is called droop [10].

The initial challenge of parallel operation mode is rise to power oscillations, which is resolved by using the speed droop mechanism. When several units are operating together on the grid, the overall required system power production is achieved without specification of the individual power production at each unit. The only prerequisite is that the sum of the power contributions is as required. Using a grid with only two power producing units A and B as an example, the system

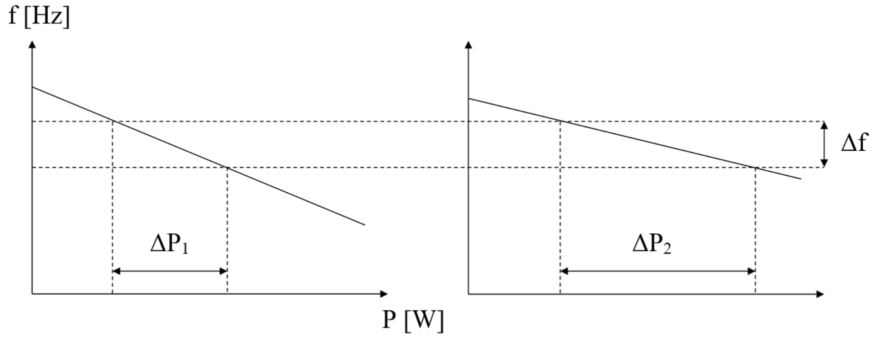


Figure 3.3: Droop and strength of regulation in two different units on the same grid

requires a power production of, say, 150 MW. Unit A producing 50 MW and unit B producing 100 MW is as good as both producing 75 MW each. This might give rise to power oscillations between the power plants if the operational point is not well defined [11].

Droop, or permanent speed droop, can be defined as "the percentage change in the frequency for a 100 % change of the power output from the production unit" [9]. As an example would a droop of 4 %, meaning a frequency deviation of 4 %, result in a 100 % change in the power output from the generator [10]. Also, it means that if the frequency drops from 50 to 49 Hz, a drop of 2 %, a governor with a droop of 5 % will increase power output by 40 % [9]. This is shown using equation 3.1.

$$\frac{\Delta P}{P_{max}} b_p = \frac{\Delta f - \Delta f_{db}}{f_0} \tag{3.1}$$

where ΔP is the change in power production, P_{max} is the maximum power from the turbine, b_p is the droop, Δf is the change in frequency, Δf_{db} is the deadband, which is chosen to zero, and f_0 is the nominal frequency [15]. The deadband is explained later. Rearranging equation 3.1 and inserting values:

$$\frac{\Delta P}{P_{max}} = \frac{\Delta f}{f_0} \times \frac{1}{b_p} = \frac{50Hz - 49Hz}{50Hz} \times \frac{1}{0.05} = 0.4 \tag{3.2}$$

The droop setpoint for each turbine is required to be chosen in the range of 2 - 12 % [15]. The strength of regulation of a turbine governor is decided by the droop and the nominal power of the unit. The strength of regulation is the unit's ability to change active power as a consequence of frequency deviation [16]. The strength of regulation, R, is given by:

$$R = -\frac{\Delta P}{\Delta f} \tag{3.3}$$

where ΔP is the active power deviation and Δf is the frequency deviation [9]. Figure 3.3 illustrates the droop and regulation strength of two different production units connected to the same electrical grid.

Two other requirements for a full-fledged governor are for the deadband and the measurement accuracy. The measurement accuracy of the frequency governing must be equal to or more accurate than 0.01 % = 0.005 Hz. It is also required that the deadband, Δf_{db}, is in the range of 0 - 0.5 Hz. Deadband is illustrated in figure 3.4 and found by utilizing equation 3.1. The turbine governor's frequency control loop gives a stationary contribution, ΔP, as a function of the frequency, Δf, outside of the deadband, Δf_{db}, see figure 3.4 [15].

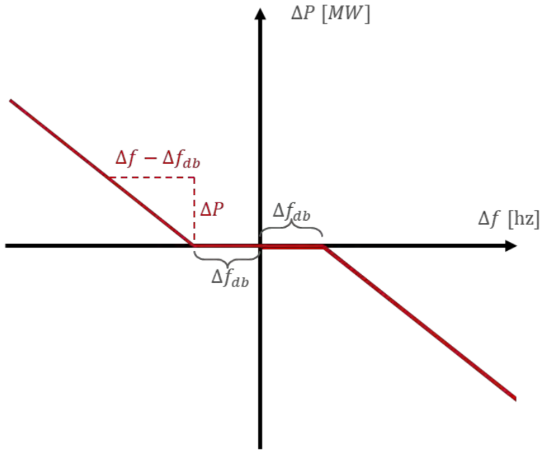


Figure 3.4: Governor response as a function of frequency for parallel units [15]

3.2.2 Island mode

Sometimes the grid connections fail due to various reasons, and so can one power plant be isolated from the rest of the electrical grid and hence be the only power plant, or even the only power production unit, in the area. This is called island mode or isolated grid, which means that the turbine unit operates in constant speed control [10].

High quality frequency governing is required to keep the frequency in the required range between 49.9 Hz and 50.1 Hz, and at the same time produce the required amount of power [11]. The Norwegian government requires that if a power plant does not have a full-fledged governor[2] it shall disconnect at overfrequency. This is because turbine units without frequency governing are not able to control the frequency, which can cause instability on the grid [15].

Island mode is very rare today, but in Norway it has been relevant for sizing hydropower plants, as the worst case scenario is that the production unit is the only power producing unit on a local isolated grid and hence must keep the system stable all by itself. Island mode is the most challenging mode of governing, and this mode is the case when investigating governing stability [10].

3.3 Governing stability

To investigate whether a hydropower system is stable or not, it can be modeled in the frequency plane using Laplace transformation. The transfer functions for the most important parts of the system, such as the governor, water channels and turbine unit, are put into a block diagram as illustrated in figure 3.5 [11]. As explained earlier, the rigid equations do not take the elastic properties of the water or the water channels into consideration. The elastic functions will therefore be more complex to solve, but will give a more realistic picture of the hydropower system. The transfer functions for the rigid system are derived in Appendix A.

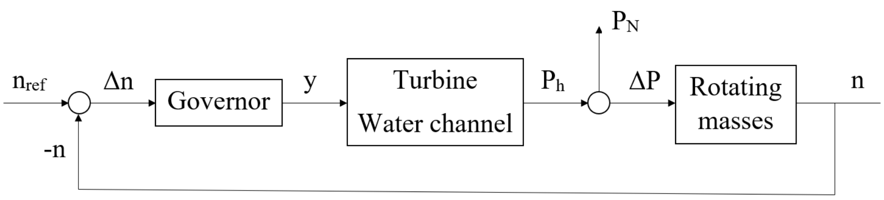


Figure 3.5: Block diagram for a hydropower system

The governor consists of two or three equations, based on the proportional (P), integral (I) and derivative (D) terms. A PI controller for a rigid system can be described by the transfer function

[2]A full-fledged governor is required to have a frequency governing function [15].

in equation 3.4.

$$H_G = K_p \frac{(1+T_i s)}{T_i s} \tag{3.4}$$

where K_p is the proportional constant and the T_i is the integral time.

The transfer function for the rotating masses is given in equation 3.5:

$$H_R = \frac{1}{T_a s} \tag{3.5}$$

The rigid transfer function for the turbine and waterway is given in equation 3.6:

$$H_W = \frac{(1 - T_w s)}{(1 + 0.5 T_w s)} \tag{3.6}$$

The total block diagram will look like figure 3.6 [11].

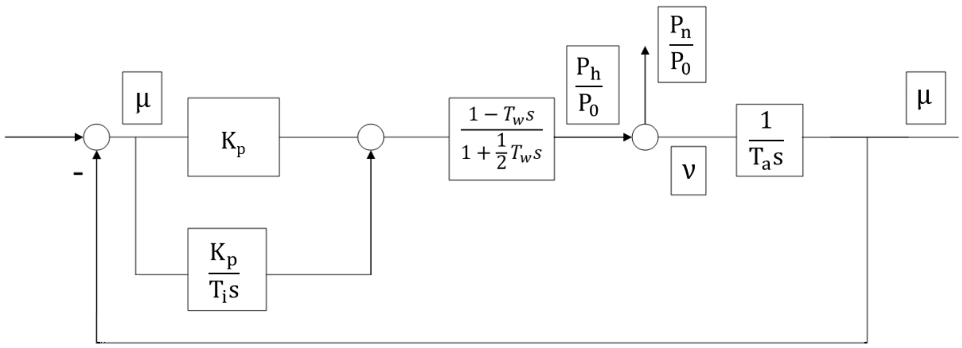


Figure 3.6: Block diagram for the total system

As described in Balchen's "Reguleringsteknikk" ([1]), the block diagram can be reduced as illustrated in figure 3.7. This is also explained in Appendix B.

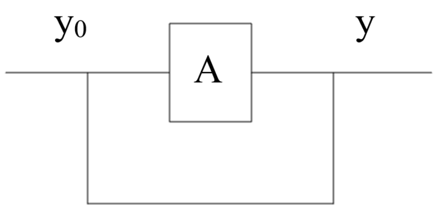


Figure 3.7: Reduced block diagram

Reducing the block diagram in figure 3.6 gives the total transfer function for the system, rigid and elastic, in equations 3.9 and 3.10 [12]:

Rigid:

$$A(s) = \frac{K_p}{T_i T_a} \frac{(1+T_i s)}{s^2} \frac{(1-T_w s)}{(1+0.5T_w s)} \tag{3.7}$$

Elastic:

$$A(s) = \frac{K_p}{T_i T_a} \frac{(1+T_i s)}{s^2} \frac{(1-2h_w tanh(\frac{L}{a}s))}{(1+h_w tanh(\frac{L}{a}s))} \tag{3.8}$$

where the K_p is the proportional constant and T_i is the integral time [3]. The equivalent functions for a PID controller are:

Rigid:

$$A(s) = \frac{K_p}{T_i T_a} \frac{(1+T_i s)(1+T_D s)}{s^2} \frac{(1-T_w s)}{(1+0.5T_w s)} \tag{3.9}$$

Elastic:

$$A(s) = \frac{K_p}{T_i T_a} \frac{(1+T_i s)(1+T_D s)}{s^2} \frac{(1-2h_w tanh(\frac{L}{a}s))}{(1+h_w tanh(\frac{L}{a}s))} \tag{3.10}$$

T_D is the derivative time and the h_W is Allievi's constant, which is defined as the relationship between T_w and T_r as equation 3.11 says [11]:

$$h_w = \frac{T_w}{T_r} = \frac{\frac{Q_0 L}{g H_0 A}}{\frac{2L}{a}} = \frac{Q_0 a}{2 A g H_0} \tag{3.11}$$

The elastic equations are more complex to solve than the rigid ones. The tanh (hyperbolic tangent) function is notoriously unstable. It has similarities with the tan function, and has to be solved utilizing that. The elastic equation expressing the relationship between the hydraulic power, P_h, and guide vane position, y, is as equation 3.12 shows:

$$\frac{P_h}{y} = \frac{1-2h_w tanh(\frac{L}{a}s)}{1+h_w tanh(\frac{L}{a}s)} \tag{3.12}$$

where s is the complex variable

$$s = j\omega \tag{3.13}$$

The complex term s in the expression tanh((L/a)s) is problematic. This is solved by moving the complex term j outside of the parenthesis and so the tanh((L/a)s) is transformed into jtan((L/a)ω), and can now be solved as shown in equation 3.14 [12]:

$$tanh(\frac{L}{a} j\omega) = j tan(\frac{L}{a}\omega) \tag{3.14}$$

Now that the functions are solvable, the results can be presented in a Bode-diagram, or Amplitude-Phase-Frequency (APF) diagram, which shows the stability margins, as illustrated in figure 3.8. The Nyquist stability criterion, which is given in equation 3.15, is used to establish whether a system is stable or not. A system is stable if:

$$\angle A(j\omega) > -180° \tag{3.15}$$

when

$$|A(j\omega)| < 1 = 0dB$$

The terms phase margin, ψ, and gain margin, Δk, are used to ensure stability. The phase margin tells where the phase curve is located relative to the -180° line at the cross frequency point[3]. The gain margin is the distance between 0 dB and the amplitude curve when the phase curve crosses the -180° line [11]. This is illustrated in figure 3.8.

Often it is also required that:

$$\psi > 45° \tag{3.16}$$

and

$$\Delta k > 2 = 6dB \tag{3.17}$$

Disturbances that occur above the cross frequency will not influence the stability of the governing. Therefore, it is desired that the elastic frequency is located above the cross frequency. This can be adjusted by tuning the PID parameters.

[3]The cross frequency is the point where the amplitude curve crosses the 0 dB line [11].

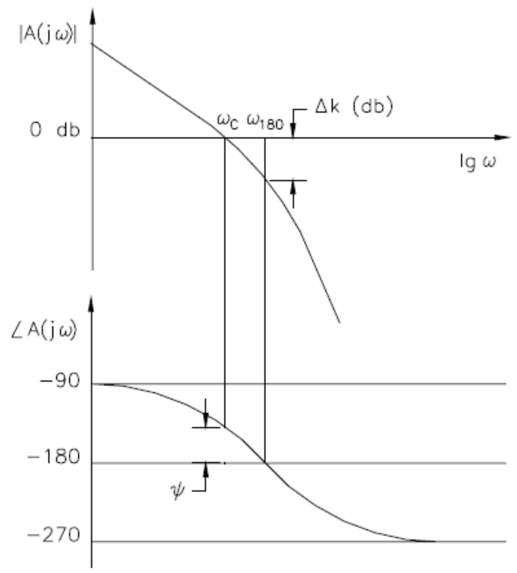


Figure 3.8: Example of an APF diagram for a rigid system [3]

3.4 PID controller

The simplest form of controller is the proportional controller, or P controller. The frequency controller measures the rotational speed, n, and compares it to the reference speed, n_{ref}. The guide vane opening, y, is changed proportionally to the deviation between the rotational speed and the reference speed, as shown in equation 3.18:

$$\Delta y_P = K_p \Delta n \tag{3.18}$$

which means that the guide vane opening degree is changed until the rotational speed becomes constant. In other words; the rotational speed will stabilize at another value than n_{ref}, which is not sufficient in a hydropower system. This means that this type of controller will not give stable governing.

To obtain stability, the integral term is introduced. The PI controller consists of both the proportional term and an integral term, as given in equation 3.19:

$$\Delta y_{PI} = -(K_p \Delta n + \frac{K_p}{T_i} \int \Delta n dt) \tag{3.19}$$

where the T_i is the integral time. With a PI controller a zero deviation between the rotational speed and the reference speed can be achieved. This means that the PI controller gives stable governing, and therefore is sufficient in a hydropower plant, but sometimes a derivative term is introduced in addition, to make the governing more precise. This results in a PID controller, which is shown in equation 3.20 [11]:

$$\Delta y_{PID} = -(K_p \Delta n + \frac{K_p}{T_i} \int \Delta n dt + K_p T_D \Delta n) \tag{3.20}$$

where the T_D is the derivative time. The derivative term reacts on the rapid changes in the deviation, which means that it does not influence the speed deviation [1]. In hydro turbine governing some degree of measuring error and noise is unavoidable. This can trigger the derivative term. Because this noise causes no damage, a PI controller is often sufficient [11].

There are several ways to find suitable PID parameters. In this thesis it will be focused on two methods; the P-I-D-method and the Ziegler-Nichols' closed loop method.

3.4.1 P-I-D method

The P-I-D method is a simple and intuitive method, but can be time consuming because it is based on trial and error.

The initial values of the PID parameters are set to obtain a pure P controller, with K_p and T_D set to zero and T_i set very high. Then the K_p is increased to a value where stability is obtained, meaning that the response gives damped fluctuations. Often the value of K_p is chosen to 1 and then increased or decreased to a value that gives damped fluctuations.

When the K_p value is found, the T_i is adjusted to obtain a PI controller. The value of the integral time is reduced to a value that gives a little too poor stability. An estimate for this value can be $T_i = T_p/1.5$. The T_p is the period of the damped fluctuations in the P controller. When T_i is found, the K_p is slightly reduced, until stability is once again achieved.

If the derivative term is necessary, the same procedure as for the PI controller can be used, but the value of the integral time is set to $T_i = T_p/2$. In addition, the T_D is set to a value of about 1/4 of T_i. This can cause instability, which can be restored by slightly increasing the K_p [7].

3.4.2 Ziegler-Nichols method

This method is based on results from experiments published by Ziegler and Nichols in 1942. This method is one of the most popular methods for tuning PID parameters.

For this method as well, the initial value for K_p and T_D is zero, and T_i is set to a very high value. Then K_p is increased to a value where the system becomes unstable. This value is called K_{max} [5]. The K_p is now reduced by an amount according to table 3.1:

Table 3.1: Settings for P, PI and PID controllers according to the Ziegler-Nichols method

	K_p	T_i	T_D
P controller	0.5 K_{max}	∞	0
PI controller	0.45 K_{max}	$T_{crit}/1.2$	0
PID controller	0.6 K_{max}	$T_{crit}/2$	$T_{crit}/8$

When the K_p is found, and the system is stable, T_i and T_D are calculated according to the table above/below. T_{crit} is the critical period where $K_p = K_{max}$. If the system becomes unstable after introducing the integral or derivative terms, the K_p can be adjusted down until stability is achieved [7].

Chapter 4

Description of method

The analytical part of this thesis includes calculating time constants as discussed earlier, meaning the reflection time, T_r, closing time, T_c, acceleration time for the rotating masses, T_a, and the acceleration time for the water masses, T_w. To find these constants, measurements from the Waterpower Laboratory were used. These measurements are shown in figures 4.1, 4.2 and 4.3.

All tests for this project are conducted in open loop mode, which means that the water flows from the overhead tank to the tailwater, which gives a head of 12 m. Figures 4.1, 4.2 and 4.3 illustrate the lengths of the different pipes and tanks, as well as the diameters of the pipes in the Waterpower Laboratory.

The downstream tank and the pressure tank will be filled with water, and will therefore not be regarded as free surfaces. The calculations are simplified by neglecting the flow meters and the diameter variation in the narrowest pipe, as well as using estimates for some of the lengths due to complex geometry. These simplifications will have little impact on the calculated time constants because the variation in diameter is small compared to the total length of the pipe system.

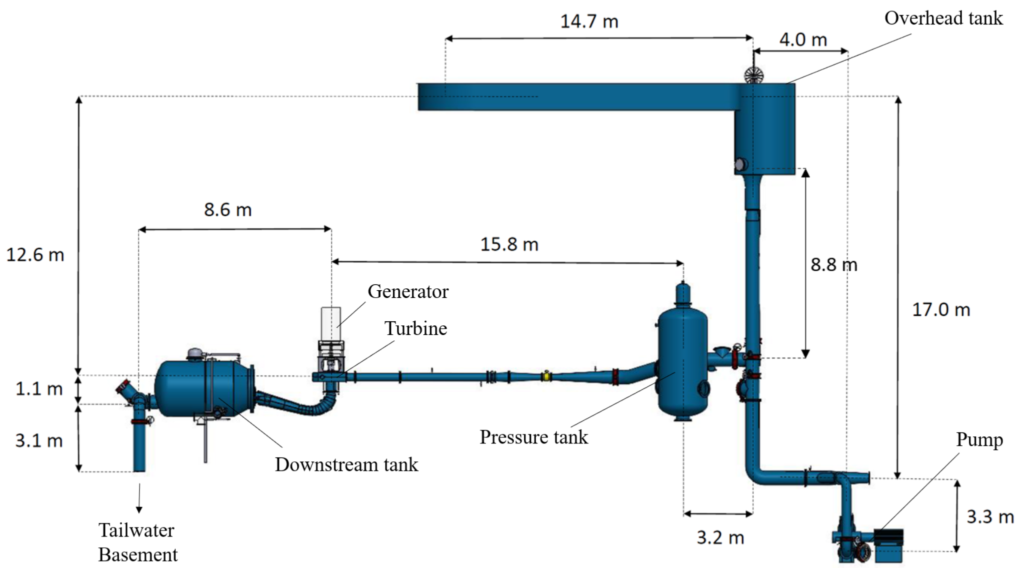


Figure 4.1: Pipe lengths in the Waterpower Laboratory

4.1 Reflection time

The reflection time is calculated using equation 2.22, where a = 1200 m/s. The length, L, is measured from the water surface in the overhead tank to the turbine. This length is, as seen from figures 4.1 and 4.2:

$$L = 12m + 3.2m + 3.8m + 15.8m = 34.8m \tag{4.1}$$

This gives the reflection time, T_r:

$$T_r = \frac{2L}{a} = \frac{2 \times 34.8m}{1200m/s} = 0.058s \tag{4.2}$$

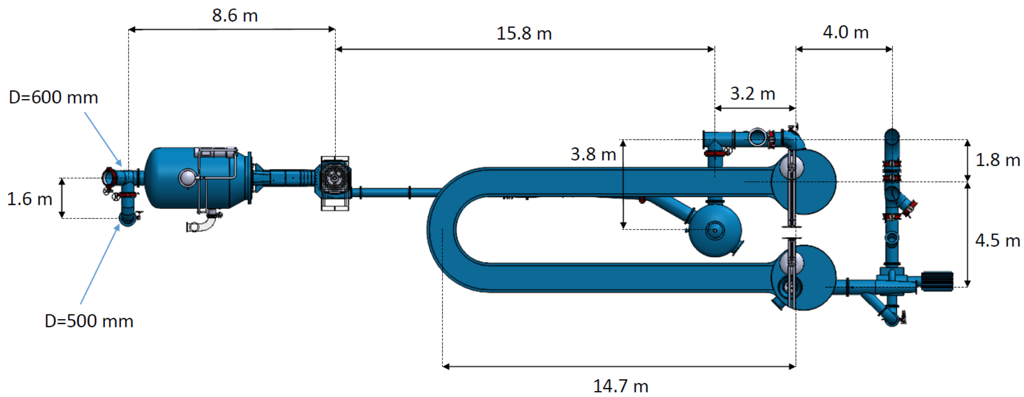


Figure 4.2: Lengths and diameters seen from above

4.2 Closing time

The closing time, T_c, is found using equation 2.23. From figure 4.3 the necessary diameter is found for calculating the change in the water velocity, Δc, at the turbine inlet. Running the water from the overhead tank gives a head of 12 m. Measurements from an earlier project at the Waterpower Laboratory show that the volume flow at 12 m head is 0.2 m^3/s [18]. The diameter of the pipe is 350 mm, which gives a cross section area of approximately 0.1 m^2. The change in water velocity, Δc, is then:

$$\Delta c = \frac{\Delta Q}{A} = \frac{0.2m^3/s}{0.1m^2} = 2m/s \tag{4.3}$$

As stated in chapter 2.2, the pressure increase in the pipe, Δh_c, should be 10 % of the dimensioning pressure, which in this case is 100 m. Equation 4.4 gives the closing time:

$$T_c = T_r \frac{a\Delta c}{g\Delta h_c} = 0.058s \times \frac{1200m/s \times 2m/s}{9.81m/s^2 \times 10m} = 1.42s \tag{4.4}$$

4.3 Time constants for the water masses and the rotating masses

The calculation of the acceleration time of the water masses, T_w, is simplified as described earlier in this chapter. T_w is in this case the time it takes to accelerate the water masses from the overhead tank to the tailwater in the basement, from a water flow rate of 0 m^3/s to 0.2 m^3/s at 12 m head. T_w is found using equation 2.5, and will in this case be:

$$T_w = \frac{Q}{gH}\sum \frac{L}{A} = \frac{0.2m^3/s}{9.81m/s^2 \times 12m} \times \left(\frac{19m}{0.28m^2} + \frac{15.8m}{0.1m^2} + \frac{4.3m}{0.11m^2} + \frac{4m}{4.85m^2} + \frac{1.6m}{0.28m^2} + \frac{3.1m}{0.2m^2}\right) = 0.49s \tag{4.5}$$

As mentioned earlier, it is desired that the T_w is less than 1 s to prevent large pressure deviations in the pipes and obtain stable operation. This result shows that this will not cause damage to the system in the Waterpower Laboratory. The next criterion is that the relationship between the acceleration time for the water masses and the acceleration time for the rotating masses, T_a/T_w, should be larger than 6. This gives a minimum value of T_a = 2.94 s, as shown in equation 4.6:

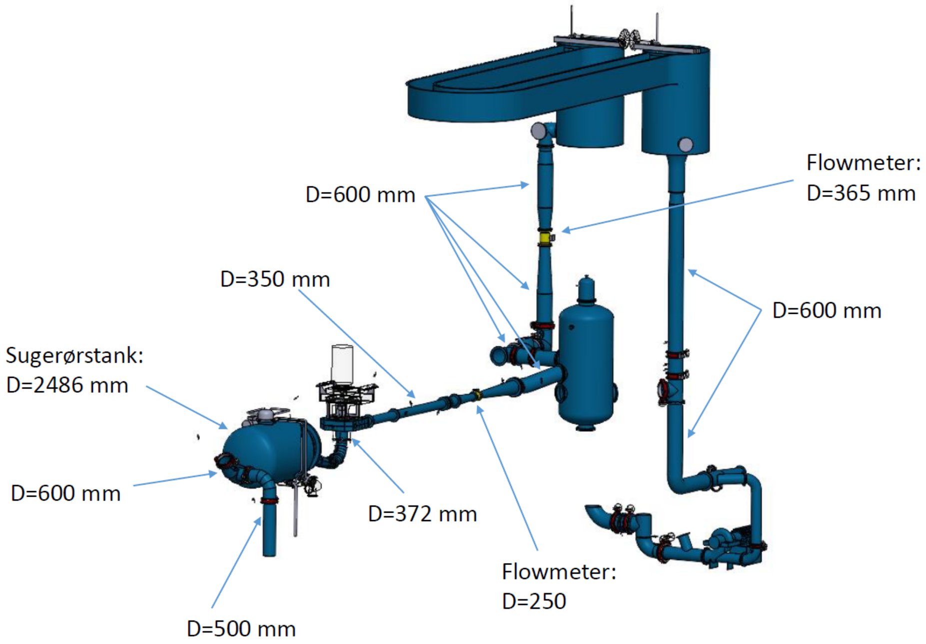


Figure 4.3: Model showing diameters of the pipes at the Waterpower Laboratory

$$T_a = 6T_w = 6 \times 0.49s = 2.94s \tag{4.6}$$

The acceleration time for the rotating masses, T_a, is in this case the time it takes to accelerate the turbine unit from an angular velocity, ω, of zero to the nominal angular velocity, ω_0, at the water flow rate, Q, of 0.2 m^3/s. T_a controls how fast the rotational speed changes based on a power contribution. A T_a of 2.94 s is short, and will allow the turbine unit to change the rotational speed fast. In the experimental tests, which are described later, the T_a is chosen to 6 s and 60 s at different times, to best imitate tests from actual power plants, where the turbine unit is larger and therefore has slower changes in rotational speed.

All the time constants calculated in this chapter are important for making the APF diagrams and concluding if the system is stable, as explained in chapter 3. This is discussed further in chapter 5.

4.4 PID tuning

The more practical part of this thesis is to connect the governor to the Francis turbine test rig, and test if the turbine in the Waterpower Laboratory can be operated as a turbine in a real hydropower plant. This involves finding dimensions and implementing parts, such as a servomotor and an oil pressure system, as well as looking at the governor's behaviour. This behaviour depends on the PID parameters.

It is important that the governor finds the correct guide vane angle fast, but not so fast that the pressure increases to a damaging level. It is also important that the system stabilizes in reasonable time. These properties are decided by the PID parameters. To find the PID parameters for this system none of the earlier mentioned PID methods are used. Most methods for PID tuning are based on instability, and this can be damaging for hydropower systems. Therefore, the PID parameters are found based on the elastic APF diagram to avoid the risk of causing damage. The required parameters for making the elastic APF diagram are T_a, T_w, T_r, K_p and T_i. It was decided that a PI controller is sufficient, to avoid instability caused by the derivative time, T_D. T_a is set to a value of 6 s, and T_w and T_r are set to the calculated values of 0.49 s and 0.058 s, respectively. The L/a ratio and Allievi's constant, h_w, are also parts of the transfer function for the system. Calculating T_r and h_w gives equations 4.7 and 4.8.

$$\frac{L}{a} = \frac{34.8m}{1200m/s} = 0.029s \tag{4.7}$$

$$h_w = \frac{T_w}{T_r} = \frac{0.49s}{0.058s} = 8.45 \tag{4.8}$$

To complete the APF diagram, the PI parameters must be found. The process of finding suitable values for K_p and T_i is based on trial and error, and are found suitable at values of 2 and 12 s, respectively. These values give quick and precise governing. The elastic APF diagram, as well as the rigid APF diagram, is presented in chapter 5.

It is unusual to use the earlier mentioned PID tuning methods for frequency governors in hydropower plants, due to the risks associated with instability. The common procedure for finding PID parameters while avoiding instability is using a simulation for the hydropower system. Even though the PID tuning methods are not used for the frequency governing at the Waterpower Laboratory, the PID parameters for the servomotor are found using the Ziegler-Nichols method.

If the turbine in the Waterpower Laboratory is to be operated as a turbine in an actual power plant is operated, the turbine governor has to be tested as the ones in power plants are tested. These tests are required by Statnett ([15]), and are conducted according to this.

4.5 Experimental tests

Several tests based on the requirements set by Statnett were conducted to ensure stable governing of the Francis turbine test rig at the Waterpower Laboratory during this project. The first test is measuring the closing time. The oil pressure system connected to the servomotor is fitted with throttle valves that limit the oil supply rate, so the measured closing time is longer than the calculated minimum closing time, and prevents damage to equipment or personal injury.

The next test is a frequency step test, which is normally conducted with the turbine unit connected to the grid. In this thesis the test is really a rotational speed step test. Using rotational speed or frequency will give the same result, but instead of applying a frequency disturbance, the rotational speed of the turbine unit is changed. This will cause the governor to change the amount of produced power. The test is conducted with a power setpoint of 80 % of max power and a droop, b_p, of 6 %. In this case the power setpoint is set to 80 % of 30 kW, which is 24 kW. It is required that the production unit contributes with a change in power, ΔP, which is found from equation 4.9. The deadband, Δf_{db}, is set to 0 Hz.

$$\frac{\Delta P}{P_{max}} b_p = \frac{\Delta f - \Delta f_{db}}{f_0} \tag{4.9}$$

Using the rotational speed of the turbine unit instead of frequency and rearranging equation 4.9 gives equation 4.10:

$$\Delta P = \frac{\Delta n}{n_{nom}} \frac{P_{max}}{b_p} \tag{4.10}$$

These equations are used to verify the test results in the next chapter.

The last, and probably the most important, test is the isolated grid test. The results from this test show if the governor is able to hold the frequency stable when the grid conditions are constantly changing. Also here the power setpoint is 24 kW, but now the droop is set to 2 % as described in Statnett's "NVF" ([15]).

Experiments in laboratories with large equipment introduce risks of harming both equipment and people. The Waterpower Laboratory has developed a risk assessment report, and an additional risk assessment report has been made for this project specifically, which is attached as Appendix C. All HSE internal procedures have been followed.

The following chapter contains the results from all the tests in addition to the block diagrams and APF diagrams, which are all described and discussed.

Chapter 5

Results and discussion

Testing the turbine governor involves several tests, as described in the previous chapter. Stationary conditions before the tests were commenced are shown in table 5.1.

Table 5.1: Stationary conditions before tests

Max power, open loop	P_{max}	30 kW
Head	H	12 m
Volume flow	Q	0.2 m^3/s
Nominal rotational speed	n_0	342 rpm
Servomotor's max stroke	l_{max}	60 mm
Guide vane's max angle	φ_{max}	13.6°

5.1 Analytical results

As described in chapter 3.2, block diagrams and APF diagrams are helpful to decide if the system is stable or not. The reduced block diagrams for this system, rigid and elastic, are given from equations 3.7 and 3.8. As described in chapter 3.2 and Appendix B, the block diagrams for the rigid and elastic systems with a PI controller are reduced to figures 5.1 and 5.2.

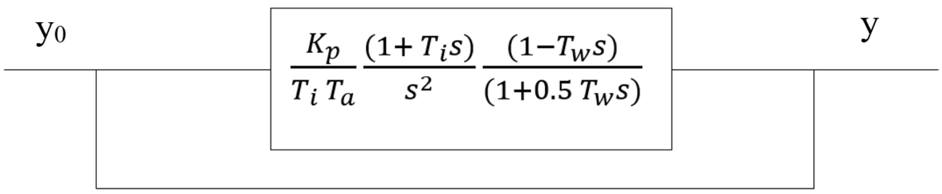


Figure 5.1: Block diagram for the rigid equation

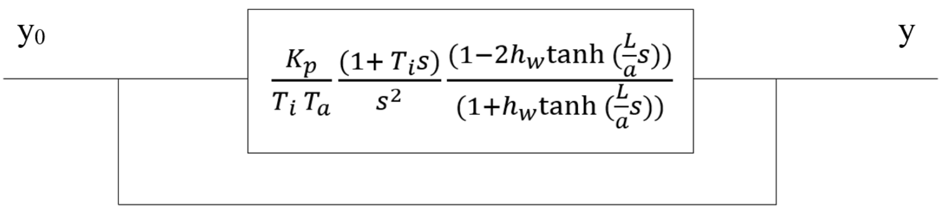


Figure 5.2: Block diagram for the elastic equation

The APF diagrams are obtained by inserting the necessary values. For the rigid equation the necessary values are $K_p = 2$, $T_i = 12$ s, $T_a = 6$ s and $T_w = 0.49$ s. The APF diagram for this equation is shown in figure 5.3.

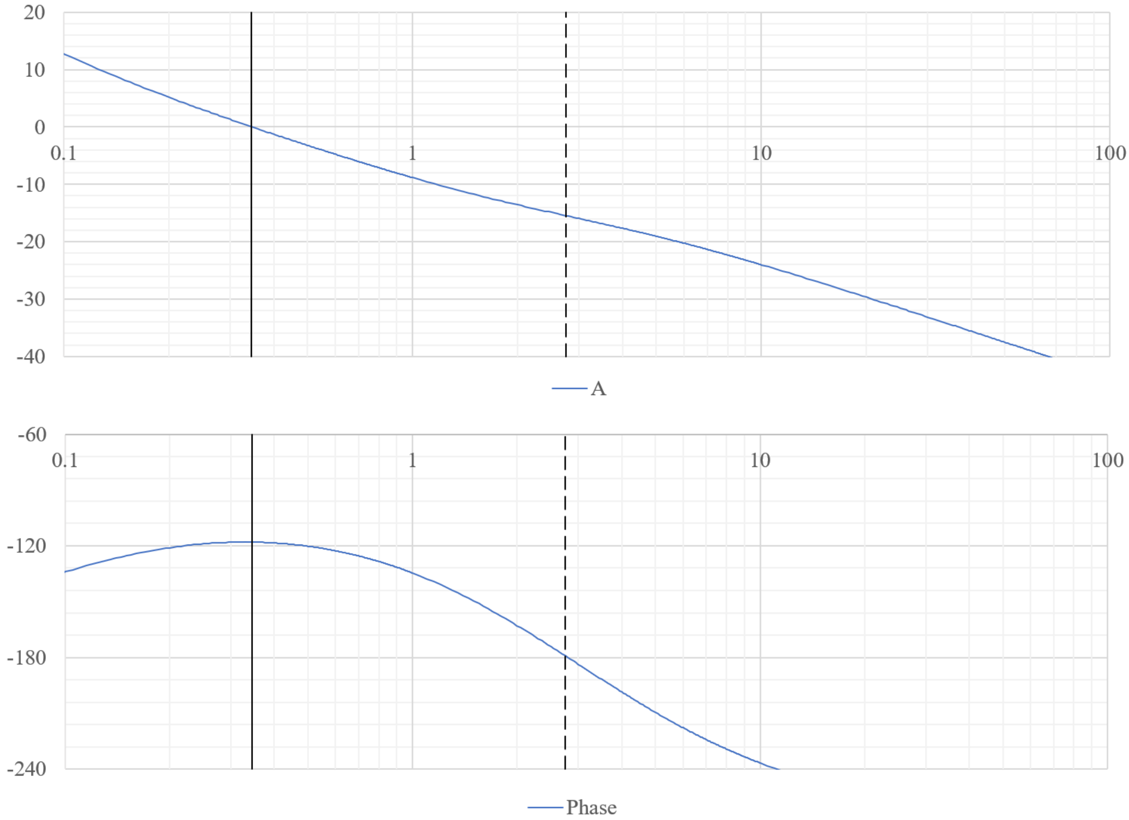


Figure 5.3: APF diagram for the rigid equation

In the rigid APF diagram, as shown in figure 5.3, the amplitude curve crosses the 0 dB line where the phase curve is at -118°, which is marked with the solid black line in the diagram. According to the Nyquist criterion, as given in chapter 3.2 and equation 3.15, the system is stable since -118 ° is above the -180° line. However, it is often required that the phase margin, ψ, should be larger than 45° and that the gain margin, Δk, should be larger than 6 dB. The phase margin is the difference between the phase at the cross frequency point and the -180° line. Again, this is marked in the diagram with the solid line. Here the phase margin is calculated in equation 5.1:

$$\psi = (-118°) - (-180°) = 62° \tag{5.1}$$

which means that the phase margin is larger than 45° as desired.

The gain margin, Δk, is the difference between the 0 dB line and the amplitude curve where the phase curve crosses the -180° line, which is represented with a dashed line in the diagram. This gives the gain margin in equation 5.2:

$$\Delta k = 0dB - (-15.5dB) = 15.5dB \tag{5.2}$$

which is larger than 6 dB as desired.

The phase margin and gain margin are requirements set to ensure stability. The rigid hydropower system has fulfilled both requirements as well as the Nyquist criterion, which suggests that the system is stable.

The elastic equation is more complex than the rigid one, and as mentioned it must be transformed in order to be solved, as shown in chapter 3.2. This transformation is implemented in the spreadsheet where the APF diagram is made. The necessary values are therefore $K_p = 2$, $T_i = 12$ s, $T_a = 6$ s, $h_w = T_w/T_r = 0.49$ s/0.058 s = 8.45, and L/a = (34.8 m)/(1200 m/s^2) = 0.029 s^2. This makes the APF diagram for the elastic equation look like figure 5.4.

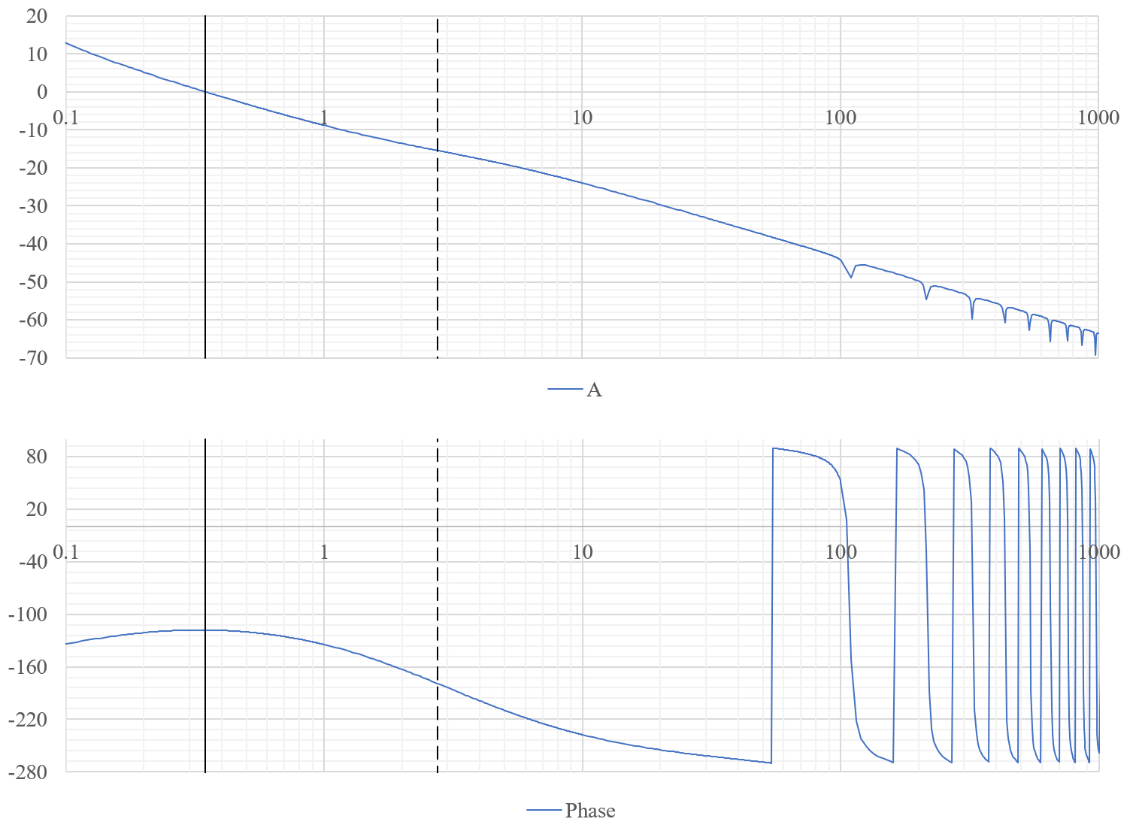


Figure 5.4: APF diagram for the elastic equation

In the elastic APF diagram the cross frequency point occurs at -118°, as the solid black line show. This is identical to the rigid diagram. This means that, according to the Nyquist criterion, the system is stable when considering the elastic properties as well. This gives a phase margin, ψ, of 62°, the same as for the rigid system.

The gain margin, Δk, for the elastic equation is given in equation 5.3:

$$\Delta k = 0dB - (-15.4dB) = 15.4dB \tag{5.3}$$

which is larger than 6 dB as desired.

The phase margin and the gain margin are very similar for both APF diagrams, and the values are within the requirements. The Nyquist criterion is also fulfilled in both cases. A significant difference between the diagrams is the oscillation in both the amplitude and phase in the elastic diagram. It is desirable that oscillations are far to the right of the cross frequency to avoid instability in the system.

5.2 Closing time

As explained in chapter 2.2, the closing time, T_c, is calculated to find the shortest amount of time the guide vanes can use to close without creating a harmful pressure increase in the water channel. The value of T_c, which was calculated in chapter 4, equation 4.4, is 1.42 s. The actual closing time is longer than the calculated minimum closing time, because the speed of the servomotor is limited by throttle valves on the oil pressure system. They limit the oil supply, and therefore slow the servomotor down. This is done to prevent damage to the mechanical and hydraulic equipment as well as personal injury. The real closing time is shown in figure 5.5.

Figure 5.5 shows the angle of the guide vanes as a function of time. The test started after approximately 17 s, which gives the real closing time of 3.1 s. According to the measurements, the guide vanes do not close fully, as seen in figure 5.5. This is probably because of a calibration error or a measuring error. In reality the guide vanes are fully closed.

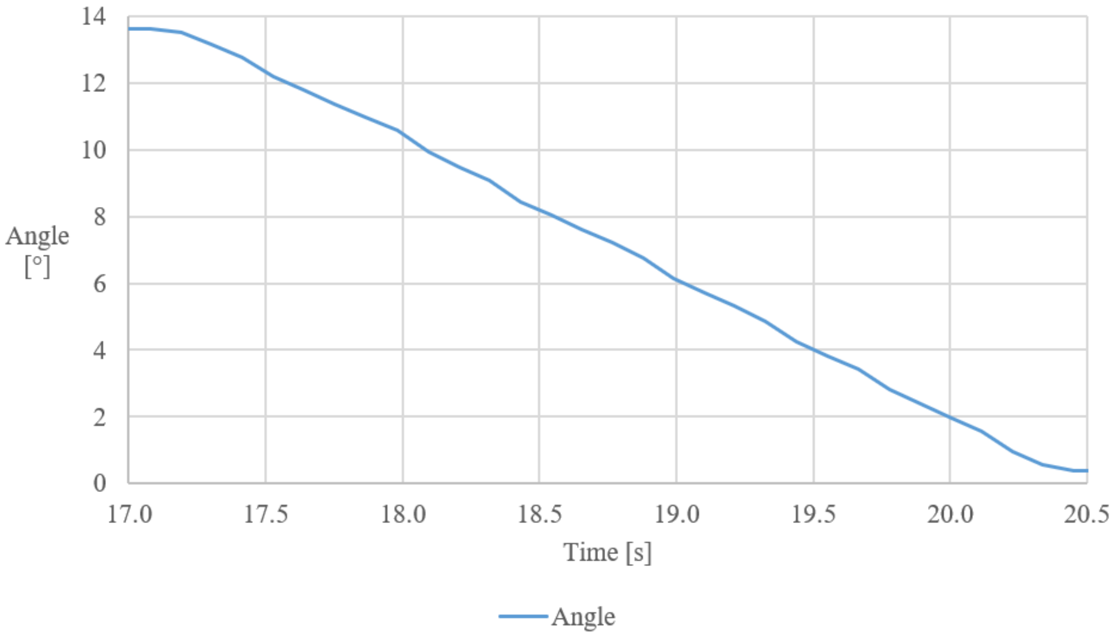


Figure 5.5: Measured closing time

The throttle valves can be adjusted to reduce the closing time, but with a higher risk of damaging mechanical or hydraulic equipment, such as oil hoses or mechanical equipment connected to the guide vanes. Short closing time, which gives a high speed of the servomotor, does also increase the risk of harming people nearby. The maximum stroke of the servomotor is 60 mm. With a closing time of 3.1 s the servo motor speed, u_t, is as given in equation 5.4:

$$u_t = \frac{60mm}{3.1s} = 19.4mm/s \tag{5.4}$$

With a closing time of 1.42 s, the servomotor speed, u_c, becomes:

$$u_c = \frac{60mm}{1.42s} = 42.3mm/s \tag{5.5}$$

which is more than twice as fast as the throttled servomotor. It is important to remember that the turbine unit in the Waterpower Laboratory is robust and is dimensioned for experimental tests, so acceleration of the rotating masses with a closing time of 3.1 s will not cause any damage. Also, the equipment is handled by people, so the closing time is set to this value to lower the risk of personal injury from the servomotor stroke.

5.3 Frequency step test

The next test is a frequency step test. According to Statnett, the frequency step test consists of steps up and down in frequency, to see if the turbine unit behaves as desired. As mentioned in the previous chapter, the turbine unit shall increase the power production with a power contribution, ΔP, according to equation 4.10:

$$\Delta P = \frac{\Delta n}{n_{nom}} \frac{P_{max}}{b_p}$$

For the tests in this project the steps were done in rotational speed instead of frequency, as seen in figure 5.6. The test will show the same result as for frequency steps. The nominal rotational speed, n_0, is 342 rpm, which is equivalent to the nominal frequency of 50 Hz. The test includes frequency steps of 0.1 Hz and 0.5 Hz up and down, which is equivalent to steps of 0.7 rpm and 3.4 rpm up and down. However, the steps are set to whole numbers, which gives steps of 1 rpm and 4 rpm up and down. The different steps are marked with solid black lines in the diagram.

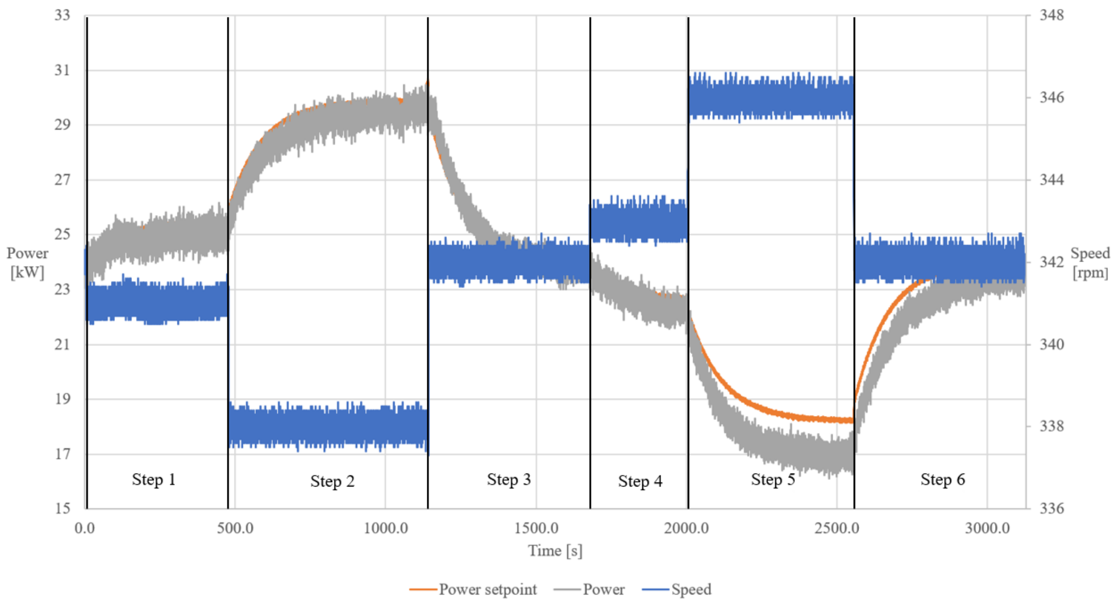


Figure 5.6: Power output based on frequency change

The first step is from the nominal rotational speed of 342 rpm to 341 rpm. When the speed drops the power output starts to increase, as desired. After short time the power output is stable at about 25 kW. Inserting these values in equation 4.10 gives the required power contribution ΔP_1:

$$\Delta P_1 = \frac{1rpm}{342rpm} \times \frac{30kW}{0.06} = 1.46kW \tag{5.6}$$

The power curve in the first step shows a power jump from the nominal power of 24 kW to 25 kW. This gives a ΔP_1 of 1 kW, which is lower than required.

In the second step the rotational speed is reduced to 338 rpm. The power increases fast, and stabilizes after a few minutes. This stabilisation indicates that the power deviation from the grid is decreased, and the power demand is covered. Inserting into equation 4.10 gives:

$$\Delta P_2 = \frac{4rpm}{342rpm} \times \frac{30kW}{0.06} = 5.85kW \tag{5.7}$$

The power stabilizes at 29.5 kW, which gives a change of 5.5 kW, which is also a little lower than the requirement.

The third step is back to the nominal speed of 342 rpm. The power curve decreases as it should be. After some time, when the power is approaching the correct value, the fourth step happens. When the speed jumps to 343 rpm, the power decreases again, and decreases even more when the speed is changed for the fifth time, to 346 rpm.

The power curves for the fourth and fifth step gives $\Delta P_4 = 1.46$ and $\Delta P_5 = 5.85$ kW, respectively. The power curve at step 4 stabilizes at 22.6 kW and for step 5 at 17 kW. The differences between these and the reference power of 24 kW are 1.4 kW and 7 kW. The measured power contribution from step 4 is a little low, but from step 5 it is higher than the requirement.

According to "NVF" and the method described in chapter 4, the governor at the Waterpower Laboratory responds to the frequency step test by changing the power production, which is desired, but the power contributions are mostly lower than required. Also, there is a deviation between the power and power setpoint curves in the fifth step. This is due to the deviation in the P/Y diagram.

5.3.1 P/Y diagram

The orange curve in figure 5.6 describes the power setpoint, which is calculated based on the P/Y (Power/Guide vane opening) diagram. Figure 5.7 show the P/Y diagram for the frequency step test. The blue line in the diagram, the P/Y curve, is a standard estimate used in the commissioning of hydropower plants, which will not be discussed further in this thesis. The P/Y curve converts the power setpoint to guide vane opening setpoint.

The blue line shows the P/Y curve that was used to find the guide vane opening setpoint and power setpoint for the frequency step test, shown in figure 5.6. After the frequency step test was finished, when the deviation between the power and the power setpoint was discovered, the power was measured at five different guide vane angles, to compare the P/Y curve to the actual behaviour. These five measurements are shown in table 5.2, and marked with orange points in figure 5.7.

Table 5.2: Reference points for the P/Y curve

Angle [°]	Opening [%]	P [kW]
4.04	31.1	8.8
6.02	46.3	13.5
8.00	61.5	18.4
10.06	77.4	23.6
11.99	92.2	27.5

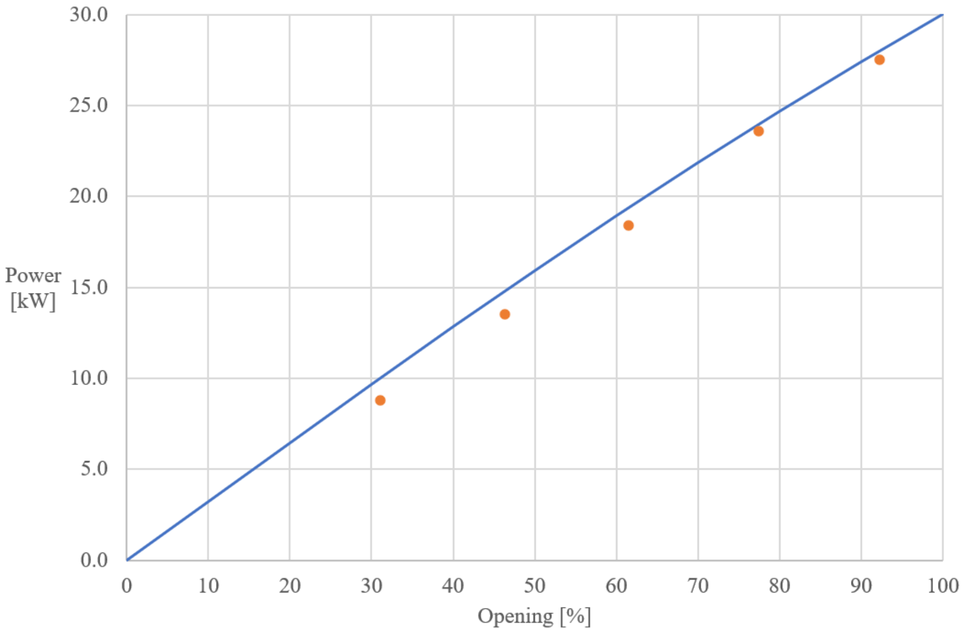


Figure 5.7: P/Y curve

There is a significant deviation between the curve and the points where the guide vane openings are at 31.1 %, 46.3 % and 61.5 %. The reason for the deviation is that the estimated P/Y curve is not accurate enough compared to the measured value. In practice the result is that the P/Y curve gives the wrong guide vane opening, which gives the wrong power. This inaccuracy was uncovered after the frequency step test, and was corrected before the isolated grid test.

5.4 Isolated grid test

The most demanding test when commissioning a new hydropower plant is the isolated grid test. This test actually investigates the governing quality. As explained earlier, the isolated grid test shows the behaviour of the hydropower system when governed based on the frequency signal. When the frequency drops, the governor opens the guide vanes to increase the produced power. The first 266 seconds of the isolated grid test is illustrated in figure 5.8, with an acceleration time for the rotating masses, T_a, of 60 s. T_a is set to such a high value to avoid sharp changes in the rotational speed.

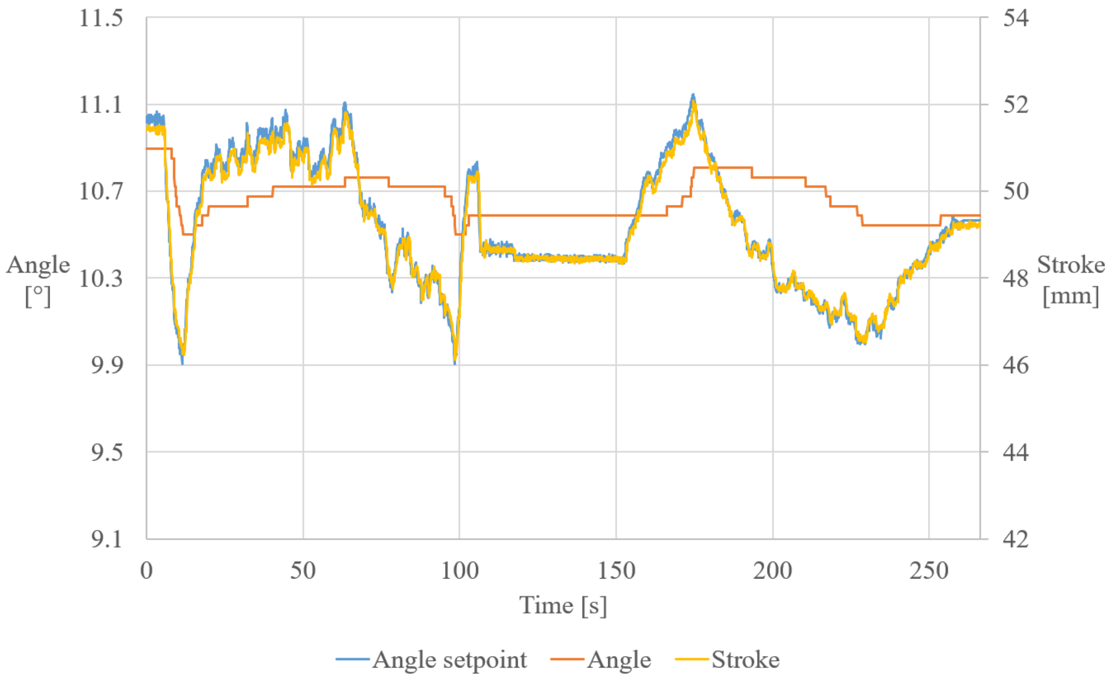


Figure 5.8: Guide vane angle compared to servomotor stroke

From figure 5.8 it is clear that the stroke follows the guide vane angle setpoint precisely, but the guide vane angle does not. This is due to backlash somewhere in the mechanical parts between the servomotor and the guide vanes. The guide vane angle does not follow the servomotor stroke as it should, so the guide vane angle does not reach the desired value. This creates a deviation between the power demanded and the power produced, which again creates a frequency deviation. The frequency deviation causes the governor to try to change the produced power, which again causes a change in the guide vane angle setpoint that the guide vanes are not able to follow. An oscillating deviation that does not disappear is the result, which means that the governor will always be in motion and that the hydropower system will not become stable. Figure 5.9 illustrates the guide vane angle and power as a function of time, showing that the power follows the guide vane opening, which testifies that the problem is backlash between the servomotor and the guide vanes.

Figure 5.10 shows the rotational speed, speed setpoint, simulated load and power as a function of time. The rotational speed of the turbine unit changes quite rapidly and with large variations. As described in the frequency step test, it is desired that the power increases when the frequency drops, to try to increase the frequency back to the nominal value. This means that it is desired that the power contribution, ΔP, is 0, because this means that the frequency is stable at the nominal value. Since the power does not stabilize, the frequency, or in this case rotational speed, is not stabilizing.

The speed variations in figure 5.10 show far from stable governing. Keeping the APF diagrams in mind, showing stable systems for both the rigid and the elastic equations, this result is unexpected. According to the requirements the rotational speed should be in the range of 341.3 - 342.7 rpm, which is equivalent to the frequency range of 49.9 Hz to 50.1 Hz. Figure 5.10 shows the first 266 s of the isolated grid test, where the speed varies from 336 rpm to 352 rpm, which is almost 1.8 % under and more than 2.9 % over the nominal speed. This is equal to a frequency range of 49.1 to 51.45 Hz. According to the requirements the speed can deviate from the nominal frequency range of 49.9 Hz to 50.1 Hz for short periods of time. In this test the variation is very large and happens quite often, and can therefore probably not be tolerated.

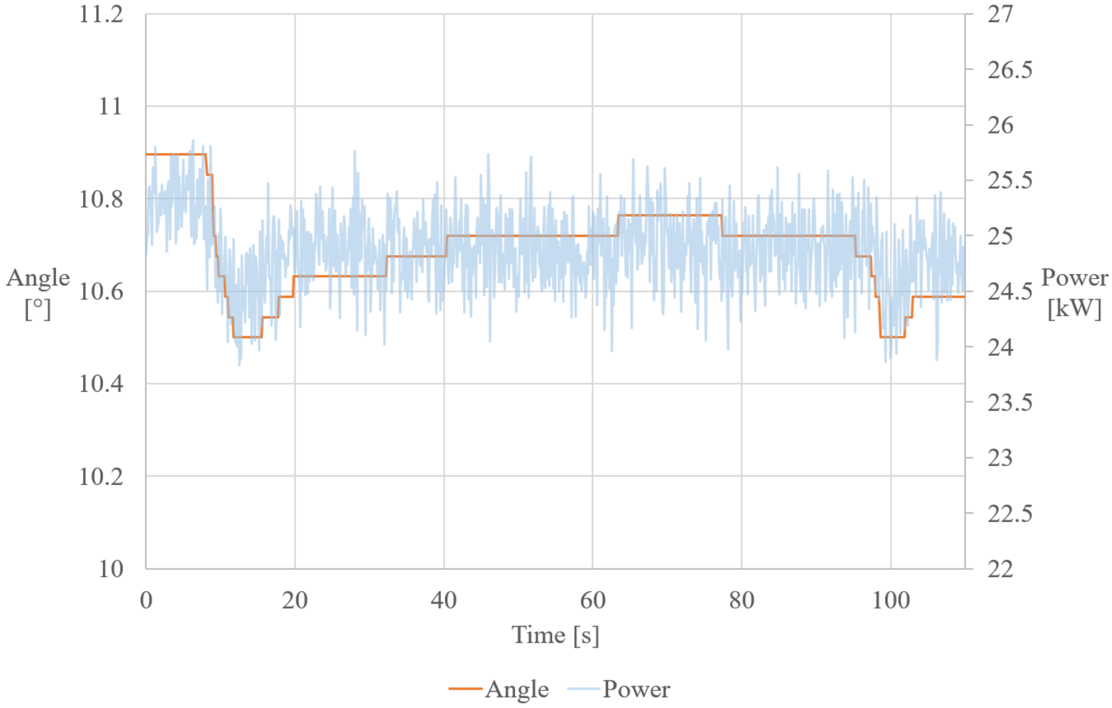


Figure 5.9: Power compared to guide vane angle

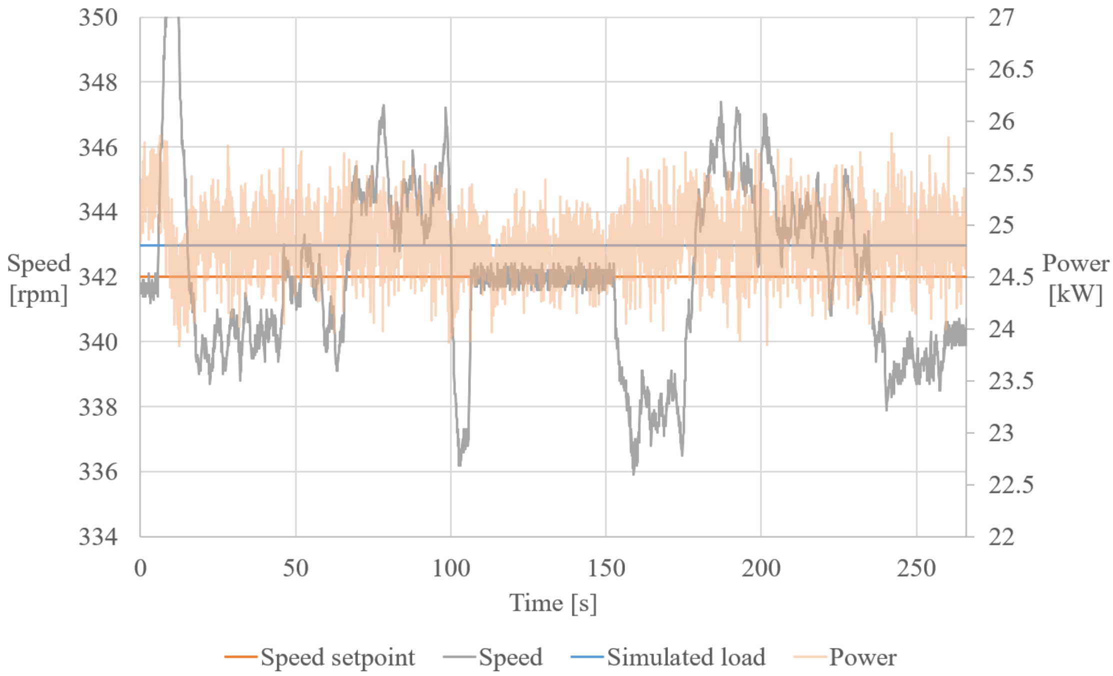


Figure 5.10: Variation in power and rotational speed in the isolated grid test

5.4.1 Backlash

Backlash creates problems because the guide vanes will never find the desired angle, which will result in a deviation in produced power and frequency, which again will cause the governor to try to correct the error. The problem is that the error will never disappear because of the backlash. For smaller load changes the guide vanes might not move at all. This is shown in figure 5.11.

Figure 5.11 illustrates how the backlash influence the guide vane angle at small steps. This section is from a later moment of the isolated grid test. The yellow line illustrates how the

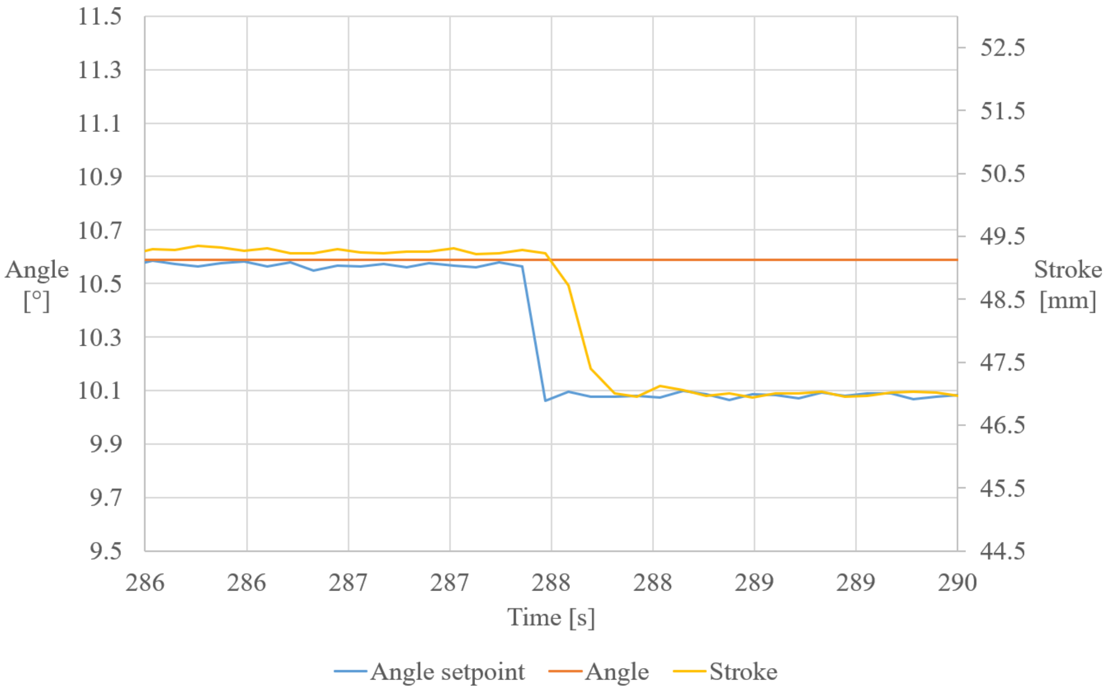


Figure 5.11: Behaviour of the guide vane angle for a small step

servomotor reduces the stroke when the guide vane angle setpoint is changed. The servomotor motion reacts about 0.1 s after the signal is given, but reaches the desired stroke slightly slower than the angle set point. Either way, the time constant for the servomotor is about 0.3 s, which is almost as fast as these tests are able to measure, and makes the servomotor reaction sufficient for this system.

The blue line shows that the guide vane angle setpoint is reduced from about 10.6° to approximately 10.1°. However, the actual guide vane angle does not change at all. The servomotor is moving, but the backlash hinders the guide vanes from moving. Figure 5.11 shows a backlash of approximately 2 mm, but for other tests it was observed to be more than 3 mm, which will contribute with a significant deviation in power and frequency.

The best way to correct the backlash properly is probably to remove it mechanically, either by changing the mechanical links between the servomotor and the guide vanes, or modifying the existing solution.

Chapter 6

Conclusion

The aim of this master thesis has been to implement a turbine governor on the Francis turbine test rig at the Waterpower Laboratory at NTNU. The governor parameters were found, and experiments were conducted to ensure high quality governing, so the test rig can be operated as a turbine in an actual power plant is operated.

The reflection time, closing time, acceleration time for the rotating masses and acceleration time for the water masses were all calculated based on measurements from the Waterpower Laboratory. Most of these time constants were used to make APF diagrams for both the rigid and the elastic equations. The APF diagrams were used to find the suitable PID parameters to ensure stable governing.

When the calculations and APF diagrams showed that the hydropower system in the Waterpower Laboratory is stable, experimental tests were conducted. These are tests required for real hydropower plants, and are supposed to show if the turbine governor is able to do the most important job, to keep the frequency stable. Several tests were conducted, and the results were verified according to requirements. Even though the APF diagrams showed a stable system, the results from the tests showed that the governor was not able to stabilize the frequency in reasonable time.

After several tests it was discovered backlash in the mechanical parts between the servomotor and the guide vanes. The isolated grid test showed that the guide vanes did not behave as they should. When the guide vane angle setpoint was changed, the servomotor reacted as it should, meanwhile the guide vanes did not follow. For small steps, as shown in figure 5.11, the guide vanes did not follow at all. Since the guide vane opening controls the water flow rate through the turbine and such controls the power produced, the deviation in guide vane opening results in deviation in produced power, and hence also frequency.

The findings from the different tests testify that the governor behaves as desired, but that there is backlash somewhere between the servomotor and the guide vanes, which hinders stable governing. The backlash results in a deviation in the guide vane angle that is difficult to remove without changing or improving some of the mechanical parts. As of today the turbine governor is not able to ensure stable frequency governing, due to the backlash challenges. For the governor to be able to control the Francis turbine test rig according to the requirements, the best way is probably to remove the backlash in the mechanical components.

Chapter 7

Further work

At this point in time the Francis turbine test rig cannot be operated as a turbine in an actual power plant, due to backlash in the mechanical parts between the servomotor and the guide vanes. Even though the test results show that the governor most likely works as desired, it cannot be concluded that the governor is able to maintain stable operation without removing the backlash. Before conducting more tests regarding the governor, the backlash problem should be solved.

Since the backlash is a mechanical problem in one or several parts between the servomotor and the guide vanes, it is necessary to do a visual investigation of the equipment. When the source of the problem is found, it must be figured out if the part, or parts, must be replaced, or if they can be improved, so the backlash is removed for a long time. If improving or changing the parts is impossible or if it does not remove or improve the backlash sufficiently, another way is to implement the backlash in the software. This is probably a complex process and may in worst case cause instability.

When the necessary actions to improve, and hopefully remove, the backlash are done, the tests conducted in this thesis must be repeated. If the isolated grid test shows that the governor is able to stabilize the frequency in reasonable time, load rejection tests can be conducted.

With the current setup in the Waterpower Laboratory the time constant for the water masses, T_w, cannot be changed. For tests regarding existing power plants, it could be useful to adjust the T_w in the laboratory to match the real T_w for a specific power plant, to obtain very realistic test results. To change the value of T_w would involve changing the length of the water channel. This could be done by connecting more of the pipes and/or installing valves to lead the water a longer distance from the reservoir to the turbine. It could also be interesting to investigate U-tube oscillations in the Waterpower Laboratory with a longer T_w.

At a later time the Waterpower Laboratory and the Smart Grid Laboratory will be connected, so the test rig will be exposed to realistic grid conditions. This process will probably come with new tests and challenges which must be solved.

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Appendix A

Transfer functions in the hydropower system

Symbols

y	Guide vane opening	[%]
t	Time	[s]
K_p	Proportional constant	[-]
n	Rotational speed	[rpm]
n_0	Nominal rotational speed	[rpm]
T_i	Integral time	[s]
L	Length	[m]
g	Gravitational acceleration	[m/s^2]
A	Cross section area	[m^2]
Q	Water flow rate	[m^3/s]
H	Head	[m]
P	Power	[W]
η	Turbine efficiency	[%]
ρ	Water density	[kg/m^3]
T_w	Acceleration time for the water masses	[s]
ω	Angular velocity	[s^{-1}]
J	Polar moment of inertia	[kg m^2]
T_a	Acceleration time for the rotating masses	[s]

The main components in a hydropower system is the governor, the water channels and turbine and the rotating masses, as stated in chapter 3.3. To find the transfer functions the system is simplified by assuming flow without losses and rigid water channels. These functions are derived based on chapter 5.7 in Nielsen's "Dynamisk dimensjonering av vannkraftverk" ([11]).

Transfer function for the governor

The transfer function for the governor is based on a PI controller with the following equation:

$$\frac{dy}{dt} = K_p \frac{dn}{dt} + \frac{K_p}{T_i}(n_0 - n) \tag{7.1}$$

Laplace transforming equation 7.1 gives equation 7.2:

$$\Delta ys = K_p \Delta ns + \frac{K_p}{T_i}\Delta n \tag{7.2}$$

This leads to equation 7.3:

$$\frac{\Delta y}{\Delta n}s = K_p s + \frac{K_p}{T_i} \tag{7.3}$$

which leads to the transfer function for the governor in equation 7.4:

$$\frac{\Delta y}{\Delta n} = K_p \frac{(1 + T_i s)}{T_i s} \tag{7.4}$$

Transfer function for the water channel and the turbine

The equation for the water channel is:

$$\frac{L}{gA}\frac{dQ}{dt} = \Delta H \tag{7.5}$$

The equation for the turbine is:

$$Q = y_n \frac{Q_n}{\sqrt{2gH_n}}\sqrt{2gH} = y_n k\sqrt{H} \tag{7.6}$$

where y_n, Q_n and H_n are the guide vane opening, flow rate and head at a given operating point. Laplace transforming equation 7.5 gives equation 7.7.

$$\frac{L}{gA} Qs = \Delta H \tag{7.7}$$

Introducing ratios $h = \Delta H/H_0$ and $q = \Delta Q/Q_0$:

$$h = \frac{Q_0}{gH_0}\frac{L}{gA} qs \tag{7.8}$$

The derivative of equation 7.6 is:

$$dQ = y_n k \frac{1}{2}\frac{1}{\sqrt{H}} dH \tag{7.9}$$

Linearizing equation 7.9 about the point Q_0, H_0 gives equation 7.10:

$$\Delta Q = y_n k \frac{1}{2}\frac{1}{\sqrt{H_0}} \Delta H \tag{7.10}$$

then, multiplying with H_0 and introducing $Q_0 = y_n\ k\ {H_0}^{1/2}$ gives equation 7.11.

$$H_0 \Delta Q = Q_0 \frac{1}{2} \Delta H \tag{7.11}$$

Using ratios h and q again, gives equation 7.12.

$$q = \frac{1}{2} h \tag{7.12}$$

The hydraulic power is:

$$P_h = \eta \rho g Q H \tag{7.13}$$

Differentiating equation 7.13, assuming constant $\eta$and introducing ratios q, h and $p_h = \Delta P_h/P_0$ gives equation 7.14.

$$p_h = q + h \tag{7.14}$$

The transfer function for the guide vane opening and power is shown in figure 7.1. The equation for the acceleration time for the water masses is given in equation 7.15:

$$T_w = \frac{Q_0}{gH_0}\frac{L}{A} \tag{7.15}$$

Utilizing the rules for reduction of block diagrams and using equation 7.15 gives the transfer function in equation 7.16 and figure 7.2.

$$\frac{P_h}{q} = \frac{1 - T_w s}{1 + 0.5 T_w s} \tag{7.16}$$

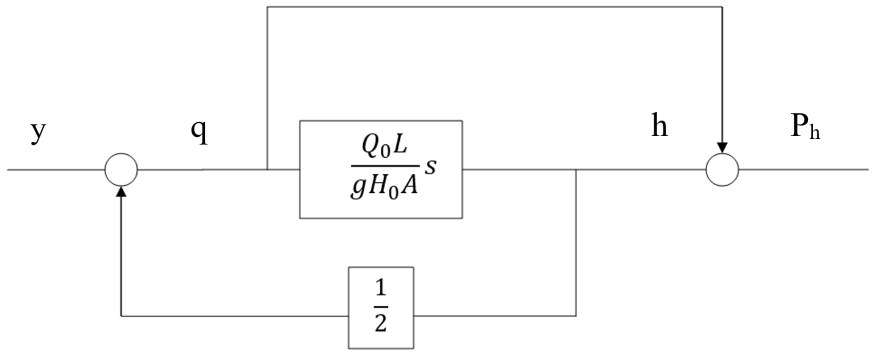


Figure 7.1: Block diagram for the relationship between power and guide vane opening

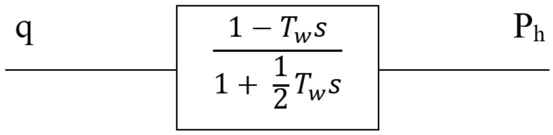


Figure 7.2: Reduced block diagram with the transfer function for P_h and q

Transfer function for the rotating masses

The transfer function for the rotating masses is found from equation 7.17:

$$\omega J\frac{d\omega}{dt} = \Delta P \tag{7.17}$$

Laplace transforming equation 7.17 gives equation 7.18.

$$\omega_0 J\Delta\omega s = \Delta P \tag{7.18}$$

Introducing ratios $\mu = \Delta\omega/\omega_0$ and $\nu = \Delta P/P_0$ and inserting equation 7.18 leads to equation 7.19:

$$\mu = \frac{\Delta\omega}{\omega_0} = \frac{1}{\frac{J\omega_0^2 s}{P_0}}\frac{\Delta P}{P_0} = \frac{1}{\frac{J\omega_0^2 s}{P_0}}\nu \tag{7.19}$$

The acceleration time for the rotating masses is given in equation 7.20.

$$T_a = \frac{J\omega_0^2}{P_0} \tag{7.20}$$

Combining equations 7.20 and 7.19 gives the transfer function between μ and ν in equation 7.21:

$$\frac{\mu}{\nu} = \frac{1}{T_a s} \tag{7.21}$$

The block diagram for the rotating masses is shown in figure 7.3.

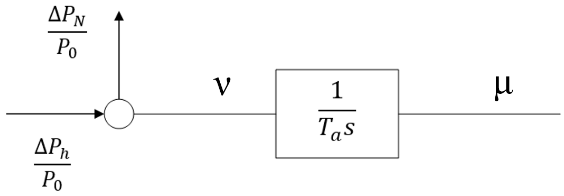


Figure 7.3: Block diagram for the rotating masses

The transfer functions for the governor, water channel and turbine, and the rotating masses can be put into one block diagram, as seen in figure 7.4.

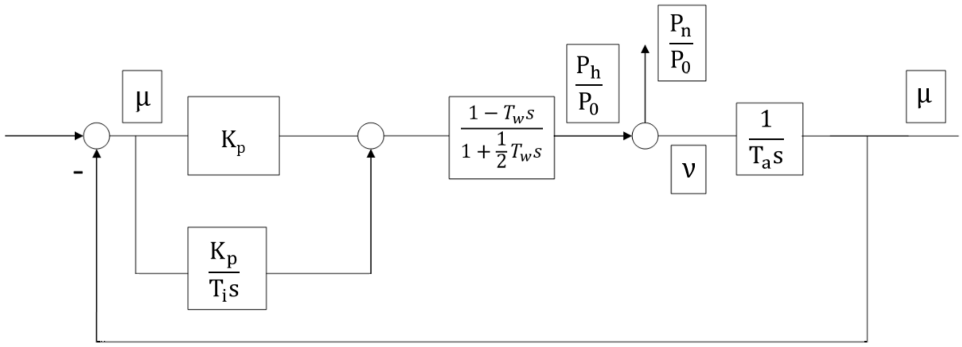


Figure 7.4: Block diagram for the total system

These transfer functions for the rigid system with a PI controller are combined and reduced to equation 7.22:

$$A(s) = \frac{K_p}{T_i T_a} \frac{(1+T_i s)}{s^2} \frac{(1-T_w s)}{(1+0.5T_w s)} \tag{7.22}$$

which gives the block diagram as seen in figure 7.5.

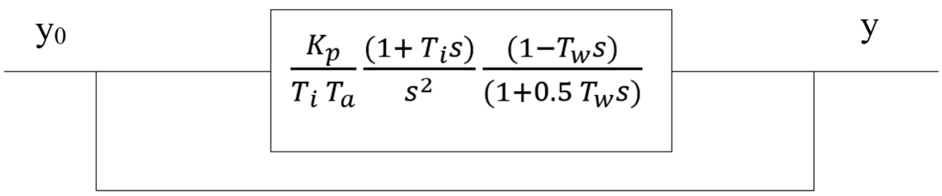


Figure 7.5: Reduced block diagram for the total system

Appendix B

Block diagram

A dynamic process can be investigated by observing the response from a disturbance. Linearizing the system around a point of operation and then Laplace transforming gives the system's transfer function described in the frequency plane. This is called frequency analysis, which shows whether a system is stable or not [11].

A hydropower system is a system with feedback as shown in figure 7.6.

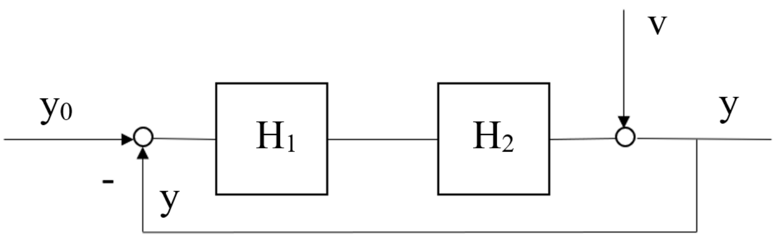
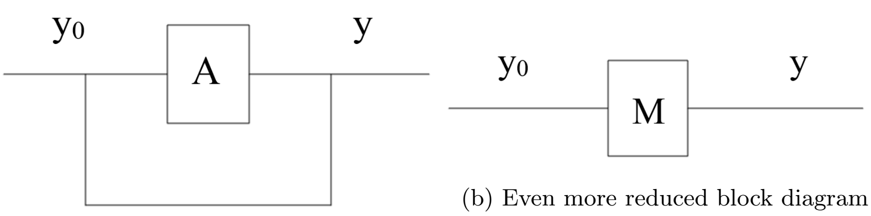


Figure 7.6: System with feedback

The guide vane opening degree, y, is measured and compared to the reference guide vane opening, y_0. The load is changed based on the deviation between y and y_0. It is desired that the y/y_0 ratio is small. The disturbance, v, causes the governor to change the power production. The governor's transfer function must be chosen such that the disturbance has as little influence on y as possible [11]. The block diagram in figure 7.6 can be reduced, so the different transfer functions are reduced to one transfer function, as shown in figure 7.7a.



(a) Reduced block diagram

(b) Even more reduced block diagram

Figure 7.7: Reduced block diagrams

A is called an open loop transfer function, which describes the system without feedback [11]. Figure 7.7a can be further reduced, as shown in figure 7.7b.

M describes how well y follows y_0 when feedback is introduced. The rules for reduction of block diagrams say:

$$M = \frac{1}{1+A} \times A \tag{7.23}$$

This means that if M = 1, the guide vane opening follows the reference perfectly, $y = y_0$ [11].

N, given in equation 7.24, is the deviation relationship that says how useful the feedback is.

$$N = \frac{1}{1 + A} \tag{7.24}$$

The equation above shows that if A » 1, N is approximately 1/A which gives M = 1 as desired. If A « 1, N is approximately 1, which means that the feedback has no function [11].

Appendix C

Risk assessment

NTNU	Hazardous activity identification process	Prepared by	Number	Date
HSE		HSE section	HMSRV2601E	09.01.2013
		Approved by		Replaces
		The Rector		01.12.2006

Unit: *(Department)* Department of Energy and Process Engineering

Line manager: Terese Løvås

Participants in the identification process (including their function)**:** Pål-Tore Storli (supervisor), Truls Edvardsen Aarønes (co-supervisor), Lena Rostad (student).

Short description of the main activity/main process: Master project for student Lena Rostad. Implementation of a turbine governor on the Francis test rig at The Waterpower Laboratory.

Is the project work purely theoretical? (YES/NO): NO

Signatures: *Responsible supervisor:* *Student:* Lena Rostad

ID nr.	Activity/process	Responsible person	Existing documentation	Existing safety measures	Laws, regulations etc.	Comment
1	Test of servomotor connected to a oil pressure system	Truls Edvardsen Aarønes	Safety course.	Safety glasses		
2	Test of turbine governor on the Francis rig	Truls Edvardsen Aarønes	Safety course. Risk assessment report for the Francis turbine test rig.			

NTNU	Risk assessment	Prepared by	Number	Date	
		HSE section	HMSRV2603E	04.02.2011	
HSE		Approved by		Replaces	
		The Rector		01.12.2006	

Unit: *(Department)* Department of Energy and Process Engineering

Line manager: Terese Løvås

Participants in the identification process (including their function)**:** Pål-Tore Storli (supervisor), Truls Edvardsen Aarønes (co-supervisor), Lena Rostad (student).

Short description of the main activity/main process: Master project for student Lena Rostad. Implementation of a turbine governor on the Francis test rig at The Waterpower Laboratory.

Signatures: *Responsible supervisor:* Pål-Tore Storli *Student:* Lena Rostad

Activity from the identification process form	**Potential undesirable incident/strain**	**Likelihood:**	**Consequence:**			**Risk Value (human)**	**Comments/status Suggested measures**
		Likelihood (1-5)	Human (A-E)	Environment (A-E)	Economy/ material (A-E)		
1	High pressure hydraulic leakage causing damage to humans and/or materials	2	D	B	B	D2	Use of safety glasses Leakage test Double check of all fittings Oil blankets
1	Electric energy causing damage to humans and/or materials	3	B	A	A	B3	Fixing components to table Use of knife disconnect terminal block
1	Danger of mechanical damage to humans and/or materials	2	C	A	A	C2	Fixing servomotor to table Keeping area tidy
2	As above						
2	Electric energy causing damage to humans and/or materials	1	B	A	D	B1	Energy to and from generator Careful testing
2	Rotating energy causing damage to humans and/or materials	2	C	A	B	C2	Keep distance
2	Energy from high water flow and water pressure causing damage to humans and/or materials	2	B	B	C	B2	Controlled design of the experiment Good knowledge of running and stopping the rig

Likelihood, e.g.:
1. *Minimal*
2. *Low*
3. *Medium*
4. *High*
5. *Very high*

Consequence, e.g.:
A. *Safe*
B. *Relatively safe*
C. *Dangerous*
D. *Critical*
E. *Very critical*

Risk value (each one to be estimated separately):
Human = Likelihood x Human Consequence
Environmental = Likelihood x Environmental consequence
Financial/material = Likelihood x Consequence for Economy/materiel

Potential undesirable incident/strain
Identify possible incidents and conditions that may lead to situations that pose a hazard to people, the environment and any materiel/equipment involved.

Criteria for the assessment of likelihood and consequence in relation to fieldwork
Each activity is assessed according to a worst-case scenario. Likelihood and consequence are to be assessed separately for each potential undesirable incident. Before starting on the quantification, the participants should agree what they understand by the assessment criteria:

Likelihood

Minimal 1	Low 2	Medium 3	High 4	Very high 5
Once every 50 years or less	Once every 10 years or less	Once a year or less	Once a month or less	Once a week

Consequence

Grading	Human	Environment	Financial/material
E Very critical	May produce fatality/ies	Very prolonged, non-reversible damage	Shutdown of work >1 year.
D Critical	Permanent injury, may produce serious serious health damage/sickness	Prolonged damage. Long recovery time.	Shutdown of work 0.5-1 year.
C Dangerous	Serious personal injury	Minor damage. Long recovery time	Shutdown of work < 1 month
B Relatively safe	Injury that requires medical treatment	Minor damage. Short recovery time	Shutdown of work < 1week
A Safe	Injury that requires first aid	Insignificant damage. Short recovery time	Shutdown of work < 1day

The unit makes its own decision as to whether opting to fill in or not consequences for economy/materiel, for example if the unit is going to use particularly valuable equipment. It is up to the individual unit to choose the assessment criteria for this column.

Risk = Likelihood x Consequence
Please calculate the risk value for “Human”, “Environment” and, if chosen, “Economy/materiel”, separately.

About the column ”Comments/status, suggested preventative and corrective measures”:
Measures can impact on both likelihood and consequences. Prioritise measures that can prevent the incident from occurring; in other words, likelihood-reducing measures are to be prioritised above greater emergency preparedness, i.e. consequence-reducing measures.

NTNU	Risk matrix	Prepared by	Number	Date
		HSE section	HMSRV2604	8 March 2010
		Approved by		Replaces
HSE		The Rector		9 February 2010

MATRIX FOR RISK ASSESSMENTS at NTNU

CONSEQUENCE	Extremely serious	E1	E2	E3	E4	E5
	Serious	D1	D2	D3	D4	D5
	Moderate	C1	C2	C3	C4	C5
	Minor	B1	B2	B3	B4	B5
	Not significant	A1	A2	A3	A4	A5
		Very low	Low	Medium	High	Very high
		LIKELIHOOD				

Principle for acceptance criteria. Explanation of the colours used in the risk matrix.

Colour	Description
Red	Unacceptable risk. Measures must be taken to reduce the risk.
Yellow	Assessment range. Measures must be considered.
Green	Acceptable risk Measures can be considered based on other considerations.

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