NTNU – Trondheim
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Frekvens- og tidsanalyse av vannturbin- og kraftsystemmodeller

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Submission date: June 2015
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Frequency and Time Series Analysis of Hydro Turbine and Power System Models

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Abstract

In this report a hydro turbine model and a power system model have been developed. The goal is to develop models capable of preforming in a frequency analysis of the Nordic synchronous area and that these models can help to investigate floating frequencies in the Nordic power system.

The hydro model has non-linear penstock and turbine dynamics and it is governed by PID-governor. The power system is modelled as a one-bus system, based on the mechanical swing equations. Both the turbine model and the power system model have been tested with a time series analysis and a frequency analysis. Particularly the effect of load damping and inertia has been tested.

The time series analysis shows that the hydro model can work as a substitute for existing models, both in a standalone comparison to other hydro models, and in a larger power system model. The time series does find that the hydro turbine is less predictable in behaviour. As the model is non-linear, instabilities were found in some cases, when trying to find the critical stable operating point.

The power system model was also tested to discover how it behaved in a time series analysis. It was scaled to an equivalent of the Nordic system and analysed to find the response to disturbances in the system. The system has been analysed to show the differences in response to a fully loaded system and a sparsely loaded system. The test show a significant deterioration in the system stability when less inertia was available in the system.

A frequency analysis was conducted on both the turbine model and the power system model to test the effect of inertia and load damping. The test on the hydro turbine model confirmed that it was unstable for a selection of system parameters. Stable simulations show that the models tend to resonate at frequencies between 0.004 Hz and 0.022 Hz. The resonance frequency, as well as amplitude, drops as the system inertia increases.

Frequency analysis of the Nordic system equivalent support this data. This test found resonance for frequencies in the between 0.014 Hz and 0.022 Hz. This corresponds to a time period of oscillation between 45 and 70 seconds. This is similar to the frequency oscillation found in the

system today. Test show that the effect of the load damping decreases as the system grows in size.

Sammendrag

Denne rapporten tar sikte på å gjøre rede for og teste to Simulink modeller av vannkraftturbin og det nordiske kraftsystemet. Målet er at modellene skal være i stand til å kjøre frekvensanalyser av det nordiske synkrone kraftsystemet, og dermed kan bidra til å undersøke frekvensavvikene som er oppdaget i systemet.

Hydromodellen består av en ikke-lineær fremstilling av turbin dynamikken og en PID-governor. Kraftsystemet er modellert med all produksjon og forbruk i et samleskinne, og er basert på den mekaniske svingligningen, som gir frekvensresponsen til et system som en funksjon av dempning i systemet, roterende masse og avvik i produksjon og last. Begge modellene er blitt testet i tidsplanet og i frekvensplanet, med et særlig fokus på påvirkningen fra svingmassen og last dempningen.

Tidsplan analysen viser at hydromodellen har egenskaper som gjør at den kan erstatte eksisterende modeller, både alene i sammenligning med eksisterende modeller, og i større kraftsystem programmer. Testene viser dog at modellen er lite forutsigbar under enkelte omstendigheter. Dette gjelder spesielt når modellen ligger på grensen mellom stabil og ustabil drift. Dette er et resultat av at modellen ikke er lineær.

Kraftsystemmodellen ble også testet i tidsplanet. Den ble skalert til å tilsvare det nordiske systemet, både under tung og lett last. Responsen til systemet ble deretter testet ved at utfall i last og produksjon ble simulert. Testen viste at systemet var vesentlig mindre stabilt med mindre roterende masse i systemet.

Begge modellene ble testet i frekvensplanet, for å undersøke effekten av svingmasse og last dempning. Testen av hydroturbinen understreket tidsplan-analysen. Den viste at modellen var ustabil for visse kombinasjoner av høy svingmasse og lav lastdempning. De stabile simuleringene viste at modellen resonerer når eksitert av frekvenser mellom 0.004 Hz og 0.022 Hz. Testene viste at både resonansfrekvensen og amplituden ble redusert når svingmassen økte.

Frekvensanalysen av det nordiske systemet støtter disse resultatene. Simuleringene fant at systemet resonerer når det blir eksitert med en frekvens mellom 0.014 Hz og 0.022 Hz. Dette tilsvarer en tidsperiode for svingningene på 45 til 70 sekunder. Dette er i størrelsesordenen til

svingningene i det nordiske kraft systemet. Analysen viser at effekten av last dempningen avtar når systemet vokser i størrelse.

Acknowledgments

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Chapter 1
Introduction

This report investigates the implementation of a non-linear hydraulic turbine model in power system analysis. The motivation behind this work is the ongoing investigation into the floating frequency deviation uncovered in the Nordic synchronous area. Measurements show a sustained frequency deviation with a period of 60-90 seconds. So far, no one knows the full cause of this deviation, nor the full consequences, but it is assumed to cause increased wear on the turbines as well as reduce the functionality of the power system control mechanisms.

This report aims to investigate possible factors behind this instability, using both time series analysis and and frequency analysis. For the time series analysis, three different power system models have been used. Two of the models, KERMIT and Cidron, are products of DNV GL. The last model is created specifically for this project, as a simple, one-bus, equivalent of the Nordic power system. The goal of the time series analysis is attempting to replicate the frequency oscillations and to test the non-linear hydro model's interaction in a large-scale model.

The frequency analysis is performed on the non-linear turbine model, as well as a reference model, to find and compare the characteristics of the two models. Similar tests are run on the Nordic equivalent system. The goal is to uncover how the models respond when excited, and the effect of inertia and load damping has on the system.

All analysis in this report is modelled in Matlab and Simulink.

Chapter 2

The 60 Second Oscillation Project

2.1 Frequency stability and active power control

One of the most important characteristics of the power grid is the inability to store energy in the grid itself[1]. The consequence is that the sum of power produced and power consumed is at all times zero. To manually control this in real time by adjusting generation to changes in consumption is impossible. Therefore it is important to have systems that keeps the grid stable during normal operation.

Unscheduled load changes occur continually in the power system. These changes cause a mismatch between energy delivered to the generator, from the prime mover, and energy extracted to the grid. An unexpected decrease in load will cause excess energy in the generator. This energy will cause the generator to speed up, thereby increasing the grequency of the system. Similarily, a increase in load will case a deficit in energy in the gererator, and consequently a decrease in frequency.

2.2 The 60 Seconds Oscillation in Nordic Power Grid

For the last years, the frequency quality in the Nordic power grid has been decreasing as seen in Figure 2.1. As a result of this, the Nordic TSOs (represented by Statnett SF in Norway) have focused on the causes of this deterioration. One of the underlying causes is the floating frequency detected in the grid.

The project Measures to Mitigate the Frequency Oscillations with a Period of 60-90 Seconds in the Nordic Synchronous System, as reported in[**?**], is uncovering the factors influencing the oscillations, but the principal cause is not yet found. Figure 2.2 shows the oscillations during a 5 minute period in November 2013. Studies show that the amplitude of the frequency primarily lies between 0.015-0.030 Hz,

[1] It is of course possible to store engergy in the power system, externally from the grid, using e.g batteries or water reservoirs. This is, from a grid perspective, considered as consumption rather than storage.

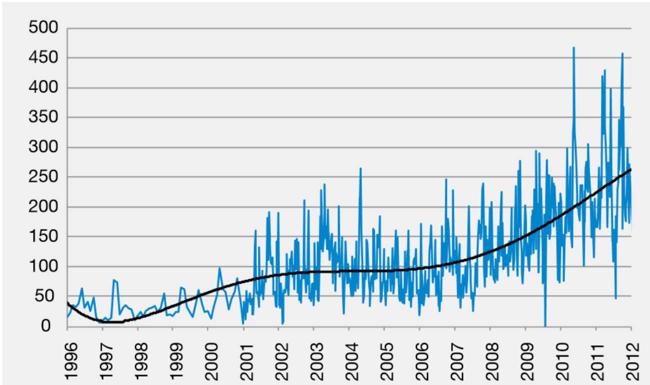

Figure 2.1: Minutes outside nominal frequency per month 1995-2011 in Nordic grid[**?**]

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while the time period is the region 60-90 seconds[**?**]. With a system as complex as the power grid, the cause of disturbances is probably complex, with more than one parameter contributing to the oscillation. This report will mainly focus on the theory presented in Section 2.3.

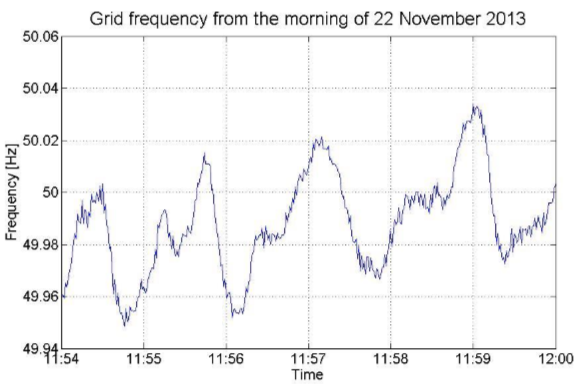

Figure 2.2: Frequency in the Nordic Synchronous System, 22 of November 2013[**?**]

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2.3 Critical Boundary of Oscillation

Critical Boundary of Oscillation (CBO) is a theory developed at DNV GL (former DNV KEMA). The theory is an attempt to uncover the cause of the oscillations in the Nordic power grid.[2] The idea is to use the swing equation to find the parameters that causes oscillation at a specified time period. If a generator were excited with a sinusoidal frequency deviation, assuming all other inputs are constant, one would expect a sinusoidal response in the power. In an unstable system, the oscillation will increase indefinitely, or until it reaches the system's physical boundaries. If the system is stable, the oscillations will not increase but will either be constant or be damped out over time to reach a new steady state. If the system is marginally stable, the system will experience a sustained oscillation[**?**]. Which of these oscillations occur during a disturbance, depends on the system parameters.

2.3.1 Derivation

The theory is built around the mutual influence between the power system and the generators. A change in production will cause the frequency to deviate, which in turn will cause the production to change. Assume that if an isolated generator is subjected to a sinusoidal frequency deviation, the result would be an oscillating response in the generator production. A necessary condition is that the oscillations are not so large that the generator response is restricted by the physical boundaries or governor limitations of the generator. Assuming a complex sinusoidal frequency deviation

$$\Delta f = a * \cos(\omega t) + j * a \sin(\omega t) = a * e^{j\omega t}, \tag{2.1}$$

Assuming the deviation in power in a generator i, from a frequency deviation Δf, will be dependent on the gain(DR) and time delay($T_o sc$)[3] of the generator, the contribution to the oscillation will be

$$\Delta P_i = -DR_i * \Delta f(t - T_i) = -DR_i * a * e^{j\omega(t-T_i)} \tag{2.2}$$

For all generators

$$\Delta P = \sum_i \Delta P_i = a * e^{j\omega t} \sum_i -DR_i * e^{-j\omega T_i} \tag{2.3}$$

The swing equation[**?**]

$$\Delta P = M * \frac{d}{dt}\Delta f + D\Delta f = (j\omega M + D) * a * e^{j\omega t} \tag{2.4}$$

Equating (2.3) and (2.4)

$$(j\omega M + D) * a * e^{j\omega t} = a * e^{j\omega t} \sum_i -DR_i * e^{-j\omega T_i} \tag{2.5}$$

[2] The CBO theory is yet to be formally published by DNV GL. It is presented here courtesy of DNV GL and Khoi Vu.

[3] The gain(droop) is the amplification of the oscillation, while the time delay is the time shift of the oscillations after a frequency disturbance is applied to a power plant. The parameters are frequency dependent, and can be found by sending a sinusoidal signal into the model.

$$(j\omega M + D) = -\sum_i DR_i * e^{-j\omega T_i} \tag{2.6}$$

$$(j\omega M + D) = -\sum_i DR_i(\cos(\omega T_i) - j\sin(\omega T_i)) \tag{2.7}$$

Separate real and imaginary part:

$$\omega M = \sum_i DR_i(sin(\omega T_i) \tag{2.8a}$$

$$D = -\sum_i DR_i(\cos(\omega T_i) \tag{2.8b}$$

2.3.2 The Effect of the Time Delay

These equations show that with given generator parameters, it is possible to detect whether there will be a sustained oscillation. It is possible to find boundary limits of inertia and load damping to ensure that oscillations fade out. Equation (2.8) has an unexpected consequence. Because the damping

$$D > 0$$

and the load sharing droop

$$DR_i \geq 0$$

in all real, physical, power systems, positive parameters, the term

$$\cos(\omega T_i) < 0$$

This gives the following

$$\omega T_i > \frac{\pi}{2} \longrightarrow \frac{2\pi}{T_{osc}} T_i > \frac{\pi}{2} \tag{2.9}$$

Rearranging (2.9) shows that for sustained oscillation, at least one generator must have a time delay

$$T_i > \frac{T_{osc}}{4} \tag{2.10}$$

Proposition 2.1. *A necessary, but not sufficient, condition for a sustained oscillation in the power system, is that at least one of the generators has a time delay that is longer than* $^1/_4$ *of the time period of the oscillation.*

With the time delay and droop of a generator, it is possible to find the dampening D and inertia M to create the critical boundary. At these values of D and M, the system will, according to the theory, be marginally stable and have a sustained oscillation.

Chapter 3
Modelling of the Power System

3.1 Power System Dynamics

3.1.1 Frequency regulation

The purpose of frequency regulation is to maintain a stable frequency in the power grid, within the specified boundaries of the system. In Norway, these boundaries are 49.90 Hz and 50.10 Hz [?]. The system frequency is closely related to the power balance in the system. A deviation between production and consumption will lead to an angular acceleration of the rotating masses in the power plant. The resulting speed deviation cases a change in frequency that will increase until a new steady state between production and consumption is reached. This new steady state depends on the available inertia in the system, as defined by [?]

$$\Delta\omega = (\Delta P_m - \Delta P_e)\frac{1}{2As} \tag{3.1}$$

For a system with n available generators, supplying inertia to the system, (3.1) becomes

$$\begin{aligned} 2A_1\dot{\omega}_1 &= P_{m,1,pu} - P_{e,1,pu} \\ &\vdots \\ 2A_n\dot{\omega}_n &= P_{m,n,pu} - P_{e,n,pu} \end{aligned} \tag{3.2}$$

Substituting for the per unit values, in unit i, for P, with the base values $P_{N,i}$ yeilds

$$2A_i\dot{\omega}_i = \frac{P_{m,i}}{P_{N,i}} - \frac{P_{e,i}}{P_N, i} \tag{3.3}$$

$$P_{N,i}2A_i\dot{\omega}_i = P_{m,i} - P_{e,i} \tag{3.4}$$

Neglecting electro mechanical oscillations

$$\dot{\omega}_1 = \dot{\omega}_i = \dot{\omega} \tag{3.5}$$

In a simplified system we assume that electric power in the generator equals the total load in the system, $\sum P_{e,i} = \sum P_L$, which along with (3.5), simplifies (3.4) to

$$\sum P_{N,i}2A_i\dot{\omega} = \sum P_{m,i} - \sum P_L \tag{3.6}$$

The inertia constant for the power system, A_{sys}, is defined as

$$A_{sys} = \sum P_{N,i}A_i \tag{3.7}$$

$$\dot{\omega} = \frac{1}{2A_{sys}}(\sum P_{m,i} - \sum P_L) \tag{3.8}$$

In the power system, the plants will be given different roles to ensure stable operation. In the Nordic grid this is defined in the System Operation Agreement. The Frequency Containment Reserve for Normal Operation (FCR-N) is a collection of the units constantly available for frequency control. The main purpose is control in normal operation. The Frequency Containment Reserve for Disturbances (FCR-D) is available for control during larger disturbances. The units in this reserve will assist in the frequency recovery, once the frequency falls outside the predefined limits. This limit is defined as 49.90 Hz. Further more, the Automatic Frequency Restoration Reserve (FRR-A) is sustained tor a specific time period. This is to counter the effects of load change in the evening and morning. In this report, all generators contributing to stability will either be classified as FCR-N or FCR-D.

In a connected power system, there will also be some frequency dependent load. The load dampening D determines the change in frequency will cause a change in load. The parameter K, the load-dampening, expresses the rate. D = 1 means that the load is changed by 1 percent if the frequency is changed by 1 percent [**?**].

$$\Delta P_e = \Delta P_L + K\Delta\omega \tag{3.9}$$

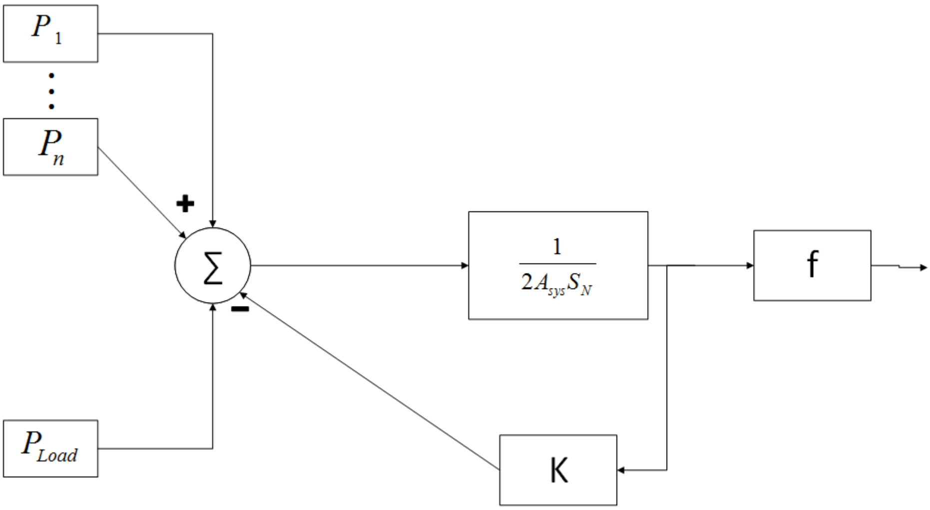

Figure 3.1: Power System frequency control. From equations (3.8) and (3.9)

When more than one generator is running in the system, an additional parameter is required to control the frequency. If parallel generators react individually to a frequency deviation by changing the production, the result would be over regulation and an unstable system. Each generator would be fighting the others to control the frequency. To prevent this, a parameter that determine the rate of change in production, as a function of frequency deviation, is added. This is the permanent droop characteristic.[1] The droop is designed to reduce the generator output as the frequency increases. Kundur[**?**] defines the speed regulation as

$$R = \frac{percent\, speed\, or\, frequency\, change}{percent\, power\, output\, change} \tag{3.10}$$

3.1.2 Spinning Reserves in the System

An essential part of the grid model is the representation of the spinning reserves [2]. This can be modelled in two different ways. The simplest is to set a constant value for the kinetic energy stored in the system. This approach can be adequate when modelling responses to load disturbances. When modelling disturbances in production, this would be inaccurate. Because the kinetic energy in the system is a

[1] Also called speed regulation, and denoted with R. Not to be confused with the transient droop r in the hydro governor or droop DR from the CBO theory in Section 2.3.

[2] In literature, the spinning reserves(kinetic energy) is called inertia. In these paragraphs, the terms kinetic energy E_k [MWs] and inertia time constant A_{sys} [s] are clearly separated. For the remainder of this report, the term inertia is considered to be the spinning reserves in MWs.

Table 3.1: Values used when calculating the inertia time constant. Data on kinetic energy from Statnett SF. Data on production from Svenska Kraftnät.[**?**]

	Norway	Sweden
E_k [MWs]	35 000	80 000
P_{hydro} [MW]	8269	8368
$P_{nuclear}$ [MW]	0	5515

product of the produced power, a drop in production would decrease the available spinning reserves.

To account for this, a different way of calculating the rotational energy is required. The kinetic energy of a rortating mass is defined as

$$E_k = \frac{1}{2} J \omega_m^2 \tag{3.11}$$

Here E_k is the kinetic energy in joule, J is the moment of inertia $[kg \cdot m^2]$ and ω_m is the speed [rad/s]. For power system studies, the kinetic energy is calculated as

$$E_k = S_N A_{sys} \tag{3.12}$$

where E_k is the kinetic energy [MWs], S_N is rated power [MVA] and A_{sys} is the inertia time constant in the system [s]. The inertia time constant A can be estimated on a unit level, or aggregated to a system level. A system does not fully reflect the different characteristics of each generator, while calculating A for every unit in the system requires very detailed and accurate data on every generator. In this report a compromise has been reached, estimating A for each generator type (hydro and thermal).

To estimate A_{hydro} and $A_{thermal}$, data on kinetic energy from Statnett and detailed production data from Svenska Krafnät was used. The data used is from May 17th 2015.

$$\begin{aligned} P_{hydro,Nor} * A_{hydro} &= E_{k,Nor} \\ P_{nuclear,Swe} * A_{nuclear} + P_{hydro,Swe} * A_{hydro} &= E_{k,Swe} \end{aligned} \tag{3.13}$$

Solving these equations yields $A_{hydro} = 4.2s$ and $A_{thermal} = 8.1$. These numbers are not intended to be fully accurate, and the simulations are not an exact representation of the Nordic grid on May 17 th 2015. The values are intended to give an estimate of the contribution of each turbine type in the system and a realistic estimate of the total kinetic energy in the system.

3.2 Power System Models

3.2.1 KERMIT

KERMIT[3] was initially created by KEMA as a renewable integration tool. It is a Simulink-based tool designed for real time simulation of the power system. Now KERMIT is part of DNV GL Energy and used in cooperation with TSOs. The KERMIT model used in this report gives a detailed description of the Nordic power grid. Figure 3.2 shows the principle structure of KERMIT.

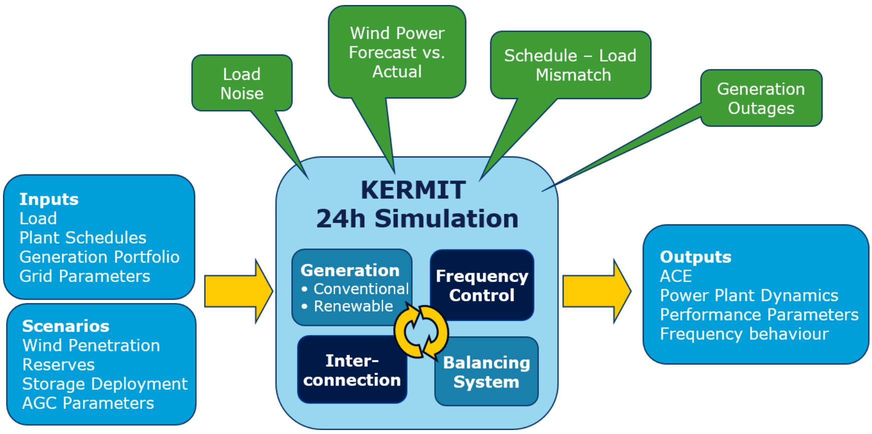

Figure 3.2: The principal structure of KERMIT. Property of DNV GL.

KERMIT allows the user to take a scheduled, steady state period, and introduce a disturbance to the system. The disturbance can small, such as a change in production or consumption, or larger, e.g. a generator trip. The model will then produce the resulting load flow, as well as other system parameters like frequency deviations and the angle for each area of the model. KERMIT works on a time period from 1 second to 24 hours. The time scale offers the possibility to inspect the oscillations over several periods. It is this possibility If the cause of the oscillations is found, KERMIT could be able to predict if the conditions in the power system facilitate the oscillations, before they occur. This allows the TSO to apply measures to reduce the consequences. The model parameters and scenario is handled externally, using Excel. This seperation allows the user to make substantial changes to the syteml, without changing the Simulink model.

[3]This section will not go into details on the architecture of KERMIT, as this is restricted information. The purpose of this section is not to facilitate the reproduction of the model, but to explain the nature of the program, and to explain why KERMIT is a part of the report. The same applies to the following section on the CIDRON model.

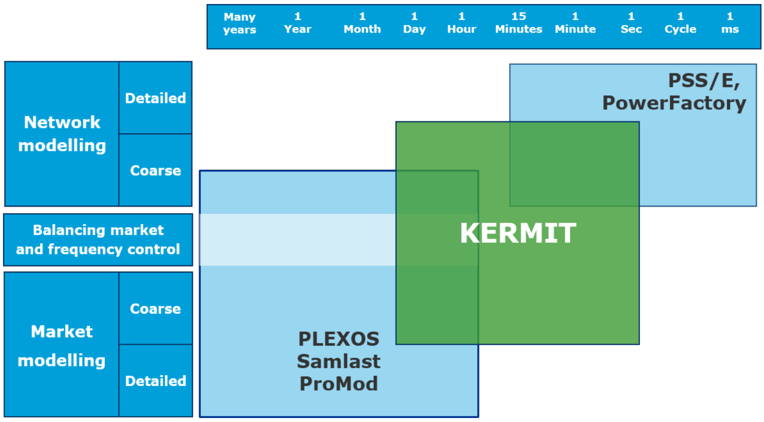

Figure 3.3: The time scale of various power system simulation tools. Created by DNV GL.

3.2.2 CIDRON

CIDRON is a tool developed by DNV GL to reproduce the frequency oscillations, based on the CBO-theory(Section 2.3). It is a Simulink model that consists of one or more generators, a simplified grid model and a load change. Inertia and damping is placed in the grid and is tuned to match the values predicted by the CBO-theory. Figure 3.4 shows the output from the CIDRON model. The inertia ($267.26MWs^2$) and load damping ($7.05MW/Hz$) calculated using the CBO theory. The result is an oscillating frequency with a time period of 60 seconds. This result show that it is possible to reproduce the oscillations for a system with 3 generators and 1 load.

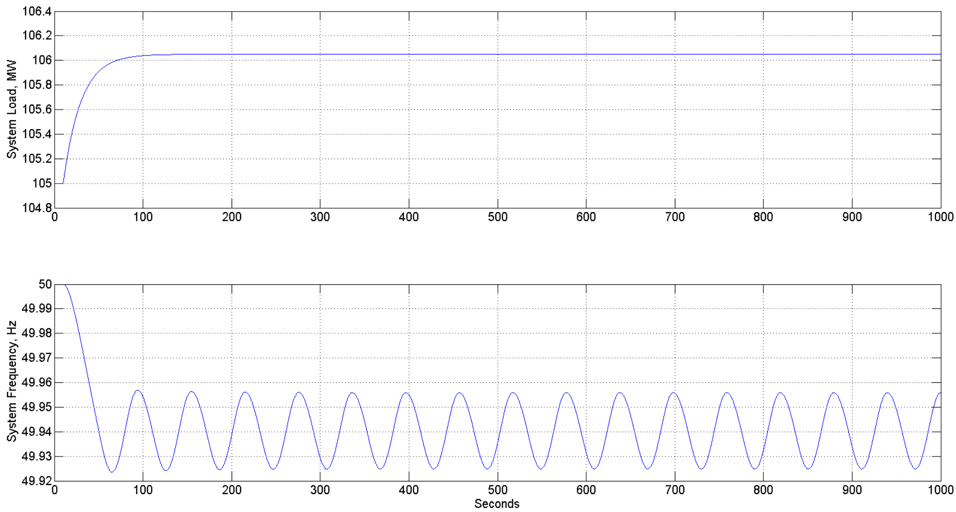

Figure 3.4: The frequency response of 3 generators (ideal hydro, Combustion and Combustion + steam turbines being subjected to a 1% increase in load. All turbine models built by DNV GL for use in the full KERMIT model.

3.2.3 Grid model

To test the HYGOV model it has been placed in a grid model. The model is designed to both preform standalone tests on a single generator as well as test to determine the dynamic interaction with multiple generators. The grid is considered lossless. In the one-generator scenario, the goal is to find the generator characteristics, and to find the frequency response of the generator. This is mainly done using different load disturbances. The multi-generator model consists of 37 generators, both hydro(HYGOV) and a simplified unit with droop and a prime mover to represent thermal generators.

The data is modelled to replicate the system dynamics during load change, with the system essentially behaving as a one-bus system. The model is subject to either a load disturbance or a generation disturbance. The load disturbance is modelled either as a continuous sinusoidal disturbance or as a step increase. Each generator is modelled with an individual inertia aggregated in the grid. As a default, each generator is loaded at 0.8 pu. This allows the generators some flexibility without triggering saturation in the model, while the production is still high enough to be considered fully committed. In reality, the loading would differ on each generator to reflect the individual efficiency and perfomance of the real life turbines.

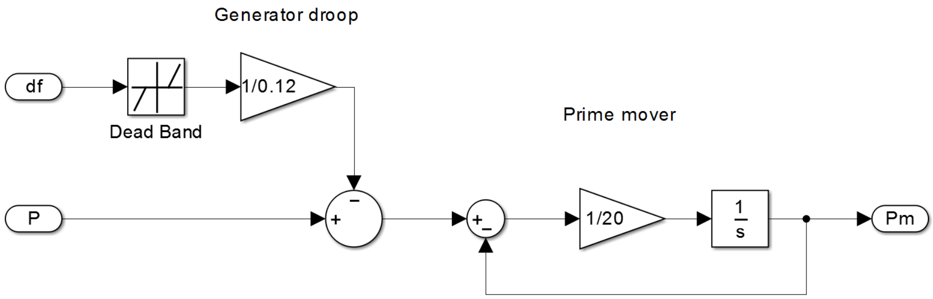

Figure 3.5: Simplified unit. Generator dynamics represented with a droop and a prime mover.

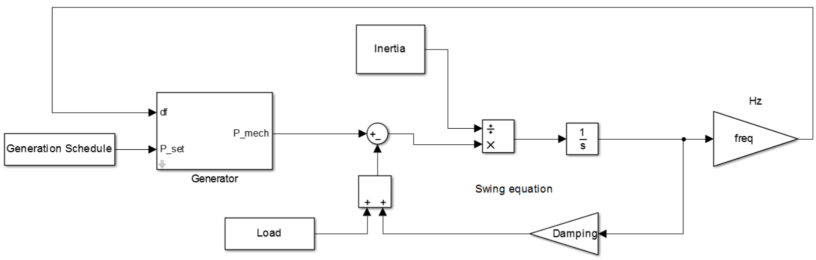

Figure 3.6: Basic design of grid model

Table 3.2: Generator data. The inertia Time Constant is defined in 3.1.2

Unit nr	1-50	51	52	53
Generation Type	Hydro	Nuclear	Nuclear	Nuclear
Rated Size [MW]	920	5000	3000	2500
Inertia Time Constant [s]	4.2	8.1	8.1	8.1

The load step change is used to detect the system characteristics, while the sinusoidal is used to uncover the frequency response of the system. The model also measures changes in production. This disturbance is directly, and proportionally, connected to the inertia in the system. Loss of a portion of production on a generator also means loss of the portion of inertia.

The model measures load, production, frequency and gain. In the case of multiple generators, the model also measures individual plant generation to gauge the individual response and commitment from each generator in event of a disturbance.

3.3 Frequency Regulation in The Power System

3.3.1 Regulation of Hydro Turbines

Hydro power plants can be split into three parts; water and penstock dynamics, turbine and governor and the generator. This part relates to the modelling of the turbine and governor. Section 3.3.2 covers the basic of the penstock and water modelling. Generator studies are beyond the scope of this work.

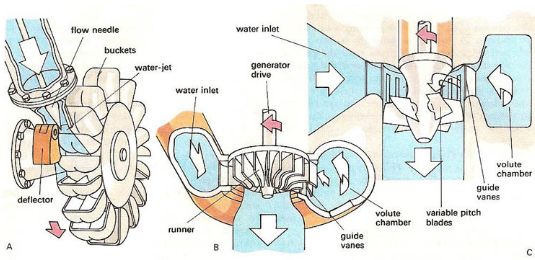

Figure 3.7: Three of the most common turbine types. Left to right: Pelton, Francis, Kaplan. Figure from Oregon State University

It is common to separate hydro turbines into two categories, impulse- and reaction turbines. The most common impulse turbine is the Pelton turbine. It is designed for power plants with a high head, and converts the kinetic energy of the water jet to torque to drive the turbine. A reaction turbine utilizes both the kinetic energy and the pressure drop across the turbine to drive the runner. For low heads and run-of-river power plants a propeller turbine, often a variable blade Kaplan, is most common. For head between the propeller turbine and the Pelton turbine, a Francis turbine is used. It is a reaction turbine, drawing power from both kinetic energy and the pressure head. The Francis turbine is the most common turbine in Norway[**?**]. There are no definite limits for when the different turbines are used. Kundur [**?**] suggests a head limit of 360 m for the Francis turbine, but Statkraft SF has installed Francis turbines in hydro plants with heads exceeding 500 m [**?**]. Often two or more turbine types are applicable, and the choice is subject to the developer's preferences, cost and the individual site characteristics.

The response of a hydro turbine, subjected to a load change, is somewhat unusual. In the same way as the inertia of the rotating masses in the system reduce sudden changes in frequency, the inertia of the water in the penstock will reduce the changes in flow in the turbine when the gate position is changed. As the pressure will drop when opening the gate and increase when closing it, a change in production schedule will initially have the opposite effect of what is desired. The time it takes for the change in production to take effect, is determined by the time water constant T_w. T_w is defined as the time it takes for water to reach the rated speed from stand

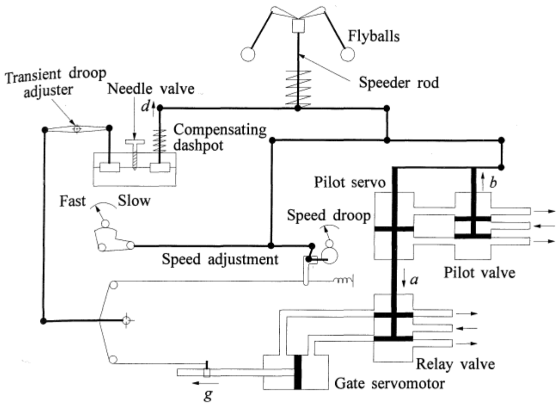

Figure 3.8: Mechancial-hydraulic governor[?]

still[?]. To compensate for this effect, the transient droop[4] r is introduced. This is designed to slow down the gate movement and reduce the delay in the response [? ?]. Another characteristic of the hydro governor is the amount of force it takes to control the mass of water. A pilot servomotor detect the speed, and act on the main servomotor to set the gate opening. This add to the time delay of the governor.

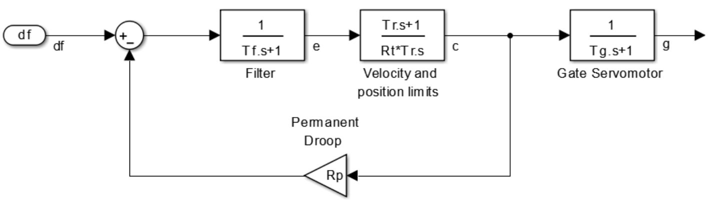

Figure 3.9: Governor model

3.3.2 Penstock and Turbine Dynamics

Modelling of the water way represents a challenge, because each water way will be individual and dependent on the local geography. The penstock dynamics will depend on the length of the conduit, the hydraulic head, pipe dimensions and characteristics and surge tank dynamics. In addition, the flow and pressure in the pipe is nonlinear. The most elementary way to represent the dynamics of the conduit, the ideal, lossless

[4]Transient droop is also called temporary droop. As the both names occur in governor models, both will appear in this report.

turbine[?] is shown in equation 3.16

$$\frac{\Delta\overline{P}_m}{\Delta\overline{G}} = \frac{1 - T_w s}{1 + \frac{1}{2}T_w s} \tag{3.14}$$

Where T_w is defined as

$$T_w = \frac{LU_0}{a_g h_0} \tag{3.15}$$

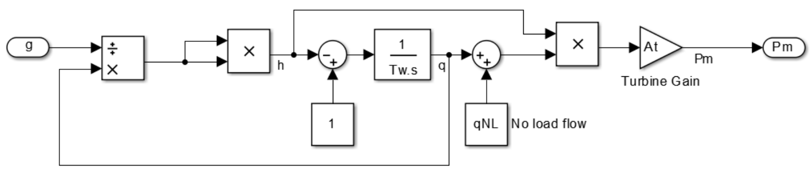

Figure 3.10: Turbine and Penstock Dynamics

The ideal turbine is not suited for studies with large fluctuations in output and input, and more advanced models are needed for this purpose[?]. Section 3.4.1 shows a model with a more advanced representation of the penstock dynamics, with a nonlinear turbine, inelastic water column and unrestricted tail race (here called HYGOV). More detailed representation of the water way, and considerations regarding their accuracy, can be found in [?]. Kundur argues that the hydro governor acts slowly compared to the dynamics in the conduit, and as such will have a larger affect on transient stability. Care should be taken when dealing with long penstocks, to compensate for the water hammer effect, and when dealing with long term studies. For this report, the two models described in Section 3.4, the ideal turbine and HYGOV, will be considered adequate.

3.4 Turbine and Governor Models

For studying the oscillations in Simulink, models of the hydro power plant are needed. These models should represent the real, and preferably individual, power plants. This can create a dilemma. Because there is not unlimited information available on all individual power plants, some of the parameters will be subject to approximation. The validity of the model will then be dependent on accuracy of the parameters and the accuracy of the model itself. Two different hydro models have primarily been considered. One is an IEEE model called HYGOV. This model is suggested in [?] as a combination of the mechanical-hydraulic governor and nonlinear turbine with inelastic water column. The other model, called the HYDROFrancis is created by DNV GL. This model represents a simplification of the HYGOV model.

3.4.1 HYGOV

The HYGOV model is a commonly used representation of the hydro turbine and governor dynamics. It is included as a built-in model in both PSS/E and PSLF, two widely used power system simulation software's[**?**]. Creating a Simulink equivalent of the HYGOV allows for the possibility of testing results and conclusions in an external environment. The Simulink model itself is based on flowcharts from Siemens describing the PSS/E model.

The HYGOV model can be split into two parts, one representing the governor and one representing turbine and penstock dynamics. [**?**] describes both parts. The governor is equivalent to the mechanical-hydraulic governor. It utilizes the speed deviation to calculate the resulting gate position. In the Simulink model, the production schedule is also an input, to make it compatible with the full KERMIT software. The governor dynamics are presented in Section 3.3.

The hydro penstock is modeled without a surge tank, unrestricted head- and tailrace and an inelastic water column. The model takes the gate position, calculated by the governor, to find the mechanical power. Using the water time constant T_w, the time it takes for the flow to go from 0 to rated, it is possible to calculate the initial flow [**?**]. KERMIT handles load dampening differently compared to other models. This is because the dampening is implemented in transmission system dynamics. The full model is shown in Figure 3.11. The model in the figure is a simplification of the model adapted from PSS/E.

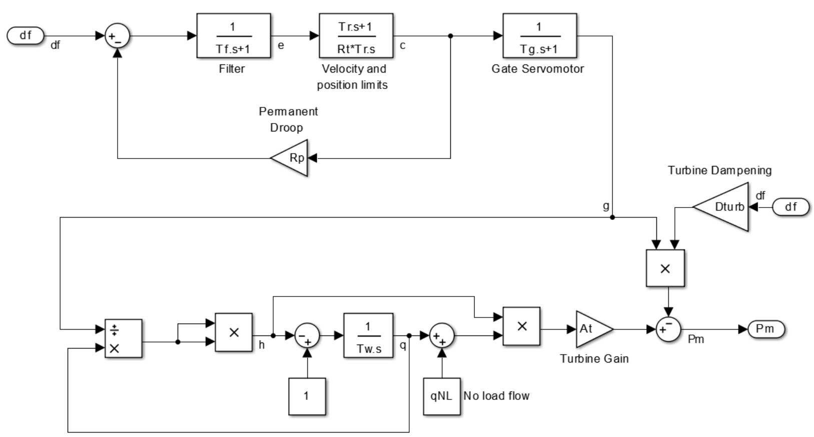

Figure 3.11: Simplified version of the Simulink HYGOV model

The model parameters are shown in Table 3.3.

Table 3.3: HYGOV parameter range as defined in PSS/E

Parameter	Symbol	Default	Min	Max
Permanent Droop	R	0.06	0.01	0.1
Temporary Droop	r	0.4	0.02	1
Governor Time Constant	T_r	8	0.02	10
Servo Time Constant	T_g	0.2	0.01	1
Filter Time Constant	T_f	0.05	0.01	1
Gate Velocity Limit	VELM	0.2	0.01	0.25
Maximum Gate Limit	GMAX	1	-1	10
Minimum Gate Limit	GMIN	0		
Water Time Constant	T_w	1.2	-1	100
Turbine Gain	A_t	2.5	0.5	3
Turbine Damping	D_{turb}	0.3	0	0.5
No Load Flow	q_{NL}	0.5	0.01	0.5

A problematic aspect of this representation, in terms of modelling, is the lack of linearity. Particularly with respect to flow and gate position. When a model includes division of physical quantities, there is a high risk of encountering singularities. Singularities occur when the flow i zero or during initialization. During initialization, the problem was solved either by rerouting the signal during the first 5 seconds, or by setting the initial conditions $\neq 0$. A bigger problem occurs when the generator is not producing. This is particularly relvant when simulating the interaction between multiple generations and response to disturbances in load or generatrion, as shown in section 3.2.3.

3.4.2 The KERMIT Francis model

DNV GL's Francis model is a simplification of the HYGOV model in PSS/E. The governor is the same as in HYGOV, but it uses a simplified penstock and turbine representation. The penstock dynamics are removed, and the turbine is modeled as an ideal, lossless turbine [**?**]. (3.16) shows the transfer function for the changes in mechanical power due to change in gate position.

$$\frac{\Delta \overline{P}_m}{\Delta \overline{G}} = \frac{1 - T_w s}{1 + \frac{1}{2} T_w s} \tag{3.16}$$

Table 4.3 shows the parameters used by DNV GL. The fixed parameters are set to make the model as close to Statnett's HYGOV in PSS/E. The other parameters are specified for individual plants in KERMIT to reflect the individual characteristics.

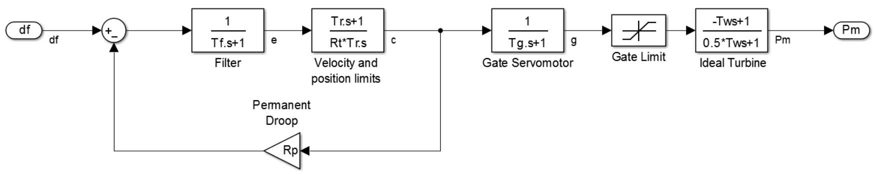

Figure 3.12: Simplifired version of HYDROFrancis used in KERMIT

Chapter 4
Simulation

4.1 Model Comparison

4.1.1 Square Wave PSS/E Comparison

The purpose of using the HYGOV model was to get results that are possible to verify and replicate in other programs. Therefore it is necessary to compare the output from the two models. In order to compare the two models, a single bus system was created in PSS/E, with one generator and one load. The PSS/E model is shown in Figure 4.1.

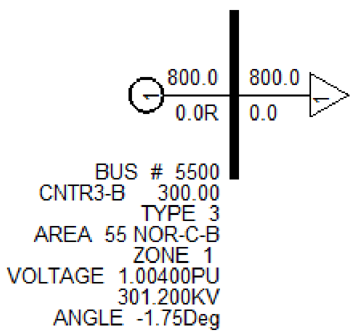

Figure 4.1: PSS/E model created by DNV GL for testing of Simulink model

The model had a constant load, and was submitted to a disturbance. This disturbance is shown in Figure 4.2. A Simulink model was created with the same parameters, and subject to an identical disturbance. Table 4.1 shows the parameters used in the models. The output of the two models can be seen in Figure 4.3. It is important to note that the two models are not completely identical. The PSS/E model contain a generator, meaning the output is electrical power. The HYGOV only has a turbine and governor equivalent, meaning the output is mechanical power. Figure 4.3 indicate that the HYGOV created in Simulink only has small deviations in the response compared to the PSS/E version.

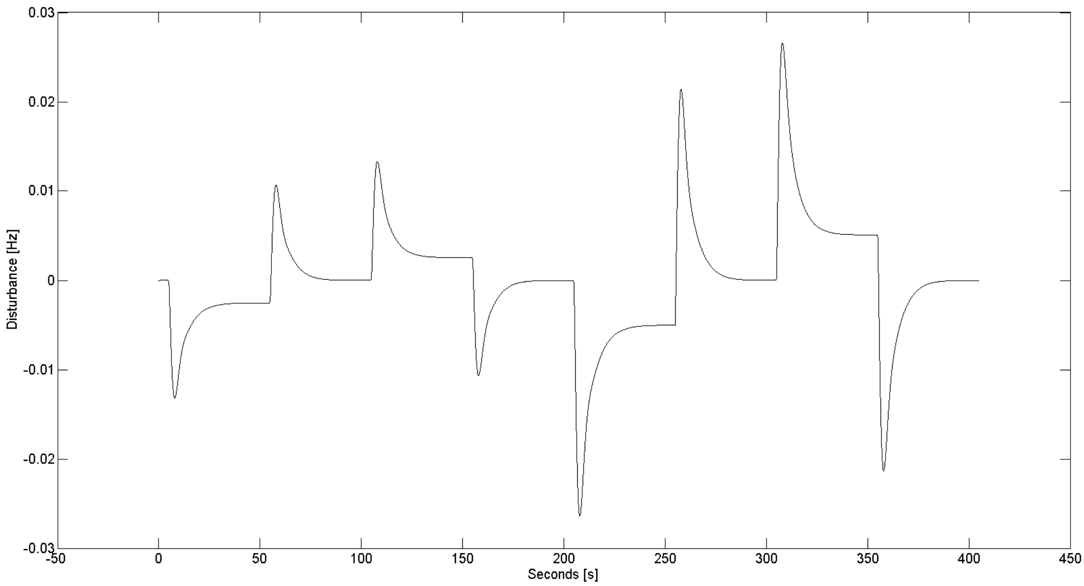

Figure 4.2: The disturbance in the square wave test

Table 4.1: HYGOV parameters used for the square wave test

Parameter	Symbol	Value
Permanent Droop	R	0.06
Temporary Droop	r	0.4
Governor Time Constant	T_r	5
Servo Time Constant	T_g	0.2
Gate Velocity Limit	VELM	0.2
Maximum Gate Limit	GMAX	1
Minimum Gate Limit	GMIN	0
Water Time Constant	T_w	1
Turbine Gain	A_t	1.1
Turbine Damping	D_{turb}	0.5
No Load Flow	q_{NL}	0.1

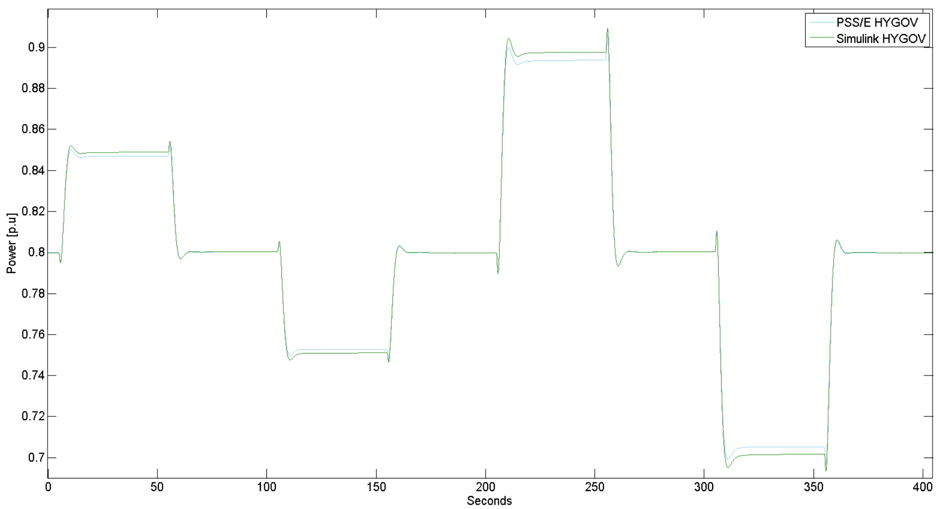

Figure 4.3: Comparison of the Simulink and PSS/E versions of HYGOV, when subjected to a square wave disturbance.

Table 4.2: HYGOV parameters used in KERMIT

Parameter	Symbol	Value
Permanent Droop	R	0.08
Temporary Droop	r	0.4
Governor Time Constant	T_r	8
Servo Time Constant	T_g	0.2
Gate Velocity Limit	VELM	0.2
Maximum Gate Limit	GMAX	1.5
Minimum Gate Limit	GMIN	-1.5
Water Time Constant	T_w	1.2
Turbine Gain	A_t	1.1429
Turbine Damping	$D_t urb$	N.A
No Load Flow	$q_N L$	0.1

Table 4.3: HYDROFrancis parameters used in KERMIT. Remaining paramers individually determined

Parameter	Symbol	Value
Filter Time Constant	T_f	0.05
Temporary Droop	r	0.4
Governor Time Constant	T_r	8
Servo Time Constant	T_g	0.2
Water Time Constant	T_w	1.2

4.2 Time Delay

On of the essential concepts of the CBO theory is the time delay. In order for the inertia and load sharing droop to have valid, positive values, the time delay must be larger than 1/4 of the time period of the oscillation. All parameters were tested over a range, to identify the parameters that affect the time delay the most(see appendix). The results show that the governor time constant and the permanent droop influence the time delay the most. A Matlab script was created to test the time delay over a range of T_r and R to find the necessary values. The full values are shown in Table 4.4. Figure 4.4 show that HYGOV can achieve the desired time delay, if the circumstances are correct.

Table 4.4: HYGOV parameters for time delay test

Parameter	Symbol	Value
Temporary Droop	r	1
Servo Time Constant	T_g	1
Filter Time Constant	T_f	0.1
Gate Velocity Limit	VELM	0.2
Maximum Gate Limit	GMAX	1
Minimum Gate Limit	GMIN	0.001
Water Time Constant	T_w	3
Turbine Gain	A_t	1.1
Turbine Damping	D_{turb}	0
No Load Flow	q_{NL}	0

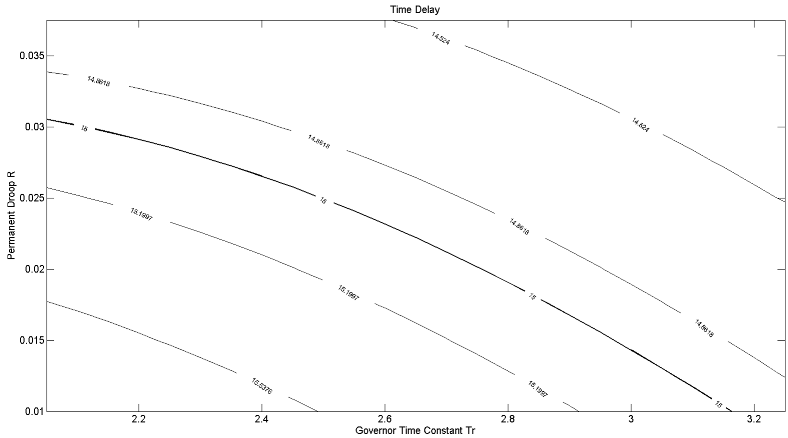

Figure 4.4: Plot shows the time delay as a function of R and T_r for a 60 second oscillation. 15 seconds is marked as the boundary.

4.3 CIDRON results

According to theory and testing so far, the HYGOV model should be able to reproduce the frequency oscillations during the right circumstances. Using the HYGOV version tested in KERMIT, with the parameters in Table 4.5, this was attemted. First a 60 second oscillation was tested. Figure 4.5 shows a plot of a disturbance and the resulting P_{mech}. This is used to estimate DR = 13.22 MW/Hz and the time delay T_i = 16.21 s. Using (2.8) we get D = 1.67 MW/Hz and M = 125.21 MWs^2.

Table 4.5: HYGOV parameters used in CIDRON

Parameter	Symbol	Value
Permanent Droop	R	0.01
Temporary Droop	r	0.4
Governor Time Constant	T_r	2
Servo Time Constant	T_g	0.2
Filter Time Constant	T_f	0.05
Gate Velocity Limit	VELM	0.2
Maximum Gate Limit	GMAX	1.5
Minimum Gate Limit	GMIN	-1.5
Water Time Constant	T_w	1.2
Turbine Gain	A_t	1.1429
Turbine Damping	D_{turb}	N.A
No Load Flow	q_{NL}	0.1

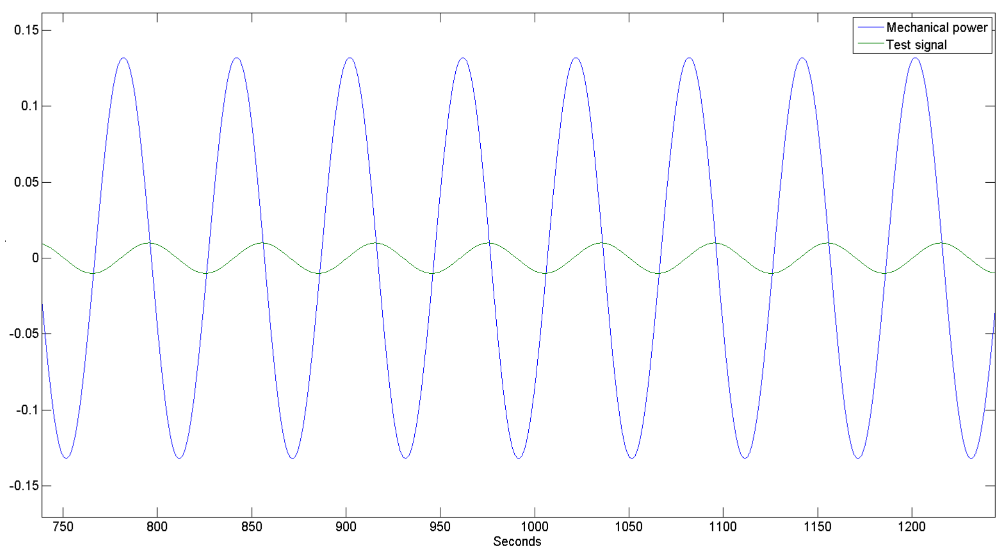

Figure 4.5: Plot shows the relation between the disturbance and the mechanical power(adjusted by the sceduled power to make it easier to compare).

The HYGOV model and the parameters were used in the CIDRON-model and tested for a 1% increase in load. Figure 4.6 shows the response. The response is clearly unstable. The distortion of the signal, as well as the magnitude, suggests that the sustained oscillation is caused by the governor constraints rather than a stabilization of the response itself. Repeating the process for oscillations with a 40 second time period and 80 second time period yeild the same result(see appendix).

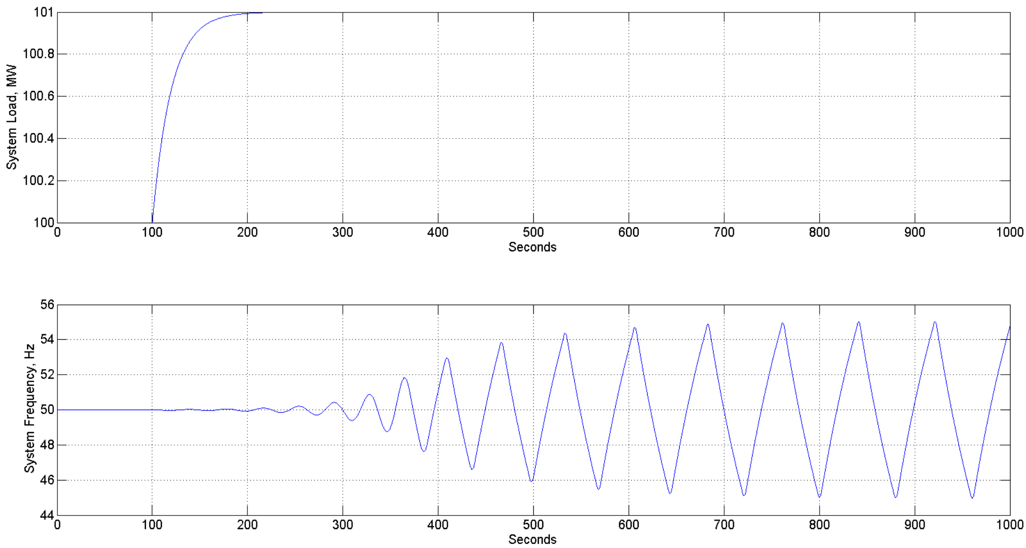

Figure 4.6: The frequency responce of 1 HYGOV subjected to a 1% increase in load.

Another test was done with three generators, again trying to achieve a 60 second oscillation, using the model from Figure 3.4 and substituing HYGOV as the hydro turbine. This initially gave promising results with what apppeared to be the required oscillations. However, a more thorough study of the results showed that there was a miscalculation in the system parameters, due to an erronous substitution of cosine for sine. Correcting the error, and recalculating D = 9.73 MW/Hz and M = 156.44 MWs^2 and running the model gave a responce where the oscillations fade quickly, as seen in Figure 4.7. Trying to get a 40 second oscillation yeilds the same result, as seen in Figure 4.8.

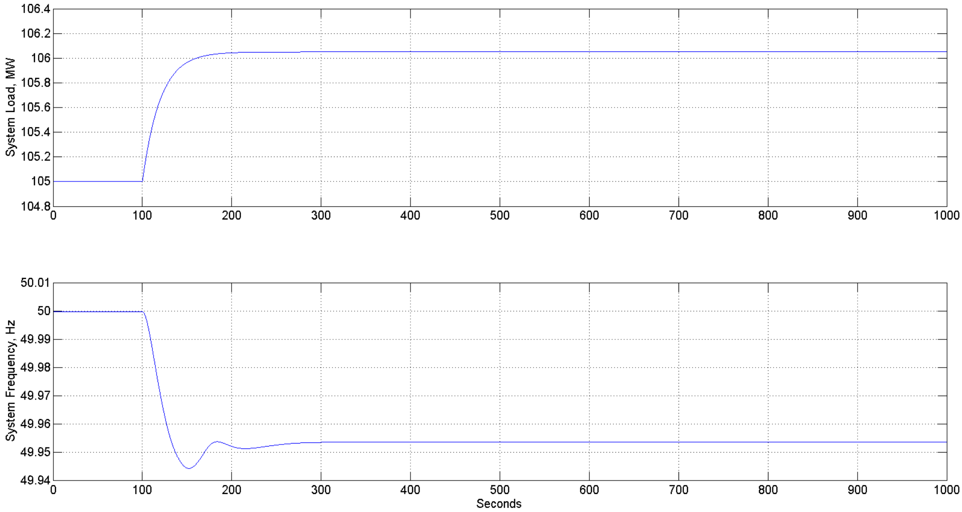

Figure 4.7: The frequency responce of 3 generators subjected to a 1% increase in load. A 60 second oscillation was expected

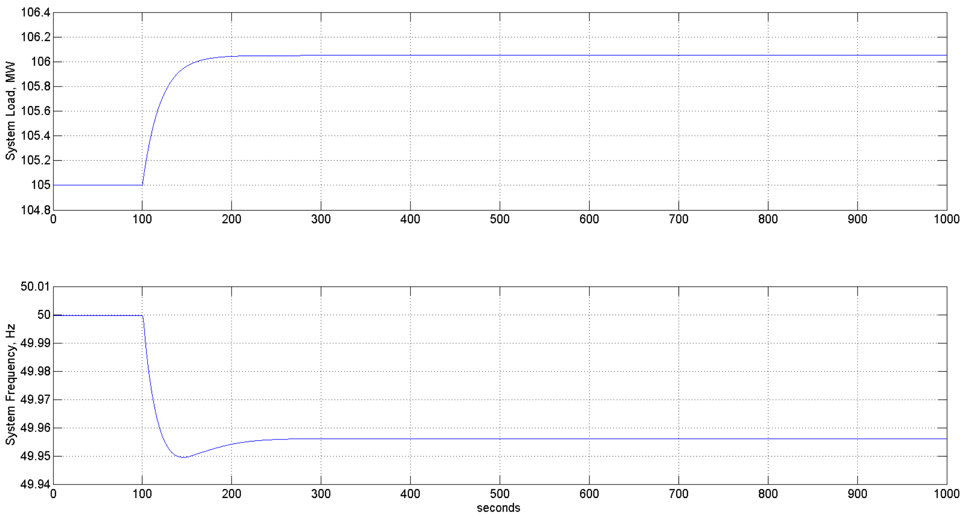

Figure 4.8: The frequency responce of 3 generators subjected to a 1% increase in load. A 40 second oscillation was expected

4.4 Frequency Analysis of HYGOV model

From the graphs above, it is clear that the HYGOV-model behaves somewhat different to the francis model when subjected to a sinusoidal disturbance. To investigate this deviation, the frequency response of the two models was analyzed. Using the grid model in Figure 3.6, with one generator, and subjecting the load to a sinusoidal frequency deviation, the generator gain was explored. The gain is calculated as the relationship between the amplitude of the load disturbance en the frequency deviation.

$$Gain = \frac{\Delta P_L}{\Delta f} \tag{4.1}$$

The generators where dimensioned at 50 MW, and loaded at 0.8 pu (40 MW). The amplitude of the load disturbance was 0.1 MW. In the figure

To compare the two models, a series of simulations were run. Using 9 different values of load damping amd inertia constant, 81 different combinations were tested in both models. Figure 4.9 to Figure 4.12 shows the response for both generators, for a select range of values. The biggest difference between the two models is that the HYGOV response is unstable for 7 of the combinations. These combinations all have load damping of 2 MW/Hz and a inertia time constant equal to or greater than 1.25 s. The response of the francis model is stable for all tested combinations.

4.4.1 Analysis of the Effect of Load Damping

The damping has very little effect on frequency of the maximum gain. It does have a significant effect on the magnitude of the response, as the load damping will reduce the effects of a load disturbance. This effect is amplified in Figure 4.9 and Figure 4.10 due to the load damping being unrealistically high in some of the cases. A load damping of 16 MW/Hz will be close to 50

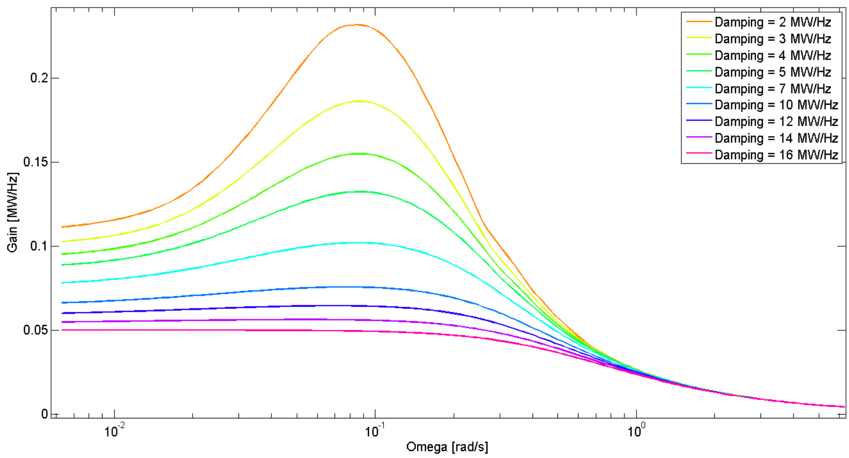

Figure 4.9: Frequency sweep of the HYGOV-model, over a select range of Damping. Constant inertia

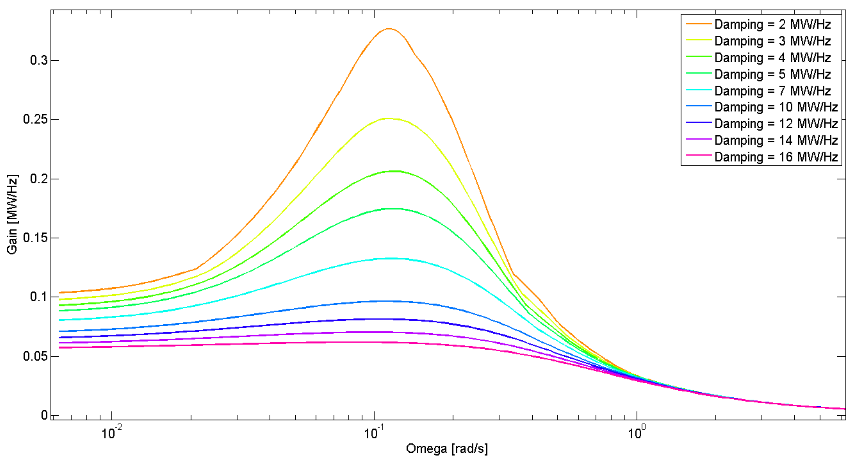

Figure 4.10: Frequency sweep of the francis-model, over a select range of Damping. Constant inertia

The magnitude of the response is similar, the gain in the francis model is between 135-140 % of the gain in the HYGOV model, with the differences highest with the lowest values of load damping. The models react to different frequencies. The HYGOV model shows the highest response for frequencies with a time period of 74.0 s. For the francis model the same value is 54.7 s.

4.4.2 Analysis of the Effect of Inertia

The inertia affects both amplitude of the response and the frequencies of the maximum amplitude. Figure 4.11 and Figure 4.12 shows that the inertia increase the ability of the system to oppose changes in load. As with the load damping results, the magnitude of the response is similar in the two models. The amplitude in the francis model lies between 116 – 145

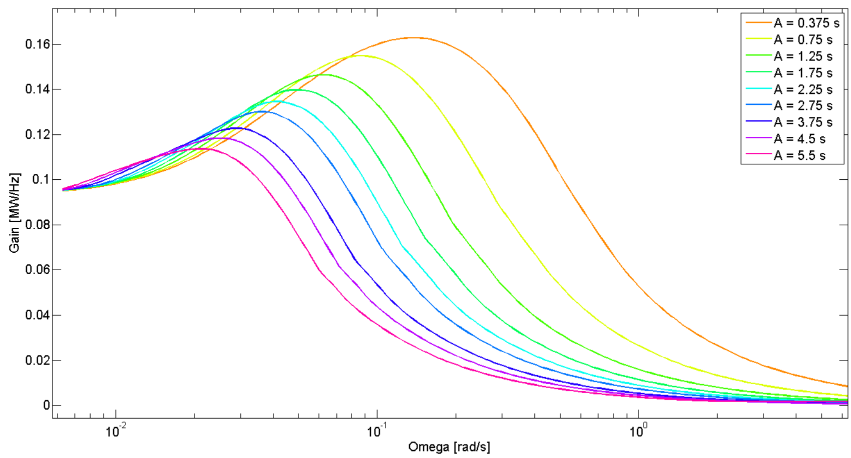

Figure 4.11: Frequency sweep of the HYGOV-model, over a select range of Inertia. Constant damping

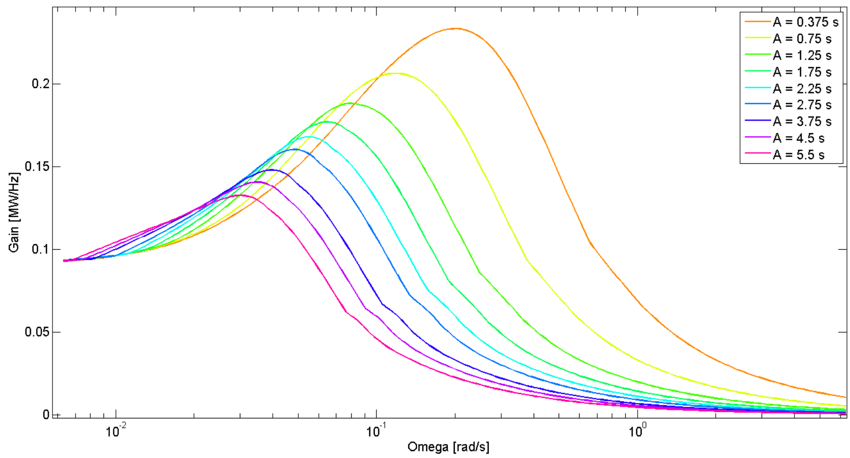

Figure 4.12: Frequency sweep of the francis-model, over a select range of Inertia. Constant damping

Figure 4.13 shows the the time period of the peak of the maximum response as a function of the time period. The figure shows that the HYGOV model is excited at lower frequencies than the francis model.

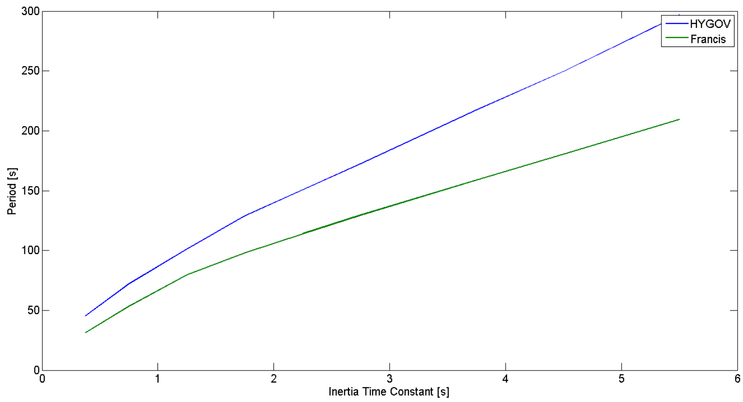

Figure 4.13: Time period of the maximum gain in Figure 4.12 and Figure 4.11 as a function of the inertia constant.

4.5 Grid Analysis of Nordic Equivalent

To test the time response of the model in a more realistic scenario, it is necessary to investigate the dynamic interaction with other generators. The model in Figure 3.6 was scaled to represent the Nordic Synchronous area. Production is split into three parts. The first two, thermal and hydro, are represented by the simplified model, Figure 3.5 and HYGOV model, Figure 3.11, respectively.

The last part consists of all generation and transmission that does not contribute to frequency stability. This includes both intermittent renewable generation and net import from outside the synchronous area (HVDC connections) and does not supply the system with any additional inertia. It is modelled as a constant, numerical increase in production. The purpose is to reflect that not all sources in the system are part of the frequency control and to prevent the constant A_{sys} from becoming too large , Section 3.1.2.

4.5.1 Load and Generation Scaling

The model is scaled to represent an equivalent of the Nordic Synchronous area. Two scenarios are considered, a high load scenario (P_L = 53200 MW based on maximum load in the Nordic system 2008 [**?**]) and a low load scenario (P_L = 28640 MW). The total installed capacity in the hydro generators is 46000 MW, divided equally between 50 generators of 920 MW. A total thermal capacity of 10500 MW is divided between 3 turbines of 5000 MW, 3000 MW and 2500 MW respectively [1]. The scheduled production on each generator is always 0.8 pu. When reducing production, the number of units in operation is reduced, not the commitment of each unit.

4.5.2 Simulation Scenarios

Table 4.6 shows the general settings for both scenarios. The values not presented in the table are specified in Table 3.3 or Figure 3.5. In each scenario, 3 different cases are studied; loss of 1 hydro generator (736 MW), loss of 1 thermal generator (2400 MW) and loss of load (50 MW).

[1]The thermal units are dimensioned to be at the approximate size of Sweden's three nuclear plants

Table 4.6: General Grid Parameters

Parameter	Symbol	Value
HYGOV parameters		
Permanent Droop	R	0.08
Temporary Droop	r	0.5
Governor Time Constant	T_r	8 s
Turbine Gain	A_t	1
Thermal model parameters		
Dead Band	DB	± 0.1 Hz
Grid Parameters		
Load Damping	K	5 MW/Hz
Inertia Constant Hydro	A_{hydro}	4.2 s
Inertia Constant Nuclear	$A_{thermal}$	8.1 s
Generator Comittment	P_{set}	0.8 pu

Table 4.7: Scenario specification

Scenario		Full load	Low load
System Load	[MW]	53200	29008
Hydro Generation	[MW]	36800	20608
Thermal Generation	[MW]	8400	4400
Additional Generation	[MW]	8000	4000
Units Commited		1-53	1-28,52,53

4.5.3 Scenario One - Full loaded System

For each case, the model is initialized at steady state. Then, at t = 500 s, a disturbance is introduced. The test assumes that no manual control action is taken by the TSOs. In the first case, a loss of a hydro generator is modelled. The effect is a loss of 736 MW production and 3091.2 MWs of inertia. The resulting frequency deviation is shown in Figure 4.14.

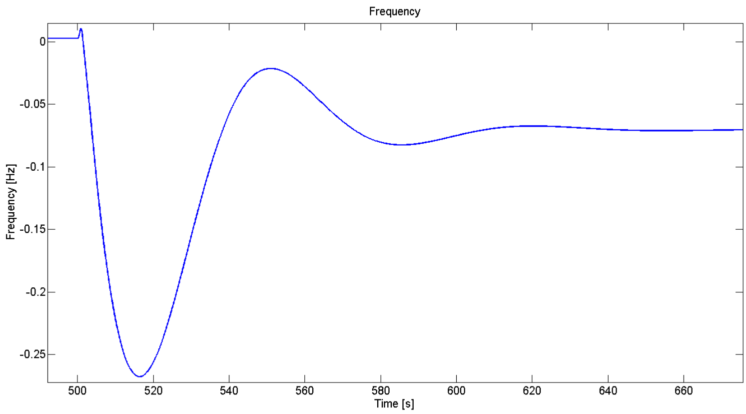

Figure 4.14: Frequency deviation after disconnection of generator 50 (hydro).

The figure shows a characteristic increase in frequency, due to increased pressure in the penstock, before the frequency drops to 49.74 Hz in t = 518 s. After 100 s the frequency stabilizes at 49.93 Hz. At the lowest point, the frequency is outside what is defined as normal operation $\pm$ 0.1 Hz. The new steady state is within normal operation limits.

The second case simulates the loss of a nuclear power plant in Sweden. Generator 52 is disconnected, resulting in a loss of 2400 MW and 19440 MWs. This represents a loss of 27.3 % of the system inertia, and the result is shown in Figure 4.15.

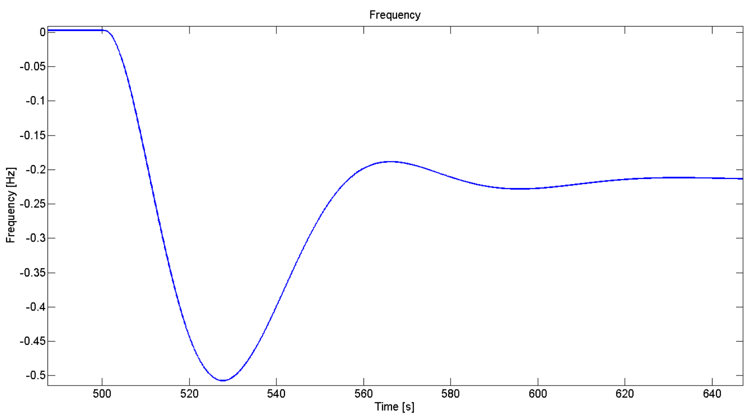

Figure 4.15: Frequency deviation after disconnection of generator 52 (nuclear).

The frequency drops to 49.5 Hz, far outside the normal operating point. The new steady state is reached at 49.78 Hz, still outside the normal operating area. Considering the magnitude of the disturbance, particularly the loss of inertia, the large frequency deviation is not surprising. To reach a satisfying operating point, the TSOs are required to take active measures.

The third case simulates a loss of load. This disturbance is the smallest of the three, causing the load to drop 50 MW. The inertia is not reduced. The frequency response is shown in Figure 4.16.

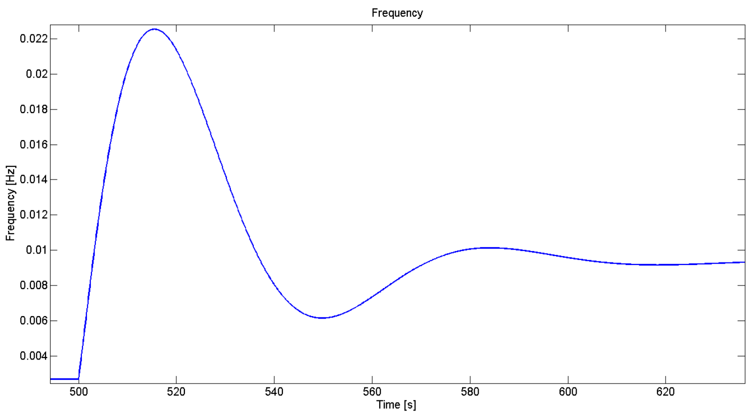

Figure 4.16: Frequency deviation after loss of load (50 MW).

The frequency is within the normal operating point for the duration of the simulation. The peak value is 50.02 Hz, while the system stabilizes at 50.01 Hz.

4.5.4 Scenario Two - Low loaded System

The 3 same cases are simulated again, this time in a system with a lower load. This makes the disturbances comparatively larger and should lead to a larger deviation in frequency. For the first scenario, with a high load system, losing one hydro generator caused to frequency in the system to go outside the normal operating region for 30 s. For a system with a lower load, and lower inertia, the response is shown in Figure 4.17.

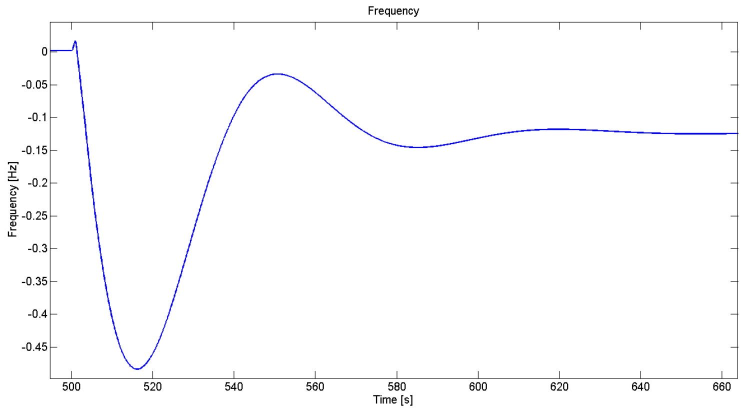

Figure 4.17: Frequency deviation after disconnection of generator 50 (hydro).

The shape of the frequency response is similar, but the magnitude is almost double. The frequency drops to 49.55 Hz, before reaching a steady state at 49.82 Hz. This is below normal operating condition, which means the TSOs are required to take additional action to ensure satisfactory operation in the system.

In the second case, loss of a nuclear turbine, the magnitudes are even larger. The disturbances cause to load to drop by 8.3% and the inertia drops by 15.9 %. The frequency deviation is shown in Figure 4.18.

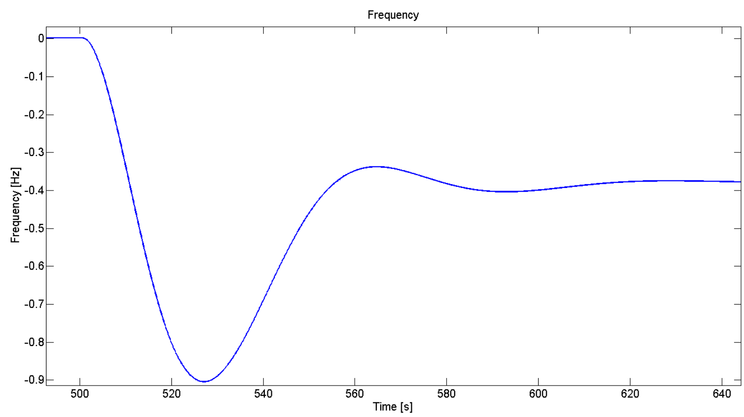

Figure 4.18: Frequency deviation after disconnection of generator 52 (nuclear).

The frequency drops to 49.1 Hz at the lowest, and the new steady state is reached at 49.61 Hz. The magnitude of the frequency deviations in this simulation are so large, one would expect electrical transients in transformers and generators. Only the mechanical transients are modelled here, and it is possible that the system is not as stable as the frequency response would suggest. In that case emergency action would have to be taken by the TSOs to prevent the system from becoming instable.

For the final case, the loss of load, the result is similar to the high load scenario. The frequency is never above the normal operating limits, and stabilizes at 50.015 Hz. The frequency deviation is shown in Figure 4.19.

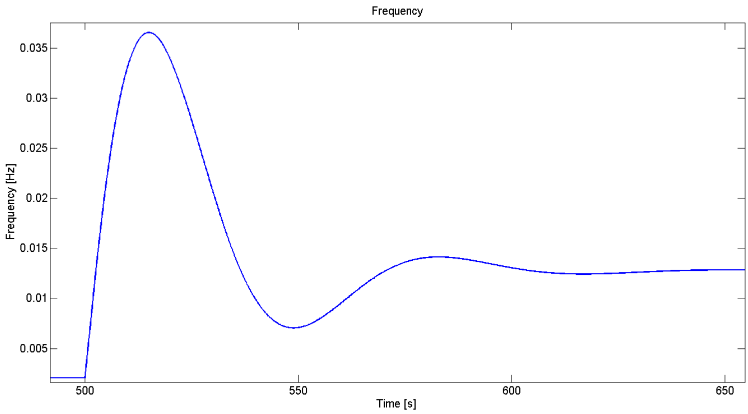

Figure 4.19: Frequency deviation after loss of load (50 MW).

These results also reveal the limitations of the system model. The test here shows that the disturbances are much more severe for the second scenario (low load) because the relative disturbance is much bigger. In a full grid model, the effect of the thermal limitations in the system would limit the power transfer in the first scenario (high load). This is not accounted for in this system model, which in turn causes it to appear more stable and robust than is actually the case.

4.5.5 Frequency Analysis of the Nordic Synchrounos Grid

A frequency sweep has been preformed on the system for the high load and low load scenarios. Two different values of load damping (5 MW/s and 50 MW/s) and two different values of inertia time constant (3 s and 5 s) were tested, on both scenarios. The results are shown in Figure 4.20 and Figure 4.21.

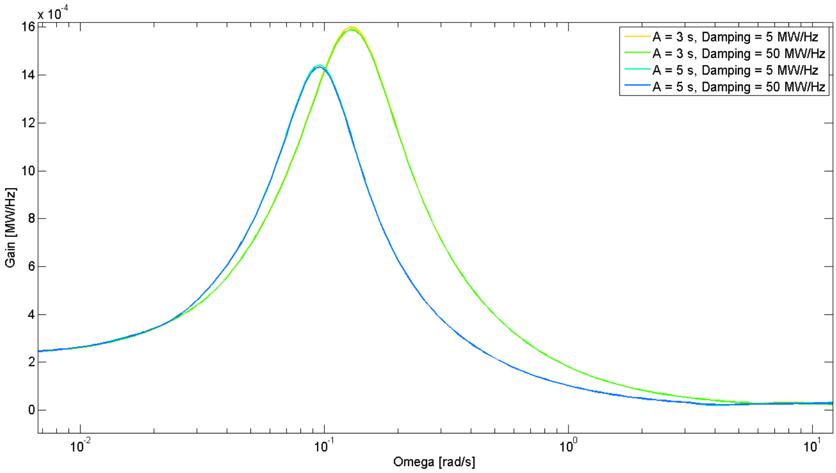

Figure 4.20: Frequency sweep of equivalent of the Nordic Synchronous Area. Load $P_L = 53200$ MW. System inertia $E_k = P_L * A$

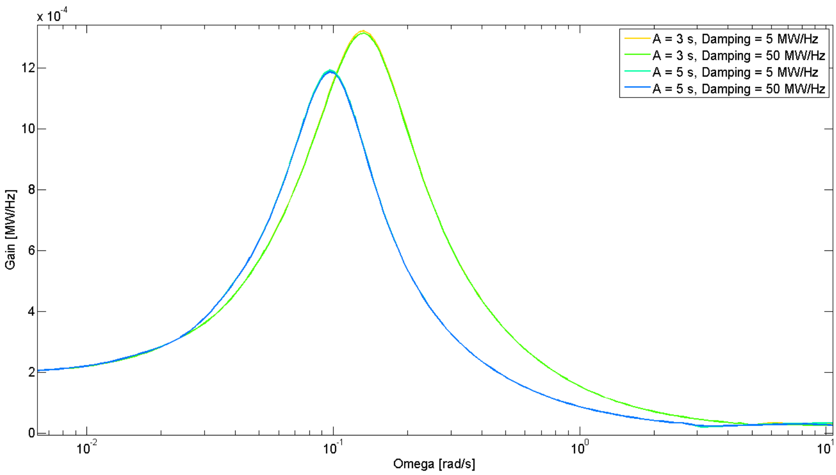

Figure 4.21: Frequency sweep of equivalent of the Nordic Synchronous Area. Load $P_L = 29008$ MW. System inertia $E_k = P_L * A$

A surprising result figures is that the effect of the inertia time constant appears to be bigger than the effect of the system loading. This would indicate that the composition of the power sytem has a bigger effect on the frequency response than the relative loading(assuming the share of the different generation types remains stable). For both systems, the maximum amplitude is excited at the same frequencies. For an inertia time constant of 3 seconds, the time period is a little under 50 seconds for both scenarios. For a time constant of 5 s, the time period is 65 seconds. These results indicate that the system has a tendency to oscillate with a period of 45-70 seconds when excited.

The figures show that a 1000% increase in load damping only produce a marginal change in the amplitude of the response. No data on the actual load damping in the Nordic grid has been found. tAs such, the numbers used here remain an educated guess, at best, and it is difficult to draw too many conclusions based on this result alone.

Chapter 5

Conclusion

The purpose of this report has been the implementation of a non-linear hydro turbine in power system analysis. The background of the work is the ongoing project into the floating frequency in the Nordic power grid, conducted by the Nordic TSOs. The work has consisted of modelling both the turbine model and a simplified one-bus, Nordic power system. The models have been analyzed using both time series simulation and frequency simulation.

The time series analysis conducted in the report can be split into three parts; model verification, reproduction of a 60 second oscillation and a time series analysis on the equivalent model of the Nordic power system. The verification process indicate that the HYGOV generator accurately represents the dynamics of the governor, penstock and turbine. The model performs well both as a standalone turbine, in comparison with commercial software, or as an alternative hydro model in KKERMIT.

The model was not able to replicate the oscillations in the power system. The goal was to tune the turbine model to be critically stable, and excite it, but this led to the system being either unstable (under damped) or stable (over damped). The results indicate that the added dynamics, making the model non-linear, cause the HYGOV model to become unstable, in comparison to existing models. The unpredictability of the model represent a possible disadvantage in power system studies.

Time series tests on the equivalent of the Nordic system indicates that the HYGOV model accurately can describe the dynamics between multiple turbines in a larger system. The test also show that the a large system can be described with a simple one-bus system represented with the mechanical dynamics, as opposed to the electrical. The model has obvious deficiencies, such as the lack of grid representation. This means that the system has no bottlenecks nor takes into account electrical (and actual) distances in the system. The time series test show that he representation is adequate for the purpose of this report.

The frequency analysis has been performed on both the turbine models and the equivalent of the Nordic system. The frequency analysis of the turbine models support the results from the attempts to replicate the frequency oscillations. The frequency sweep of the turbine show that the HYGOV model responds similar to the existing turbine model for a majority of the tests. However, the non-linarites cause it to be unstable in 7 of the cases studied. The instabilities found in this report suggests that care should be taken when using the model. For system studies in this report, the system inertia has been set to minimize chances of instability.

In the stable cases, the frequency analysis shows that the response of the model is at the highest for frequencies in the range 0.004Hz – 0.022 Hz, indicating it is susceptible to be excited at frequencies in this range. The test also show that the inertia determine this range.

A frequency sweep of the system indicate that the Nordic system is excited at frequencies in the range 0.014 Hz - 0.022 Hz (a time period of 45 – 70 seconds). This corresponds well to the frequency oscillations measured in the Nordic Synchronous area. The test also suggest that the composition of the generation in the system (the inertia time constant) has a far greater impact on the frequency response than the system load. These results suggest that increasing the inertia constant in the system could reduce the impact of the oscillations.

Chapter A

HYGOV in KERMIT

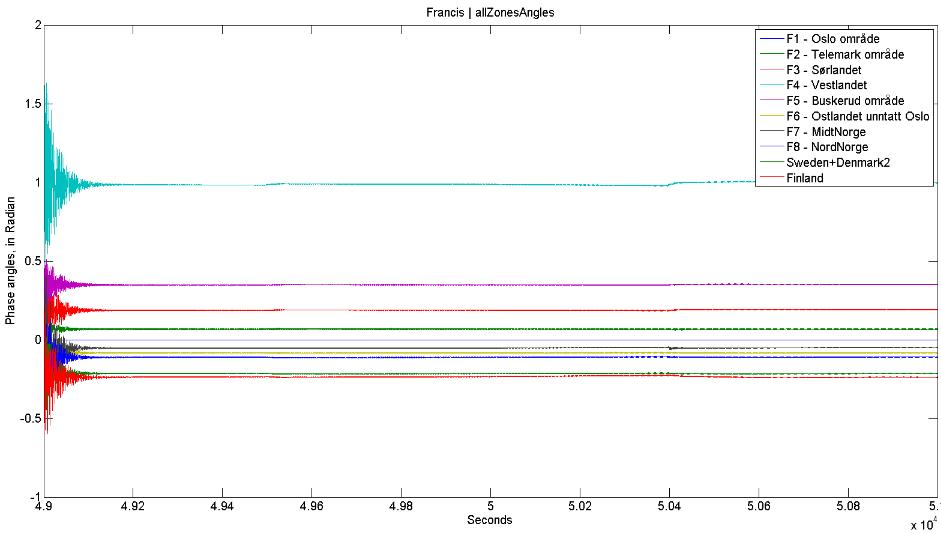

Figure A.1: Angles in each zone with original hydraulic model

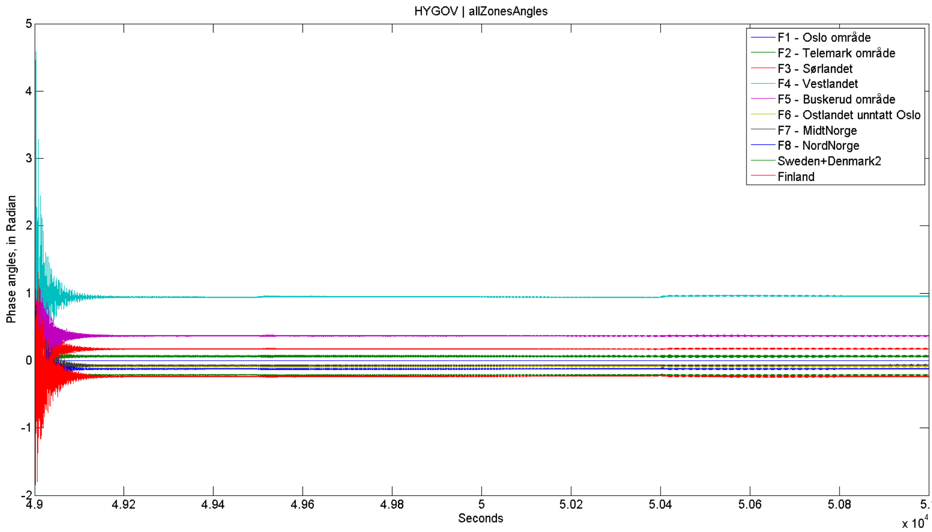

Figure A.2: Angles in each zone with HYGOV hydraulic model

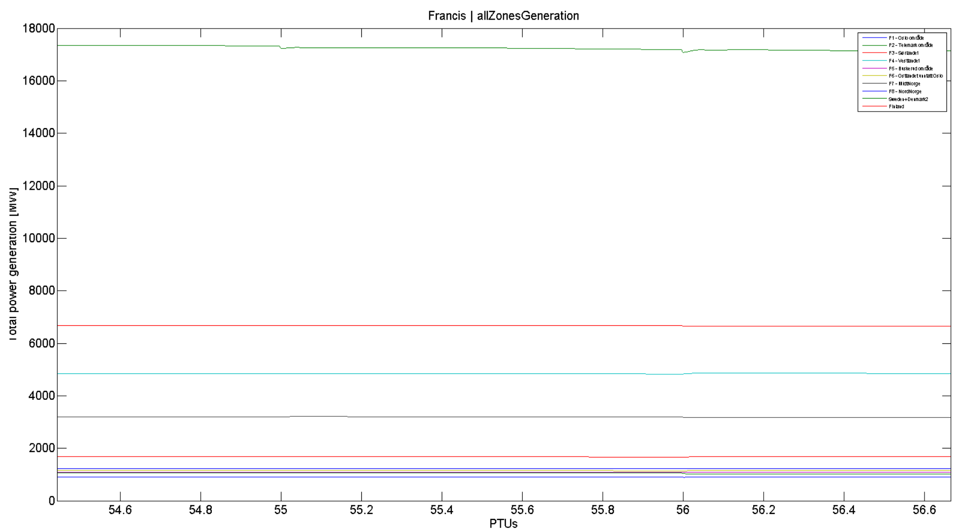

Figure A.3: Generation in each zone with original hydraulic model

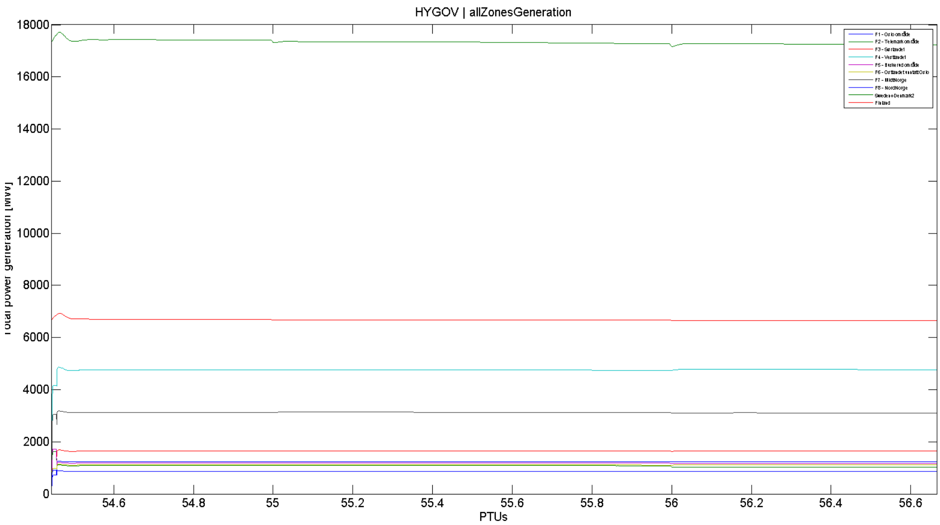

Figure A.4: Generation in each zone with HYGOVl hydraulic model

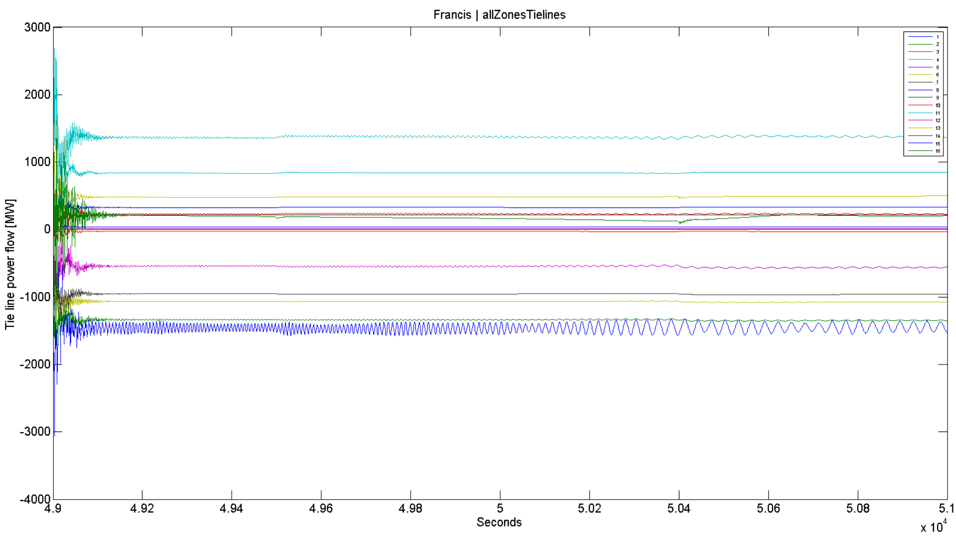

Figure A.5: Flow in each line with original hydraulic model

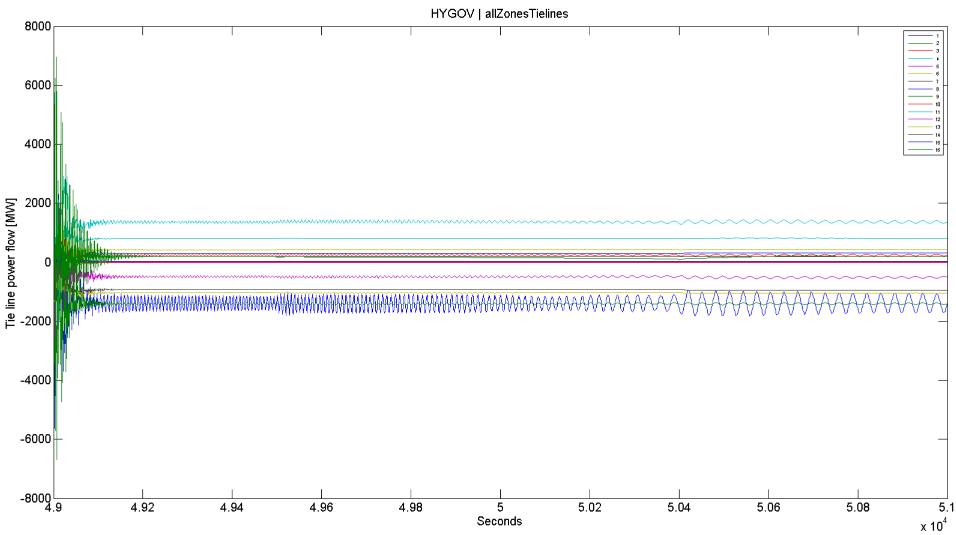

Figure A.6: Flow in each line with HYGOV hydraulic model

Chapter B
Parameter Test

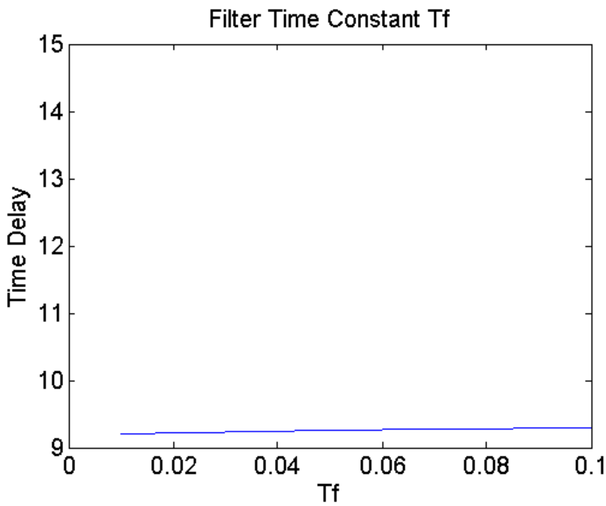

Figure B.1:

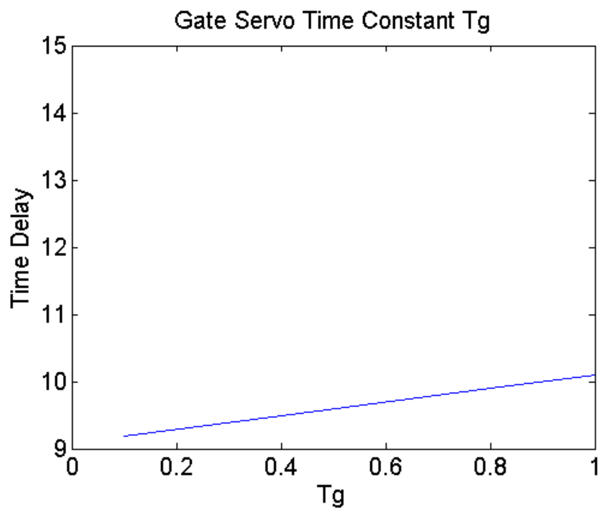

Figure B.2:

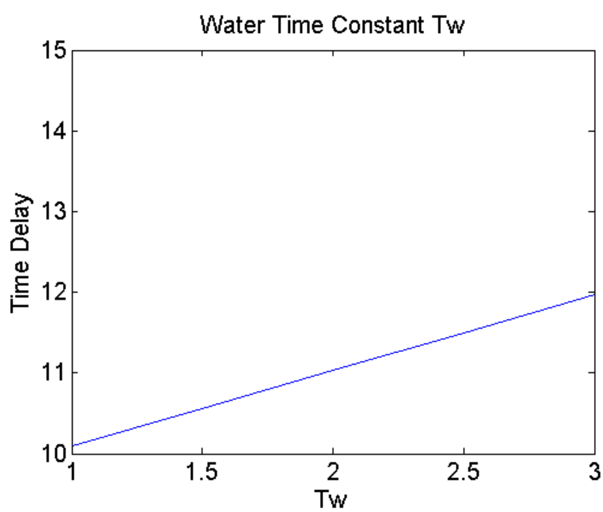

Figure B.3:

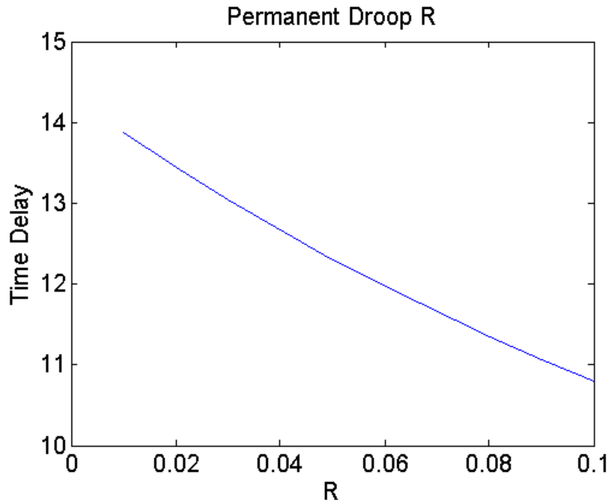

Figure B.4:

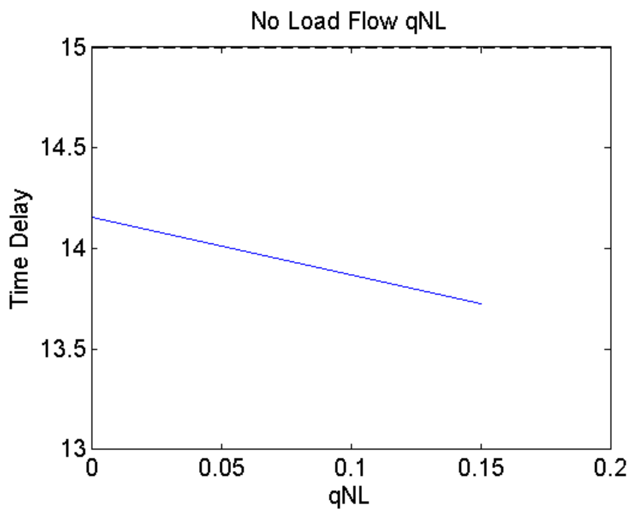

Figure B.5:

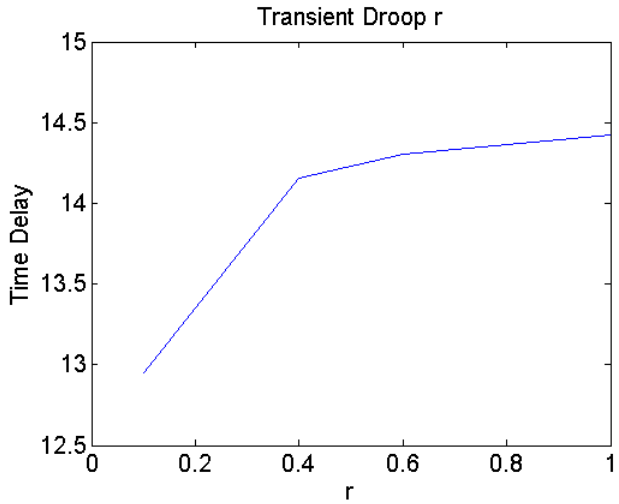

Figure B.6:

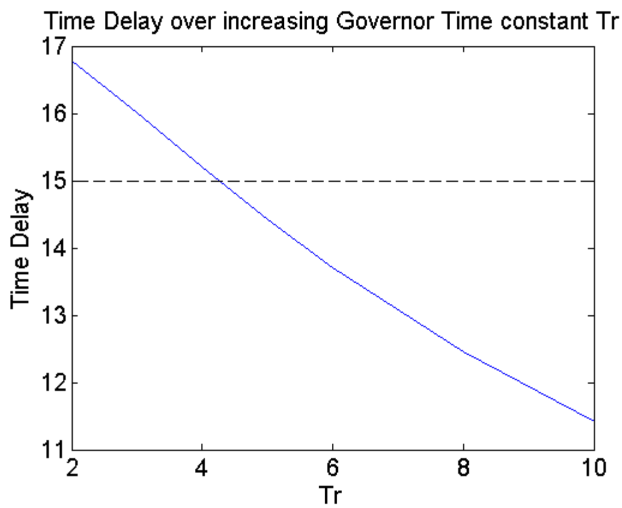

Figure B.7:

Chapter C
CIDRON Test with HYGOV

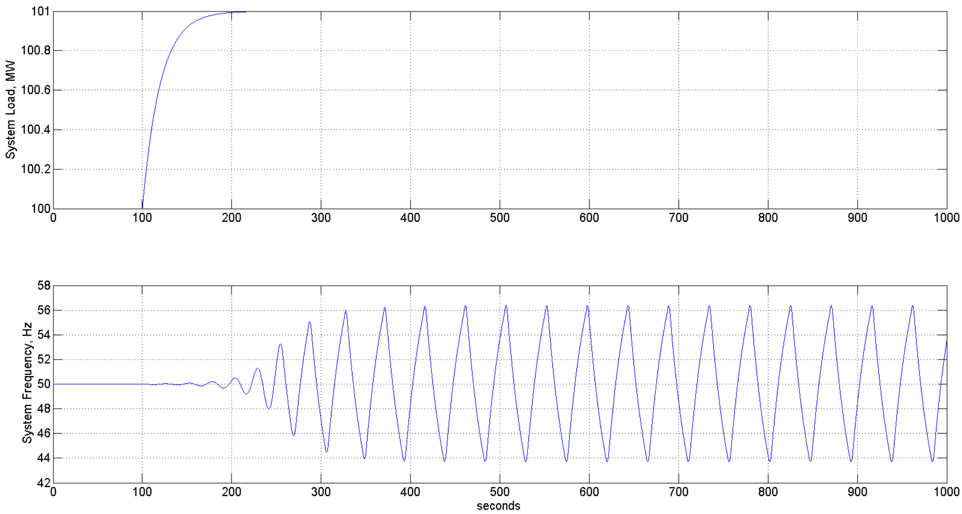

Figure C.1: The frequency responce of 1 HYGOV subjected to a 1% increase in load. 40 second oscillation expected

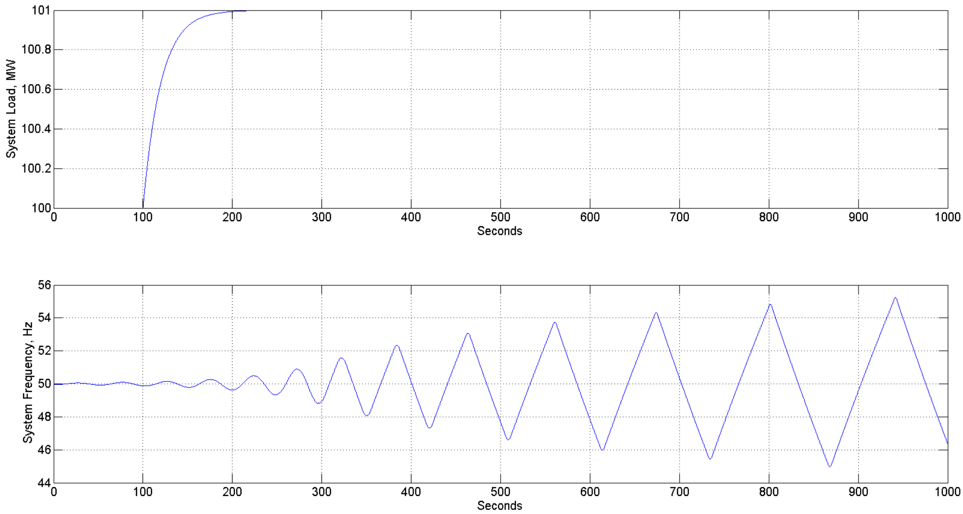

Figure C.2: The frequency responce of 1 HYGOV subjected to a 1% increase in load. 80 second oscillation expected