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# Electric hydraulic interaction

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Submission date: July 2011
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# Problem description

This Master thesis is as a continuation of a project initiated in 2005 at SINTEF Energy Research that was initiated to further develop hydropower plant models for use in power system simulation tools. In particular, the goal has been to analyze and improve models of the hydraulic side in hydro power plants equipped with Francis turbines. The main objective of this master project is to investigate the interaction between the electrical system and the hydraulic side in hydro power plants. Primarily, the project involves:

- Further development of models for the hydraulic side in hydro power plants for use in power system simulation tools, with focus on models that include turbine characteristics.

- Verification of the models. By utilizing software designed for accurate dynamic simulation of the hydraulic side of hydro power plants (LVtrans), verify the accuracy of the developed models. In addition, the accuracy of existing models of the hydraulic side found in power system simulation tools and a model suggested by IEEE is investigated.

- Investigate how the use of the different hydraulic models affects the dynamics of the electrical power system for different electrical power systems configurations and situations. Study the interaction between the power system and the hydraulic side and how the different hydraulic models represent this interaction.

# Summary


The hydraulic models representing hydro turbines and conduit system found in standard model libraries of power system analysis tools are often simplified models. Subsequently, important information about the dynamics of the hydraulic system may not be properly represented by such models, putatively resulting in insufficient representation of the interaction between the electric system and hydraulic system.

In this master thesis three different hydraulic models for hydro power plants equipped with Francis turbines for use in power system simulation software has been studied: 1) a simplified model often found in power system simulation tools; 2) a model including a surge tank and elastic water column and 3) a model that includes a surge tank, elastic water column and turbine parameters accounting for the characteristics of the hydraulic turbine.

The hydraulic models were implemented in Simpow, a power system simulation tool. A frequency scan in the range from $10^{-3}$-5 Hz was performed. The results were compared with a frequency scan from LVtrans, a program specifically designed for accurate simulation of the dynamics of the hydraulic side in hydro power plants. The comparison showed that the simplified model failed to properly represent the dynamics of the conduit system. The the model with surge tank and elastic water column was able to represent the dynamics of the conduit system with satisfactory accuracy. Best representation was achieved for the model including turbine parameters.

The three hydraulic models were implemented in three different power system configurations: a single machine infinite bus system; a system consisting of two interconnected areas; and a system that has sustained power oscillations. The resulting active power delivered from the generator were the hydraulic models was implemented, the speed of the turbine, the pressure at turbine and the flow through the turbine were investigated.

The simulation results revealed that the active power variation from the gener-

ators is in the same range for all three models, except for the simulation with sustained power oscillations. The speed variations of the turbine as a result of incidents in the electrical network are in the same range for all three models.

The model including turbine parameters is the only model able to represent the pressure variation as a result of a variation of speed of the turbine. For power oscillations with frequencies equal to the half period frequency of the water hammer effect,1.38Hz, both the model with surge tank and elastic water column and the turbine parameter model show very little response. For frequencies equal to the water hammer effect, 0.69Hz the variation in flow is also small for the two models. In general, the model with turbine parameters are better damped than the two other models.

Further work should include development of an automated routine for determining parameters to use in the model with turbine parameters as well as investigations of how the model behaves in different network configurations.

# Preface

This is my Masters thesis in Electric Power Engineering carried out at the Faculty of Information Technology, Mathematics and Electrical Engineering at the Norwegian University of Science and Technology(NTNU)

I would like to express my gratitudes to my supervisor, professor Kjetil Uhlen, Co supervisor MSc Trond Toftevaag and dr Bjørnar Svingen for assistance with this master thesis.

Ola Høydal Helle
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July 2011

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# Chapter 1

# Introduction

In 2005, a project was initiated at SINTEF Energy Research to further develop hydraulic models for use in power system simulation tools, in particular models including Francis turbines. This master thesis can be regarded as a continuation of this project.

The hydraulic models representing hydro turbines and conduit system found in standard model libraries of power system analysis tools are often simplified models. Important information about the dynamics of the hydraulic system might not be properly represented by such models. In its turn this might lead to insufficient representation of the interaction between the electric system and hydraulic system, the topic for this master thesis.

There are two main objectives for this master thesis. The primary objective is to investigate to what extent different hydraulic models for use in power system simulation tools represent the actual dynamics of the hydraulic side of a hydro power plant. The second objective is to study how the use of the different hydraulic model affect the active power delivered to the network, the speed of the turbine and the pressure and flow in the conduit system.

Three different models for the hydraulic side will be investigated: 1) A simplified model often found in power system simulation tools;2) a model with surge tank and elastic water column suggested by an IEEE work group and 3) a model with surge tank, elastic water column and turbine parameters accounting for the characteristics of the hydraulic turbine.

The accuracy of the three hydraulic models will be investigated. LVtrans is a program specifically designed for accurate simulation of the dynamics of the hydraulics in a hydro power plant. LVtrans AFF module will be utilized for performing a frequency scan of the hydraulic system. Simpow will be used for performing frequency scans of the three hydraulic model. The frequency scan from LVtrans and Simpow will be compared. Since the model with turbine parameters is an experimental model additional improvements might be necessary before satisfactory accuracy is achieved. Parts from this section of the master thesis have been published in relation to Norsk Elektroteknisk Forening's (Norwegian association of electrical engineering) 100 year anniversary, [2].

The three hydraulic models will be implemented in three different power system configurations; an infinite bus single line system, a system consisting of two interconnected areas and a system that has sustained power oscillations. The active power delivered power from the generator, the speed of the turbine, the pressure at turbine and the flow through the turbine will be investigated for generators where the different hydraulic models are utilized.

It is important to note that some parts of this report are also included in a project carried out in the autumn semester 2010. In particular, parts of the theory chapter, the chapter describing the software used for simulation and the description of the constants and parameters resemble similar chapters found in the project work.

# Chapter 2

# Theory

## 2.1 Hydraulic Equations

### 2.1.1 Hydraulic pressure

Pressure is the force per area and is defined as:

$$p = \rho g h \tag{2.1}$$

Where $p$ - Pressure [$N/\text{m}^2$ ], $\rho$ - Density of the water [kg /m$^3$], $g$ - acceleration of gravity [m/s$^2$ ], $h$ - Height [m]. It should be pointed out that in hydro power application the term *head* is often used instead of pressure.

### 2.1.2 Hydraulic Power

The hydraulic power is the product of the volumetric flow, $q$, and the available pressure, $p$:

$$P = q * p \tag{2.2}$$

Where P - hydraulic power [W], q - volumetric flow [m$^3$/s ] and p - hydraulic pressure [$N/\text{m}^2$ ].

### 2.1.3 The energy equation

.

The energy equation is often referred to as the Bernoulli equation after its originator. The equation simply states that the energy along a flow line is constant and can be contained as kinetic energy, velocity, or as potential energy, either as pressure or as height above a reference point[3].

$$\frac{1}{2}\rho v_1^2 + \rho g z_1 + p_1 = \frac{1}{2}\rho v_2^2 + \rho g z_2 + p_2 \tag{2.3}$$

Where $\rho$ - density of the fluid [kg /m$^3$], $v$ - average speed of the fluid [m/s ], $g$ - acceleration of gravity [m/s$^2$ ], $z$ - height [$m$] and $p$ represent the pressure, see equation 2.1.1.

### 2.1.4 Hydraulic losses

#### Frictional losses

The general steady state friction loss for flow in a filled duct can be expressed as [3]:

$$h_f = f\frac{L}{d}\frac{V^2}{2g} \tag{2.4}$$

By using the relationship between flow, area and average speed, flow=area*speed, the head loss can be stated as:

$$h_f = f\frac{L}{d^5}\frac{Q^2}{0.125 g\pi^2} \tag{2.5}$$

Where $h_f$ - Head loss [m], $f$ - Friction factor [-], $L$ - Length of the duct[$m$], $Q$ - Volumetric flow [m$^3$/s ], $d$ - Hydraulic diameter of the duct [m], and $g$ - the acceleration of gravity [m/s$^2$ ]. The numerical value of the friction factor, $f$, is a function of the relative roughness of the pipe, duct shape and the Reynolds number. The Reynolds number is a dimensionless number describing the relationship between inertial and viscous forces and is given by:

$$Re = \frac{\rho V D}{\mu} \tag{2.6}$$

where $\mu$ - dynamic viscosity of the fluid, [$N$s /m$^2$], $\rho$, V and D as in (2.4).

The Reynolds number is useful for separating between laminar and turbulent flow regimes. Laminar flow is smooth and steady while turbulent flow is fluctuating and agitated. For a circular pipe turbulent flow is found when the Reynolds number exceeds 4000, meaning that the flow in a hydro power plant is usually turbulent. The generally accepted relationship [3] between the relative roughness, Reynolds number and the friction factor can be stated as:

$$\frac{1}{f^{1/2}} = -2.0log\left(\frac{\epsilon/d}{3.7} + \frac{2.51}{Ref^{1/2}}\right) \tag{2.7}$$

Where $\epsilon$ is the roughness of the pipe [mm], f,d,Re as in (2.6).

#### Singular losses

Singular losses are related to sudden change in the flow. In a hydro power plant such losses occur at valves, gates, bends, expansions, contractions and trash racks. The flow in such components are highly complex and therefore calculation of the occurring loss is seldom done. However the losses can be estimated by [3]:

$$h_{loss} = K\frac{V^2}{2g} \tag{2.8}$$

Where $h_{loss}$ - Head loss [m], K-Dimensionless coefficient [-], $V$ - Mean velocity through the component [m/s ] and $g$ is the acceleration of gravity [m/s$^2$ ]. The dimensionless coefficient, K, is usually found by experiments or Computational Fluid Dynamics, CFD.

### 2.1.5 Equation of continuity

The equation of continuity gives the relationship between pressure and volume. For an infinitesimal sloping pipe section the equation is given as [4]:

$$\frac{\partial H}{\partial t} + \frac{a^2}{g}\frac{\partial v}{\partial x} = 0 \tag{2.9}$$

Where $\partial H \partial t$ is the change in pressure height per time unit. The pressure height consist of the actual pressure given as meter water column plus the height above a reference point. $a$ - Pressure wave propagation speed [m/s ], $g$ - acceleration of gravity [m/s$^2$ ], $\partial V/\partial t$ change of the volume of the infinitesimal pipe section.

A full deduction can be found in [4] and [3]. The understanding of the continuity equation is important when analyzing the dynamics of a hydro power plant. Since the $a^2/g$ relationship has a large numerical value, even a small change in the volume will give a large change in pressure at the point where the volume is changed. Since the pressure at the point where volumed is changed is different from the surrounding a pressure wave will occur. This is often referred to as the water hammer effect.

### 2.1.6 The momentum equation

For a flow in a sloping infinitesimal pipe section the momentum equation can be expressed as [4],[3]:

$$g\frac{\partial H}{\partial x} + \frac{\partial v}{\partial t} + \lambda\frac{v|v|}{D} = 0 \tag{2.10}$$

Where $\lambda$ is the friction factor, see equation 2.4, and $d$ is the diameter of the pipe. $H, v, g$ is the same is in equation 2.9. A full deduction of the momentum equation is given in [4].

### 2.1.7 Surge tank natural period

The surge tank natural period is the period of the oscillations between the surge tank and the reservoir. The surge tank natural period is defined as [5]:

$$T_s = 2\pi\sqrt{\frac{A_s L_t}{A_t g}} \tag{2.11}$$

Where $A_s$ - Surge tank cross sectional area [m$^2$ ], $L_t$ - Length of the tunnel [m], $A_t$ - Tunnel cross sectional area [$m$], g - acceleration of gravity [m/s$^2$ ]

### 2.1.8 Stability criterium of surge tank-reservoir oscillation

During operation of a hydro power plant a small change in the guide vane position will lead to a change in the flow. As a result the surge tank level will change, resulting in a pressure change in the tunnel, 2.1 which will lead to a change in the flow of the tunnel. Due to the inertia of the water column a resulting oscillation between the reservoir and the surge shaft will occur. The cross sectional area of the surge tank determines the variation the the surge level and again the pressure in tunnel. If the energy added to the system by the pressure rise is larger than the energy loss due to friction the oscillations will be unstable. This minimum cross section was first determined by Thoma, and is given by [4]:

$$A_{th} = \frac{LA}{2g\alpha(h_0 - z_o)} \tag{2.12}$$

Where $A_th$ - Thoma cross section [m$^2$ ], $L$ - Length of the tunnel [m], $A$ - Cross sectional of the tunnel [m$^2$ ], $g$ - Acceleration of gravity [m/s$^2$ ], $h_0$ - Head of the power plant [m], $z_0$ - resulting fall in surge tank level due to friction [m]. $\alpha$ is defined as f*LD/2g, see equation 2.4.

## 2.2 The Turbine

### 2.2.1 Efficiency curve

A principle sketch of a Francis efficiency curve is shown in figure 2.1. The efficiency curve is often to referred to as a hill chart due to the similarity to contour line representation of hills found in maps. Hill charts of actual turbines are often constructed from measurements from scaled down version of the actual turbines. The reason for this is that accurate mathematical modeling and calculation is difficult and time consuming due to the great complexity of the flow through the turbine. However less accurate hill charts can be constructed by simplifying the mathematical flow modeling.

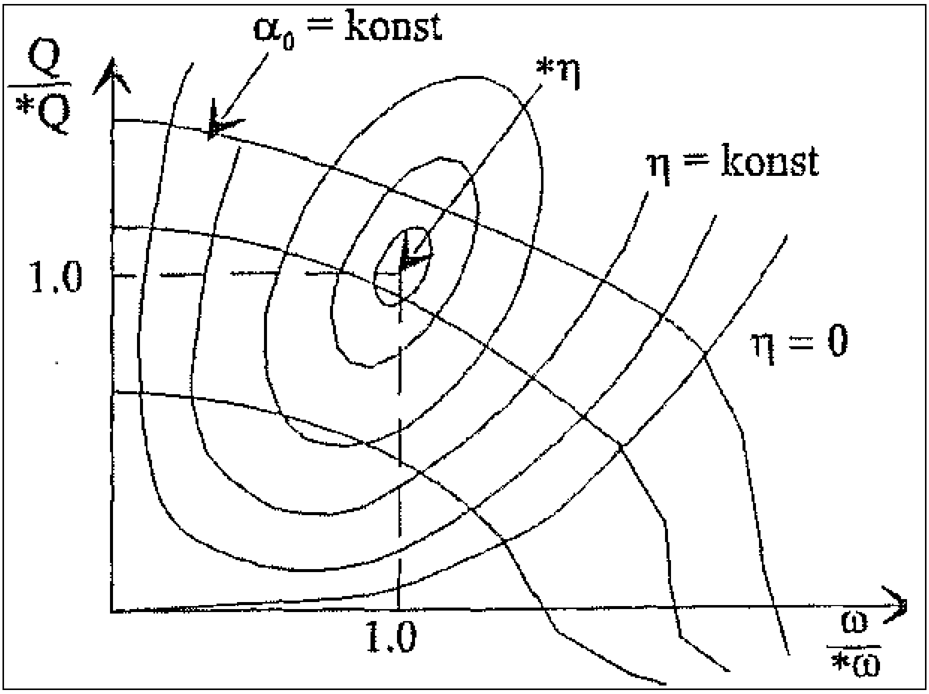


Figure 2.1: Francis characteristic diagram, [1], p. 33

As seen from figure 2.1, the characteristics diagram shows the relationship between flow, Q, speed, $\omega$ and efficiency, $\eta$ [1] . The *Q and *$\omega$ denotes the flow and speed at the best efficiency point, *$\eta$. The sloping line, $\alpha_0$ represents the behavior of the turbine for a constant guide vane opening, see chapter 2.2. Hill charts are a useful tool when analyzing a hydro power turbine since information about the flow, speed and efficiency can be extracted.

### 2.2.2 Turbine self governing

In a Francis turbine the volumetric flow is influenced by the rotational speed [4]. This is referred to as the turbine self-regulation characteristic. For a high head Francis turbine the volumetric flow is usuallyt decreasing when the rotational speed is increasing, meaning that the turbine itself has a positive influence on the stability. Turbines with such characteristics are said to have a positive self regulation. If a hill chart is available the self regulating effect can be observed by inspection of the constant guide vane lines. Figure 2.1 shows a turbine with a positive self regulation. For a low head Francis turbine the positive self regulation effect is smaller than for a high head Francis, and in some cases it may be negative, meaning that the volumetric flow is increasing when the speed is increasing.

### 2.2.3 The linearized turbine model

As seen in 2.2.1 the relationship between flow, speed, and efficiency is complex. However the change in power, volumetric flow and torque due to a change in speed, gate opening or head may be estimated by the relationship between power, flow, torque, speed, head and gate opening [6], [7]:

$$dq = \frac{\partial q}{\partial n} n + \frac{\partial q}{\partial y} y + \frac{\partial q}{\partial h} h \tag{2.13}$$

$$dm = \frac{\partial m}{\partial n} n + \frac{\partial m}{\partial y} y + \frac{\partial m}{\partial h} h \tag{2.14}$$

$$dp = \frac{\partial p}{\partial n} n + \frac{\partial p}{\partial y} y + \frac{\partial p}{\partial h} h \tag{2.15}$$

Where q - volumetric flow, y - gate opening, n - speed, h - head at the turbine, m - torque and p - power output, mechanical. All parameters given with per unit, p.u. denominations. For a high degree of accuracy the numerical value of the partial derivatives should be changed according to the actual operating point and the actual turbine. The numerical values of the partial derivatives can be obtained from hill charts [8] or field measurements.

## 2.3 Basic elements and constants

### 2.3.1 Water starting time, $T_w$

The water starting time is the time it takes to accelerate the water column from zero to rated flow by the rated head, neglecting friction. $T_w$ is defined as [7]:

$$T_w = \frac{L}{gA}\frac{Q_0}{h_0} \tag{2.16}$$

Where $L$ - Length [m], $A$ - Cross sectional area of duct[m$^2$ ], $g$ - Acceleration of gravity [m/s$^2$ ], $Q_0$ - Rated volumetric flow [m$^3$/s ], and $h_0$ - Rated head [m]. For a duct with varying cross section the sum of the $L/A$ relationship should be used.

### 2.3.2 Penstock elastic time constant, $T_e$

$T_e$ is defined as the time it takes for a pressure wave to travel from the turbine to the nearest surface with atmosperic pressure, usually the surge shaft or reservoir. $T_e$ is defined as [9]:

$$T_e = \frac{L}{a} \tag{2.17}$$

Where $L$ -Length to nearest atmospheric pressure [m], $a$ - Pressure wave travel velocity [m/s ].

### 2.3.3 Turbine modeled as a simple valve

As seen from the Bernoulli equation, 2.3 the energy along a flow line is constant. By setting the altitude at the turbine equal to zero and the velocity of the water to zero at the inlet and rewriting the velocity at the turbine as the volumetric flow divided by the gate opening,G, the Bernoulli equation become:

$$\frac{q^2}{G} = h \tag{2.18}$$

or

$$q = G\sqrt{h} \tag{2.19}$$

### 2.3.4 Accounting for the total efficiency of the turbine, qnl

As stated in chapter 2.1.2 the hydraulic power is the product of the available pressure and the volumetric flow. To account for the turbine losses the IEEE models subtract a part of the flow through the turbine when calculating the hydraulic power. The subtracted value is denoted by $Q_{nl}$ and is given by:

$$Q_{nl} = 1 - \eta_h * \eta_{mech} \tag{2.20}$$

Where $\eta_h$ and $\eta_{mech}$ are the hydraulic and mechanical efficiency of the turbine.

It should be pointed out that the qnl only compensate for the steady state efficiency.

### 2.3.5 Storage constant of surge tank, $C_s$

The storage constant of the surge tank $C_s$ is defined as [5]:

$$C_s = \frac{A_s * h_b ase}{q_b ase} \tag{2.21}$$

Where $C_s$ is given in second, $A_s$ is the cross sectional area of the surge chamber [m$^2$ ], $h_{base}$ is the head in meters, and $q_{base}$ is the flow

The storage constant gives the inverse pressure increase per second, given in per unit.

### 2.3.6 Surge impedance of the penstock, $Z_0$

The surge impedance is defined as[5]:

$$Z_0 = \frac{q_{base}}{hbase} \frac{1}{\sqrt{g\alpha}} \tag{2.22}$$

Where $Z_0$ - surge impedance [-], $q_{base}$ - Volumetric flow at rated head [m$^3$/s ], $h_{base}$ - rated head - [m], $g$ - acceleration of gravitiy [m/s$^2$ ], $\alpha$ is defined in chapter 2.1.6.

A more understandable expression for $Z_0$ by using the definition of the water time starting constant $T_w$, expression for the wave velocity and wave propagation speed:

$$T_w = Z_0 T_e \tag{2.23}$$

or

$$Z_0 = \frac{T_w}{T_e} \tag{2.24}$$

Inspection of the equation can give good understanding of the phenomena involved. $T_w$ can be regarded as an inertia affected by a force, giving an acceleration. As described in 2.1.6 the $T_e$ can be regarded as the time before a change in the flow conditions at the turbine is affecting the flow at the surge tank or reservoir. Since the change in flow conditions is delayed, the flow conditions at the start of the penstock will not be affected by a change in the flow conditions at the turbine for a time period $T_e$. This implies that when the flow is decreased the pressure at the turbine will increase since the flow into the penstock is larger than the flow out from the penstock. This is due to the delayed information of the flow conditions at the start of the penstock, as expressed by the equation of continuity, see 2.1.5. Likewise an increase in the gate opening will instantaneously decrease the pressure at the turbine.

### 2.3.7 Dynamic representation of the elastic water colum

The transfer function from head to the resulting flow can be written as [5]:

$$\frac{h(s)}{q(s)} = \frac{Z_0(1 - e^{-2T_e s})}{-2T_e s} \tag{2.25}$$

or rewritten as [10]:

$$\frac{h(s)}{q(s)} = tanh(T_e s) \tag{2.26}$$

Where $Z_0$ is the surge impedance, and $Te$ is the pressure wave traveling time.

### 2.3.8 Frictional factor,$f$, damping factor, $D$ and proportional factor, $A_t$

To account for frictional losses in the head race tunnel, penstock and surge tank the IEEE models use the general equation for frictional losses related to the volumetric flow, equation 2.5. However a conversion from absolute values to per unit values are required.

The damping factor,$D$, gives the per unit damping in cases where the turbine is running at off nominal speed. The proportional factor $A_t$ converts the per unit power of the turbine to per unit power of the generator.

# Chapter 3

# Hydraulic models

## 3.1 The classical model

By utilizing the different elements and constants presented in 2.3 different models can be constructed. A model often found in power system simulation tool is the classical model depicted in figure 3.1. [7]

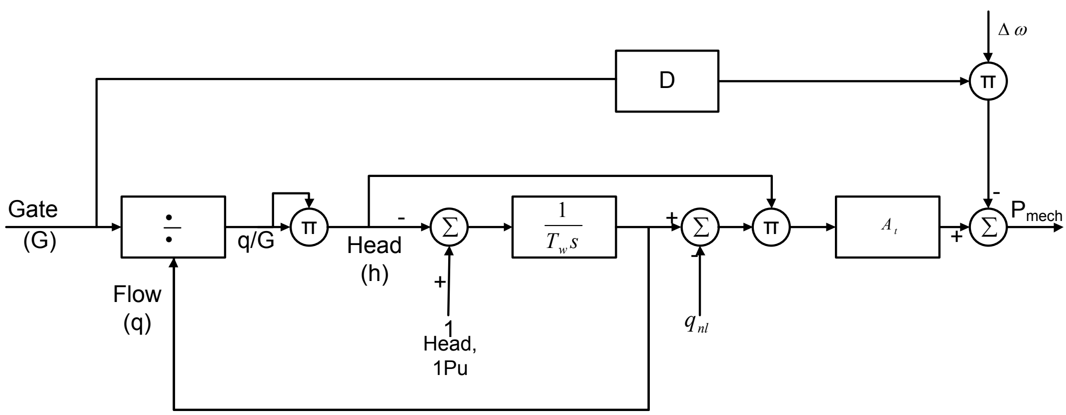


Figure 3.1: The classical model

The figure shows a conduit system modeled with an inelastic water column and no surge tank. The model has two input parameters, the gate position and the difference between nominal speed and actual speed, $\Delta\omega$. The model has one output parameter, mechanical power.

The model uses the relationship between gate and flow to calculate the available head. 1 per unit is subtracted from the calculated head, giving the excess head available for changing the volumetric flow. The volumetric flow is calculated by using the inverse of the water starting time $T_w$. $Q_{nl}$ is subtracted from the flow to account for the total efficiency of the turbine. The resulting available flow is multiplied by the previously calculated head to find the mechanical power. The mechanical power is multiplied by $A_t$. The proportional factor $A_t$ converts the per unit power of the turbine to per unit power of the generator. If the turbine is running at off nominal speed, power will be added or subtracted from the output mechanical power. The reduction of increase of output mechanical power depends on the gate position, a damping factor, D, and $\Delta\omega$.

## 3.2 Model with surge tank and elastic water column, The IEEE model

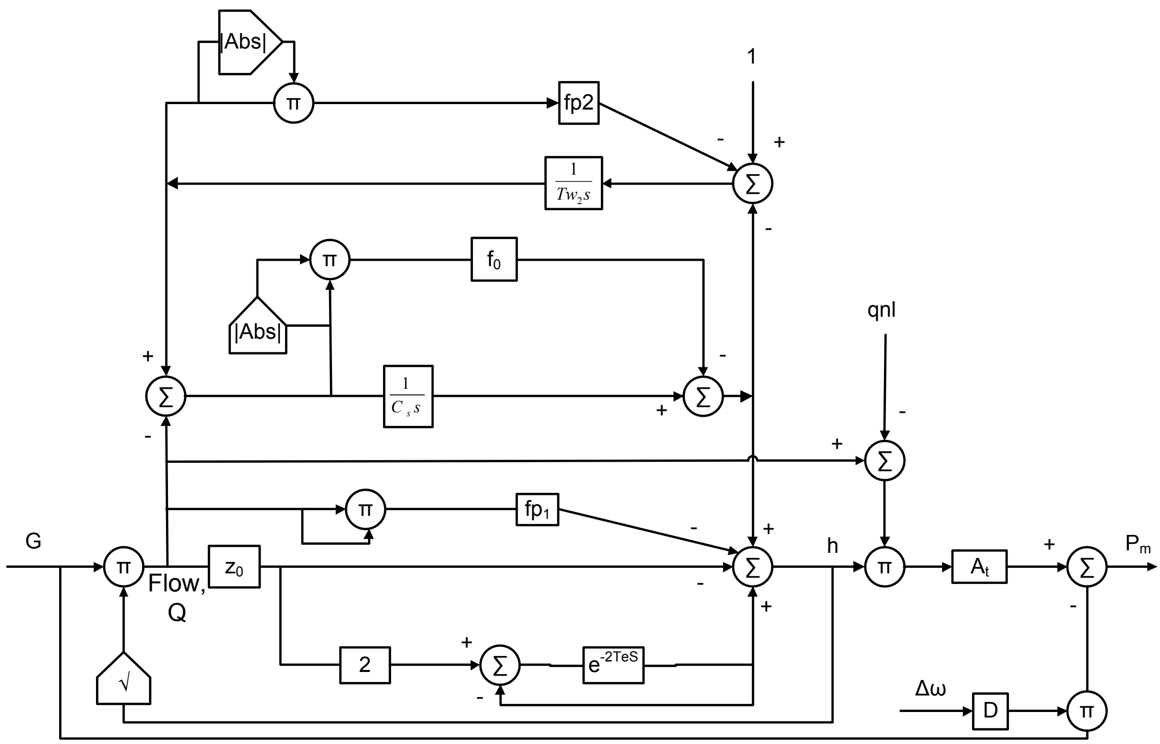


Figure 3.2: Model with surge tank and elastic water column, The IEEE model

The model is proposed by an IEEE workgroup [5] and depicted in figure 3.2. The model includes a surge tank and elastic water column, often referred to as the water hammer effect. As for the classical model two input parameters are required, the gate position and the difference between nominal and actual speed, $\Delta\omega$ and the only output parameter is mechanical power.

The relationship between gate position and head is used for calculating the flow in the penstock. By using the flow in the penstock, the dynamic representation of the water column in the penstock and the friction factor, the penstocks contribution to the total available head at the turbine is calculated.

The flow in the penstock is fed into a summation point representing the flow conditions at the inlet to the surge tank. By utilizing the flow conditions at the surge tank and the storage constant of the surge tank, $C_s$ the resulting head in the surge tank is found. The head in the surge tank is fed into the summation point for the total available head at the turbine. It should be pointed out that the head in the surge tank is the relative head compared to the head of the penstock.

The flow in overhead tunnel is found by using the total available head, the surge tank head and the water starting time of the tunnel. The resulting flow is fed into the summation point for the flow condition by the inlet of the surge tank.

The $Q_{nl}$ is subtracted from the actual flow in the penstock and multiplied with the available head at the turbine to find the mechanical power. The proportional factor $A_t$ converts the per unit power of the turbine to per unit power of the generator. If the turbine is running at off nominal speed power will be added or subtracted from the output power. The reduction of increase of output mechanical power depends on the gate position, a damping factor, D, and $\Delta\omega$.

## 3.3 Model with surge tank, elastic water colum and turbine parameters, The turbine parameter model

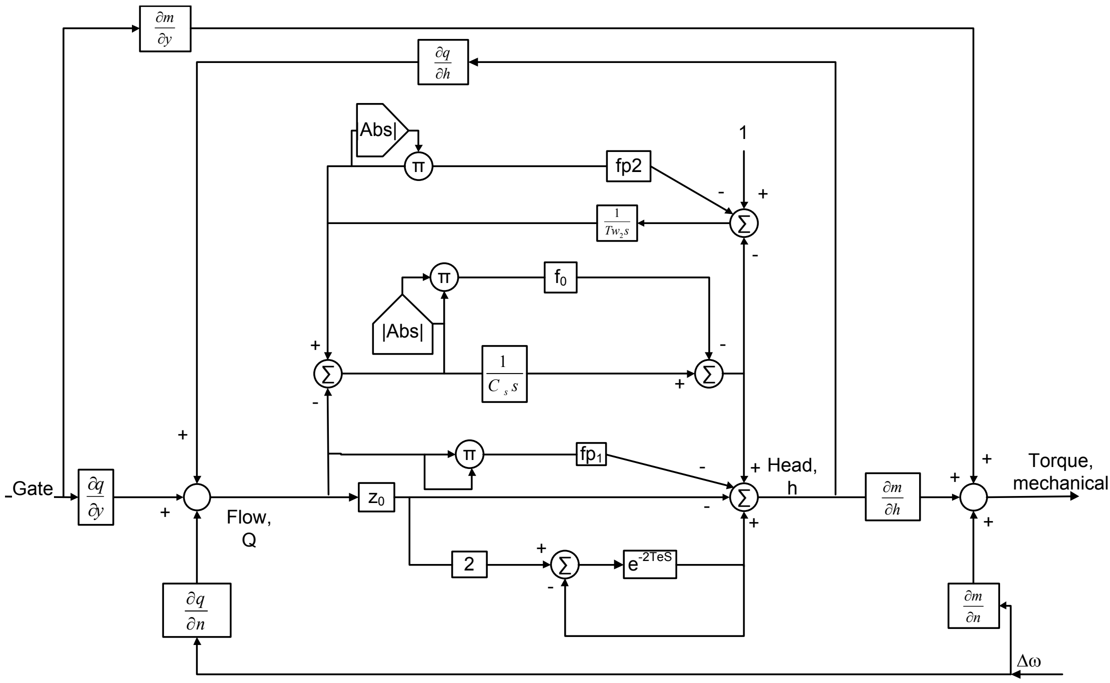


Figure 3.3: Model with surge tank and elastic water column and turbine parameters, The turbine parameter model

The model utilizes the linearized turbine model in section 2.2.3 and parts from the IEEE model. Specifically, the part from the flow to head is taken from the IEEE model. The turbine parameters are connected to the model with connection points according to the different turbine parameters. The result is shown in figure 3.3. As for the two other models the model requires two input parameters, gate position and $\Delta\omega$ and gives one output, mechanical power.

Since the turbine parameter $\partial q/\partial n$ is connected to the model via a feedback loop from the resulting speed of the turbine an additional module giving a speed deviation during the course of simulation is required. The module is only utilized during the frequency scanning of the model. When the models are implemented in power systems the additional model is no longer necessary since the variation in speed is covered by Simpows built in library. The module is found in [7].

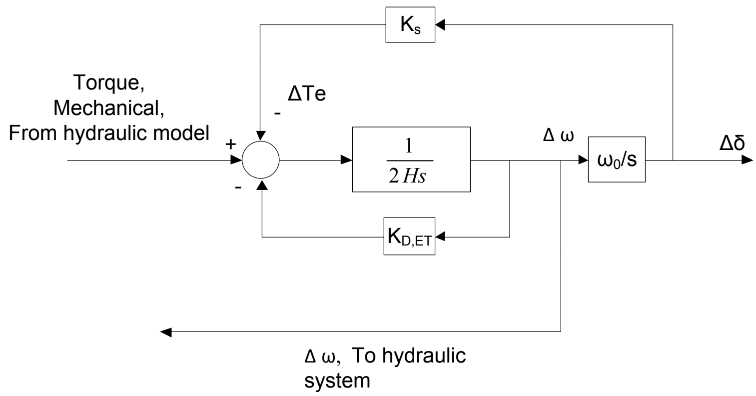


Figure 3.4: Model for simulating speed variation

# Chapter 4

# Software used in Simulations

## 4.1 Labview Transient, Lvtrans

Lvtrans, Lab View transient pipe analysis, is a simulation and calculation tool for liquid filled ducts [11]. The program is applicable to any liquid filled system, but the program was originally developed for studying the dynamics of hydro power plants. The program contains a broad range of components found in hydro power plants. This allows the user to create models of specific power plants by using the built in graphical user interface, GUI. The source code of LVtrans is object orientated, meaning that changes can be done to a specific set up of a plant without changing the source code of the program.

The program relies on the method of characteristic for solving the equations. The method allows a system of partial differential equations, PDE to be solved fast and accurate. The transients due to the water hammer effect described in chapter 2.1.5 are thus included in the solution, meaning that the program is able to provide an accurate solution of the actual systems.

The version number used in this master thesis is LVtrans86 1.31.T. This version includes a module that makes it possible to perform a frequency scan of a system created with the program's GUI.

## 4.2 Simpow

Simpow(r) is a software for simulation of power systems developed by STRI AB [12]. The software enables the user to simulate power systems in the steady state, time- and frequency domain. A built in library simplifies the process of modeling systems. The built in Dynamic Simulation Language, DSL, enables the user to define components and models. The user defined components and models can interact with components from the internal library during simulation. Simulations entirely based on user defined components are also possible. Such simulations can be set up by altering the DSL files to run independently from the internal library. This enables the user to extract data and study relationships that are not included in the standard library.

Tools for viewing, extracting and plotting results are provided. For analysis in the frequency domain the software contains helpful tools. The frequency scanning module enables the user to study the small signal stability by exiting the system with a sinusoidal signal in the desired frequency range. The amplitude of the perturbation can be selected. Studies of modal swings are simplified by a built in package. A tool for extraction and plotting eigenvalues are also provided.

# Chapter 5

# Simulation and verification of the hydraulic models

## 5.1 Scope and method

The scope of the simulations and verification was to eludicate how accurate the hydraulic models described in chapter 3 represent the hydraulic conditions in a hydro power plant. Lvtrans is believed to represent the hydraulic conditions with a high degree of accuracy. A comparison between Lvtrans result and results from simulations of the hydraulic models was used to verify the accuracy of the different hydraulic models. Simpow will be used to simulate the hydraulic models. The relationship between the guide vane opening and the resulting pressure at the turbine was simulated. The reason for selecting this relationship is the close relation between the pressure at the turbine and the available mechanical shaft power, see section 2.1.2. The shaft is where the transfer of power, or interaction, between the hydraulic and electrical system takes place. In addition the relationship is easily extracted from both programs.

The simulations study the relationship between a guide vane perturbation and the resulting pressure at the turbine. This relationship is often referred to as the head/gate relationship. The perturbation of the guide vanes was simulated for the frequency range $10^{-3}$ - 5 Hz, with a perturbation size of 0.01 per unit corresponding to a 1% change in the guide vane position. Both the magnitude and phase response were investigated. Lvtrans' built in AFF module was used to extract data from LVtrans. The hydraulic models were implemented in Simpow as independent systems by using Simpow's dynamic simulation language, dsl.

The implementation in Simpow is a continuation of the work of a former master student at NTNU, A. Lucero [13].

## 5.2 Parameters for the hydro power plant used in simulations

As described in section 3 three were simulated, one of them containing turbine parameters, see section 2.2.3. Hydraulic turbines tend to have different characteristics as turbines are adapted to a specific height and flow dependent on the geographical location and the amount of water available. This master thesis uses turbine parameters from a specific turbine and therefore data for the corresponding hydro power plant where the turbine is located are used. The data are summarized in table 5.1.

In addition to these values frictional factors are needed, see 3. The steady state friction factor for the tunnel is set to 0,06 according to [14] and 0,01 for the penstock and 0,05 for inflow in surge tank according to [3]. The frictional factors used in Simpow are in per unit, p.u, so the head loss at nominal flow was found by equation 2.5 and converted to per unit. The wave propagation speed is also needed, typical values are in the range from 1000m/s to 1200m/s , [5], [9], [4] . The wave propagation speed is set to 1200 m/s  Summary of input parameters the values is found in table 5.1.

## 5.3 Simulations results

Simpow was used to simulate the three hydraulic models described in chapter 3. It should be pointed out that the simulation results from Simpow have been edited before plotting. Data for the magnitude response has been converted from absolute values to desibel, db. By inspecting the simulation results it was discovered that data from the simulation of the phase response can be misleading. When the phase response has a lag greater than 180 degrees Simpow adds 360 degrees to the phase response so that the resulting phase response is leading by 180 degrees. Therefore some of the simulation results are shown both with an edited phase response as well as the original phase response from Simpow.

Table 5.1: Input values used in simulations

| Parameter | Value |
|---|---|
| Rated Head | 200 m |
| Rated flow | 25 $m^3/s$ |
| Pressure wave propagation speed | 1200 m/s |
| | |
| **Headrace tunnel** | |
| Length of tunnel | 1307 m |
| Diameter | 3,9 m |
| Cross sectional area | 11,95 $m^2$ |
| Steady state frictional loss factor | 0,06 - |
| Friction factor, used in Simpow, | 0,02244 |
| Water starting time, $T_w$ | 1,394 s |
| | |
| **Surge tank** | |
| Surge tank diamenter | 5 m |
| Surge tank cross section | 19,63 $m^2$ |
| Thoma cross section | |
| Inflow loss factor | 0,5 - |
| Inflow loss factor, used in Simpow | 0,012 |
| Storage constant of surge tank, $C_s$ | 157,04 s |
| | |
| **Penstock** | |
| Length | 435 m |
| Diameter | 2 m |
| Cross sectional area | 3,14 $m^2$ |
| Steady state frictional loss factor | 0,01 - |
| Friction factor, used in Simpow, | 0,0351 p.u |
| Water starting time, $T_w$ | 1.764 s |
| Penstock elastic time constant, $T_e$ | 0,3625 s |
| Surge impendance of penstock, $Z_0$ | 4,866 - |

### 5.3.1 Lvtrans results

A model of a power plant with the input values as in table 5.1 was implemented in LVtrans. In addition to these values some values needed in Lvtrans simulations was found by using the complimentary software spread sheet. Figure 5.1 and 5.2 show the simulation result for the frequency range $10^{-3}$-1Hz and 1-5Hz respectively.

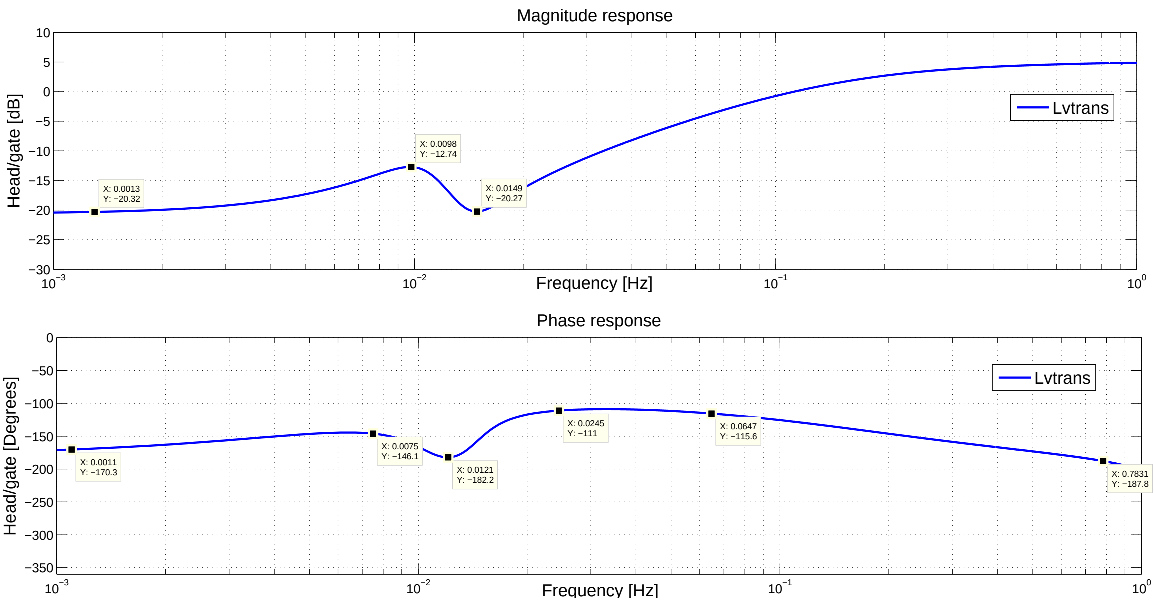


Figure 5.1: LVtrans simulation results $10^{-3}$ -1Hz

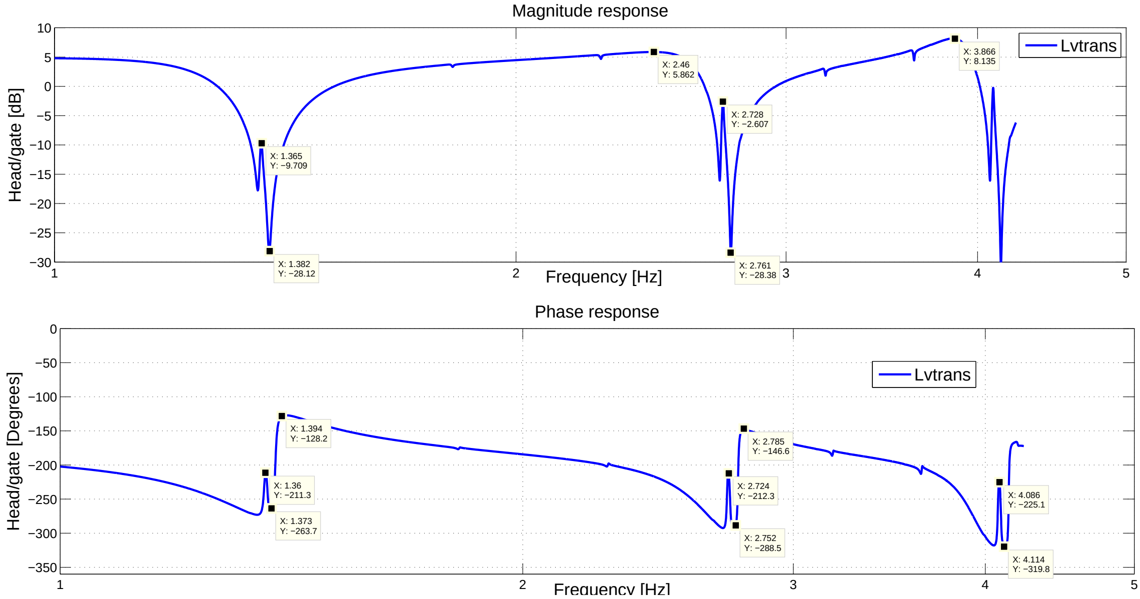


Figure 5.2: LVtrans simulation results 1-5Hz

### 5.3.2 Comparison between LVtrans results, the classical model and the model with surge tank and elastic water column

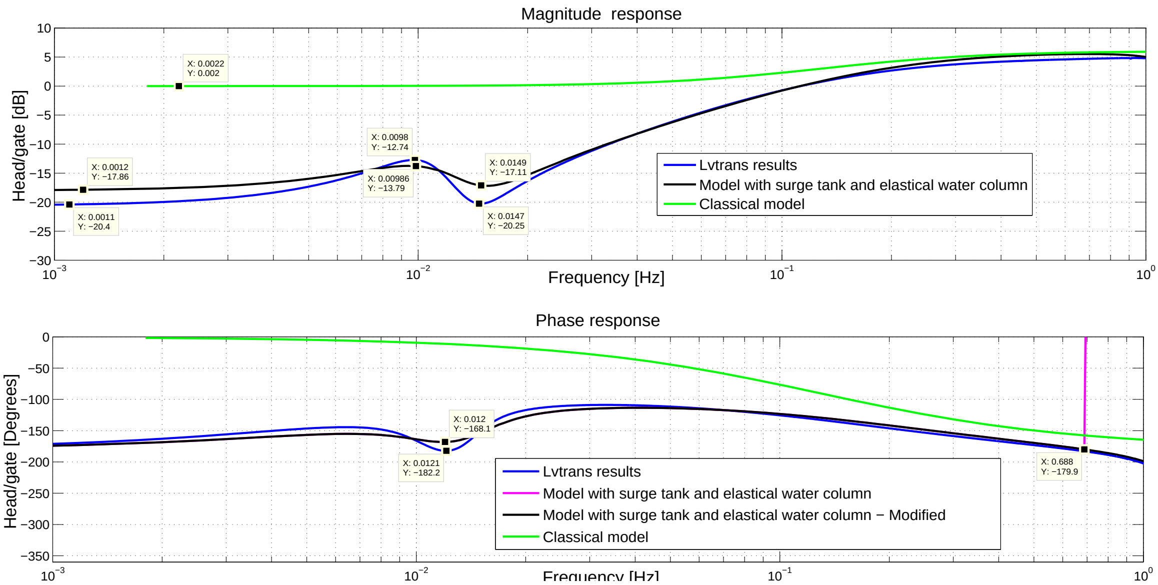


Figure 5.3: LVtrans, Model with surge tank and elastic water column, the classical model, $10^{-3}$-1Hz

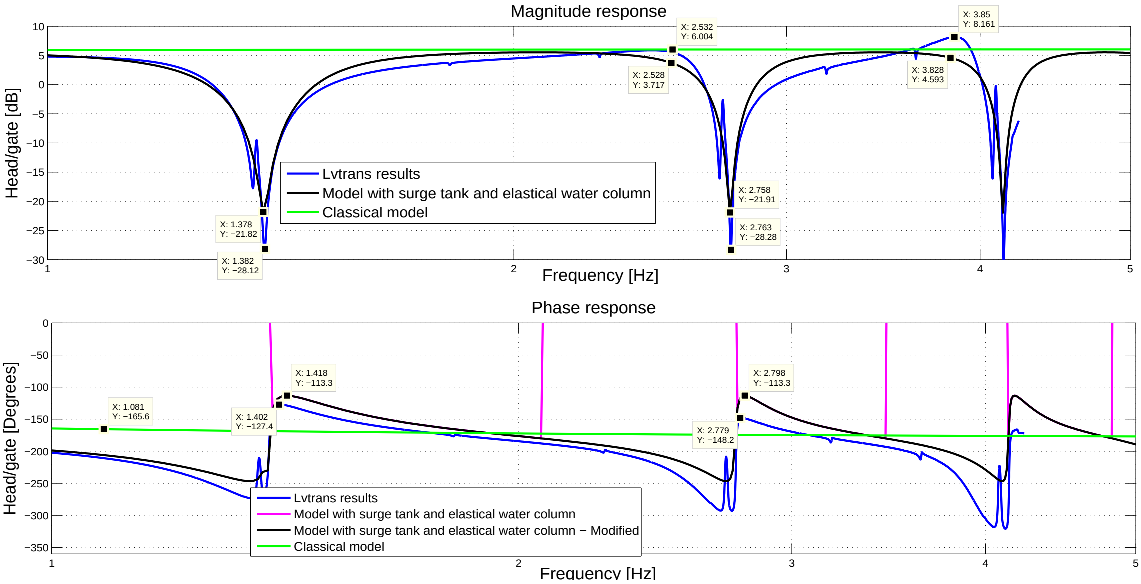


Figure 5.4: LVtrans, Model with surge tank and elastic water column, Classical model, 1-5Hz

### 5.3.3 Comparison between Lvtrans results and model with surge tank, elastic water column and turbine parameters

As mentioned in section 2.2 turbine parameters are seldom easily available and vary according to the operation point. The turbine parameters for the specific turbine used in this master thesis is given in Lucero et al. [2]. In addition, two standardized methods for finding the turbine parameters will be tested, Konidaris [9] and Strah et al [15]. The calculation of the turbine parameters as given in [15] has been adjusted to match the operation point. Figure 5.5 shows the magnitude response for the three parameter sets compared with Lvtrans. The numerical value of the turbine parameters are given in table 5.2.

Table 5.2: Turbine values used in simulation

| Turbine parameter | As in [2] | As in [9] | As in [15] | Resulting value [15] |
|---|---|---|---|---|
| $\partial q/\partial y$ | 0,64 | 1,0 | $\sqrt{h}$ | 0.949 |
| $\partial q/\partial x$ | -0,49 | 0 | 0 | 0 |
| $\partial q/\partial h$ | 0,37 | 0,5 | $y/2\sqrt{h}$ | 0.428 |
| $\partial m/\partial y$ | 1,31 | 1,0 | $h^{3/2} - D_a(\omega_n - 1)$ | 1.06 |
| $\partial m/\partial x$ | -2,44 | 0 | -$D_a y$ | -0.45 |
| $\partial m/\partial h$ | 1,97 | 1,0 | $1.5\sqrt{h} * y - qnl$ | 1.18 |

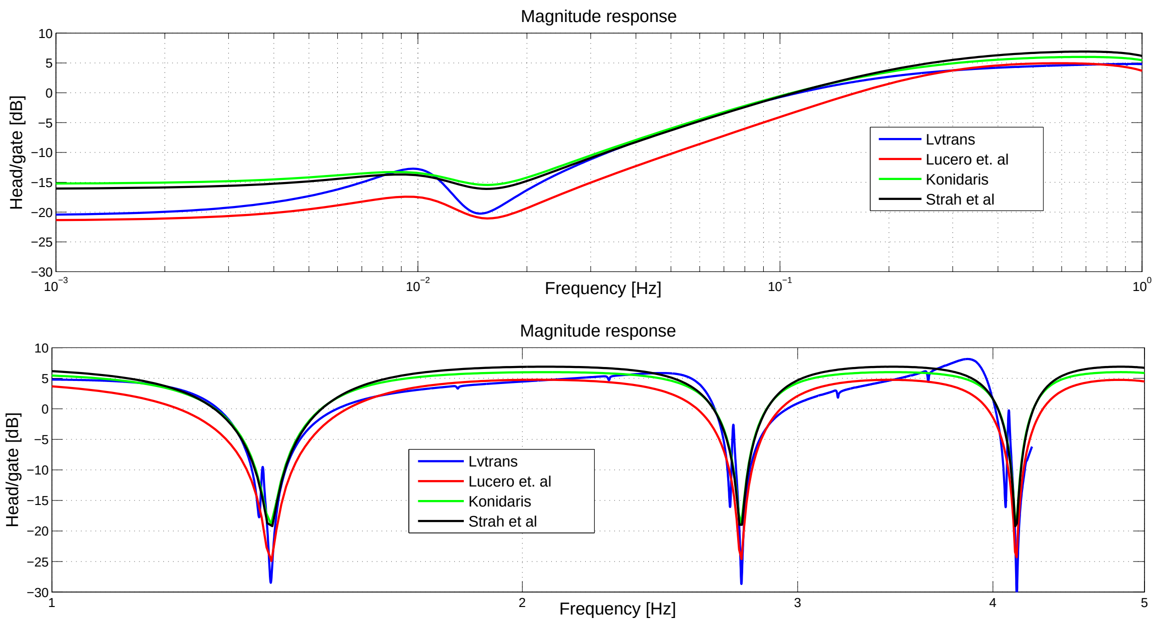


Figure 5.5: Magnitude response for model including turbine parameters, turbine parameters as in table 5.2, $10^{-3}$ -5Hz

### 5.3.4 Improvement of the parameter set for the model including turbine parameters

By examining figure 5.5 it is apparent that there is potential for improving the agreement between the Lvtrans results and the simulation results for the model including turbine parameters. Thus, It was of interest to investigate if it is possible to achieve an even better agreement between the model with turbine parameters and Lvtrans by simply changing the turbine parameter values. The parameter set suggested by Konidaris, [9], seemed to be a good starting point. The parameter values for $\partial q/\partial x$ and $\partial m/\partial x$ are zero in the Konidaris parameter set. Investigating the block diagram of the model, see section 3, it becomes apparent that by setting these parameters to zero the model becomes speed independent, as there is no feedback from the speed deviations to the model. Since the dynamic behavior of the turbine during off nominal speeds is believed to have an impact on the electric hydraulic interaction, it seems natural to include the turbine parameters $\partial q/\partial x$ and $\partial m/\partial x$ with numerical values as suggested by Lucero et. al, [2]. The friction values used in the previous simulations are steady state friction. Lvtrans uses a simple form of dynamic friction [11]. A better fit was achieved by changing the friction values used for simulation of the model with turbine parameters. Figure 5.6 shows the magnitude response of the model with the improved parameter set as well as parameter set as in [2] and [9]. Table 5.6 shows the changes in parameters.

Table 5.3: Improved parameter values for model including turbine parameter

| Parameter | Original value | Improved value |
|---|---|---|
| $\partial q/\partial x$ | 0 | -0,49 |
| $\partial m/\partial x$ | 0 | -2,44 |
| Friction factor overhead tunnel | 0,02244 | 0,005 |
| Inflow loss factor, surge tank | 0,012 | 0,005 |
| Friction factor penstock | 0,0351 | 0,015 |

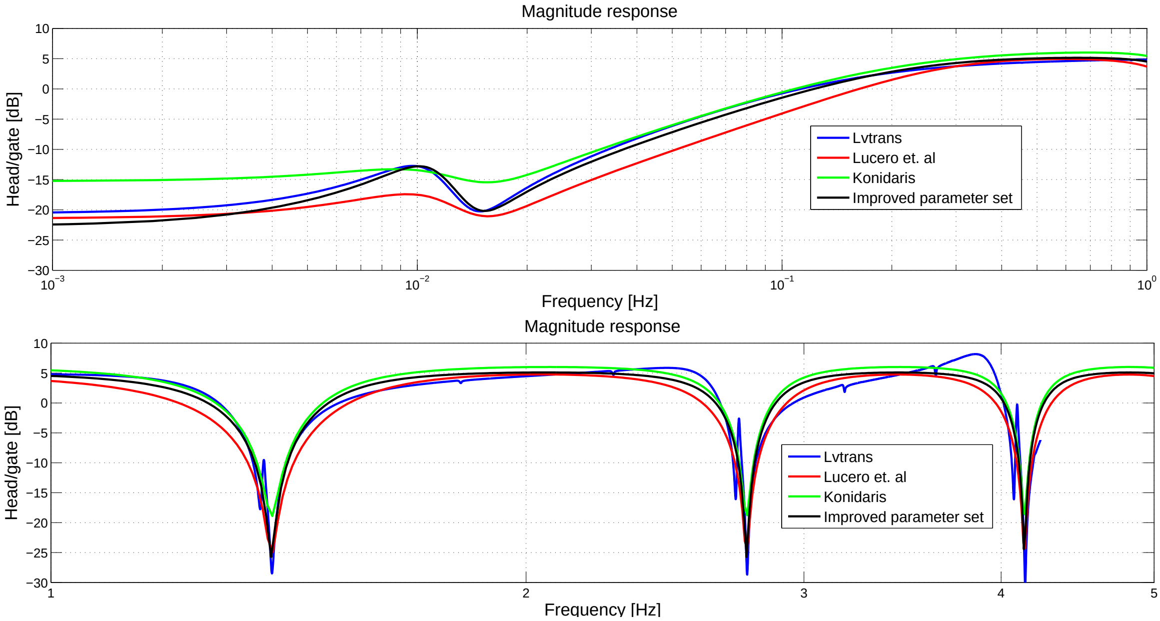


Figure 5.6: Magnitude response for model including turbine parameters with improved parameter set, $10^{-3}$ -5Hz

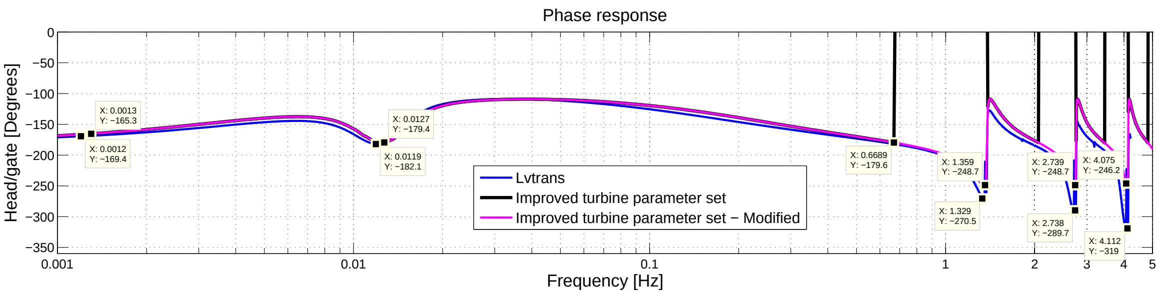


Figure 5.7: Phase response for model including turbine parameters with improved parameter set, $10^{-3}$ -5Hz

The improved parameter set seems to give best agreement with the LVtrans results. Therfore this parameter set was used in further simulations.

# Chapter 6

# The hydraulic models simulated in different power system configurations

## 6.1 Description and parameter values of key components used in simulations

### 6.1.1 Synchronous machine

The synchronous generator used in the simulations described in 6.2 - 6.4 is a machine with one field winding, one damper winding in the direct axis and one damper winding in the quadrant axis. The parameter values are given in 6.3 and shown in table 6.1.

Table 6.1: Parameter values for synchronous machine

| | |
|---|---|
| Rated power | 50 [MVA] |
| Rated voltage | 20 [kV] |
| Inertia constant | 3 [MWs/MVA] |
| D-axis synchronous reactance | 1.174 [p.u] |
| D-axis transient reactance | 0.3 [p.u] |
| D-axis subtransient reactance | 0.215 [p.u] |
| Q-axis synchronous reactance | 0.77 [p.u] |
| Q-axis subtransient reactance | 0.15 [p.u] |

## 6.2 Single machine infinite bus system

### 6.2.1 Description of the network and fault situation

The system consist of a synchronous generator connected to an infinite bus via a line and a transformer. The initial load flow situation is shown in figure 6.1. The total simulation time was 30 seconds with time step decided by the simulation software, Simpow. The simulation sequence consist of a 3 phase to ground fault at the generator terminals after 1 second, sustained for 50 millisecond. Each of the three hydraulic models described in chapter was simulated separately.

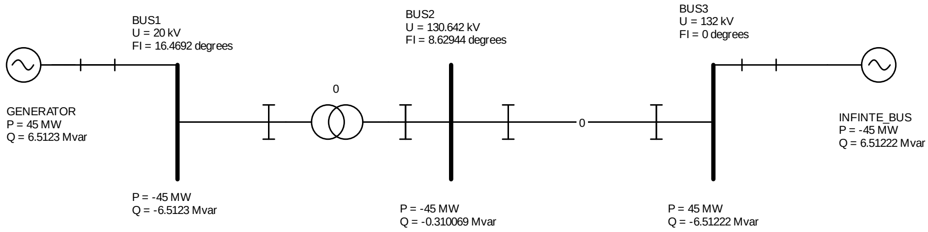


Figure 6.1: Single machine infinite bus system, Single line diagram

### 6.2.2 Simulation results

The simulation results for active power, speed of the turbine and pressure at the turbine are found in figures 6.2 - 6.4. The results are shown with and without turbine governor connected.

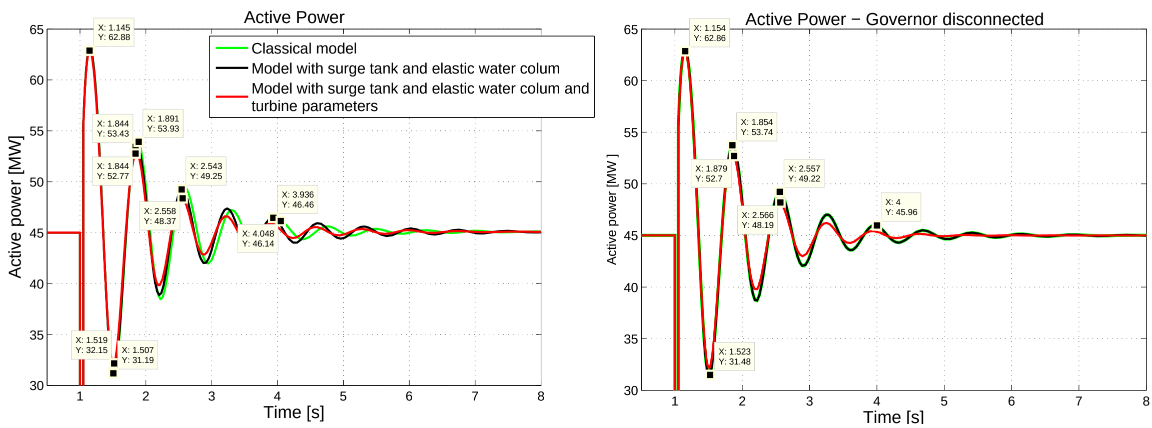


Figure 6.2: Single machine infinite bus system, Active Power

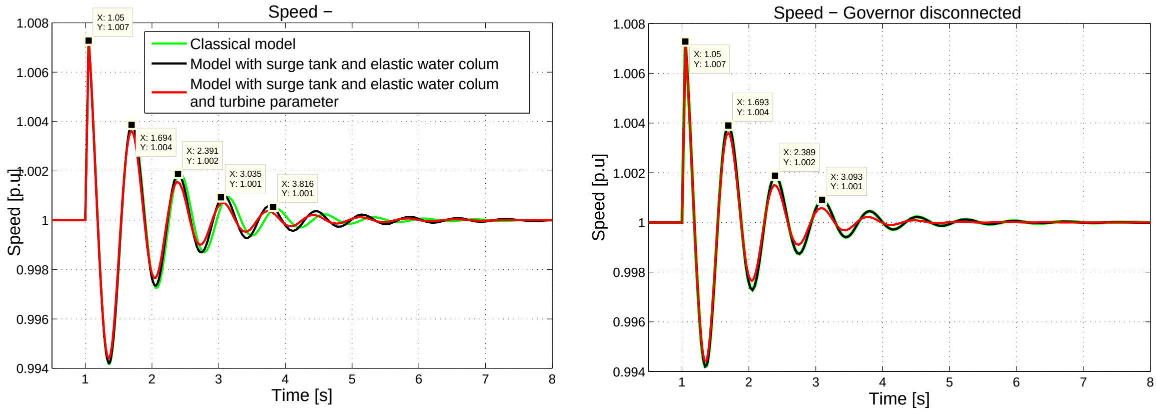


Figure 6.3: Single machine infinite bus system, Speed of the turbine

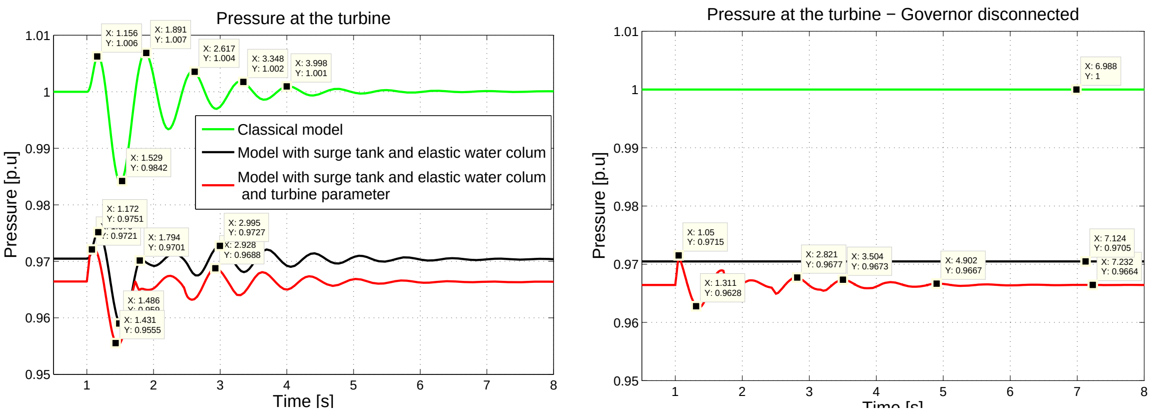


Figure 6.4: Single machine infinite bus system, Pressure at the turbine

## 6.3 Kundur two area system

### 6.3.1 Description of the network and fault situation

The system consist of two interconnected areas connected by two parallel lines. Area one is a surplus area supplying power to area two. Each area has two synchronous generators. All the generators have a nominal capacity of 55MVA. After 1 second one of the lines connecting the areas are permanently disconnected. The original case is suggested by Kunudr[7]. The original case has been changed by SINTEF energy research to be suitable for use with hydro power generators. The initial power flow and layout is shown in figure 6.5.

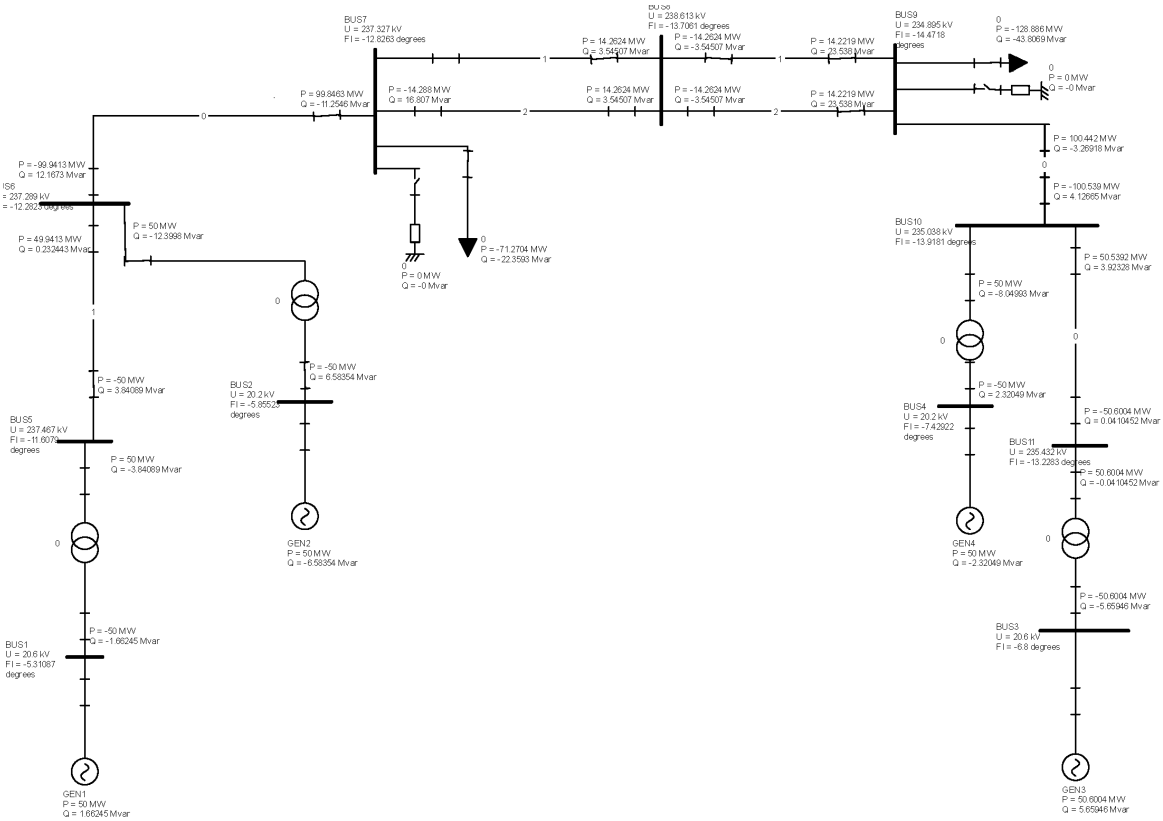


Figure 6.5: Kundur two area system, Single line diagram

## 6.3.2 Simulation results

Figure 6.6 and 6.7 shows the active power and speed of generator with the inertia constant for generator changed to 2 seconds and 4 seconds respectively. In figures 6.8 and 6.9 the effect of changing the turbine parameters is shown.

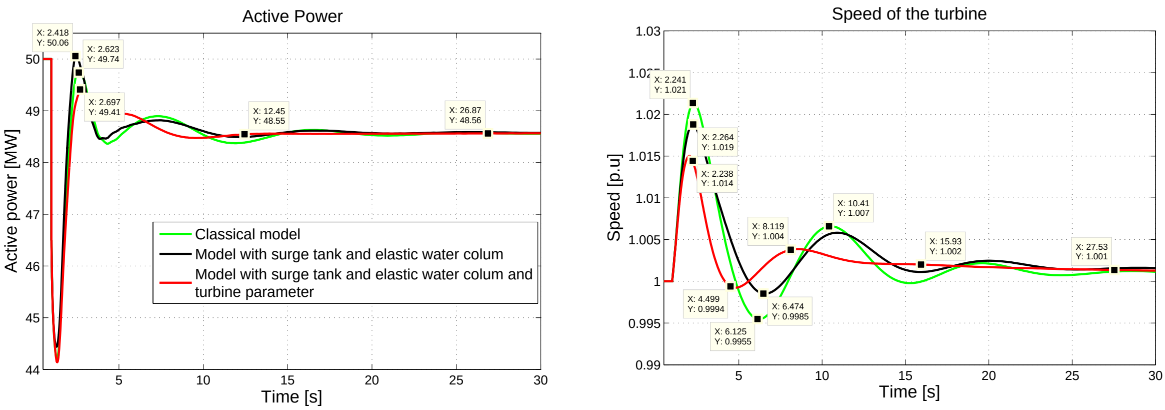


Figure 6.6: Two area system, Active power and speed, Inertia constant=2s

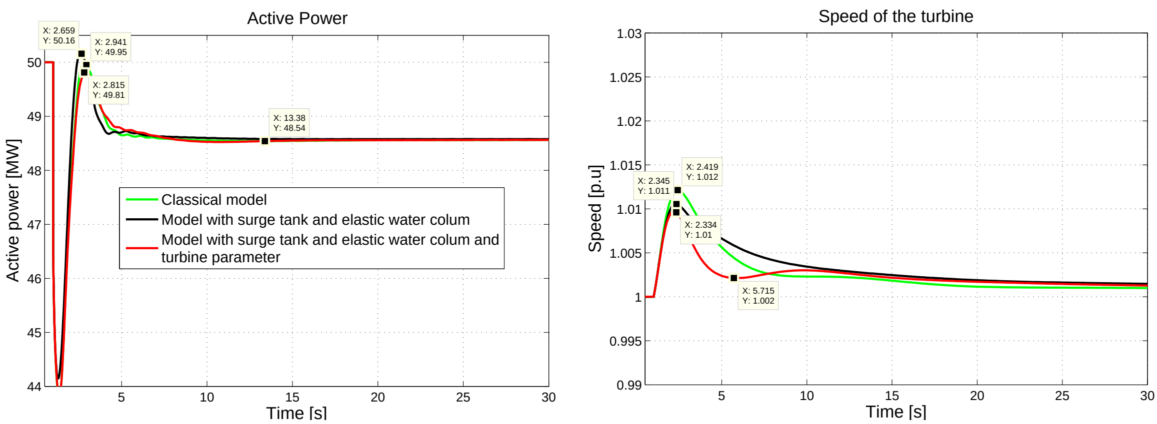


Figure 6.7: Two area system, Active power and speed, Inertia constant=4s

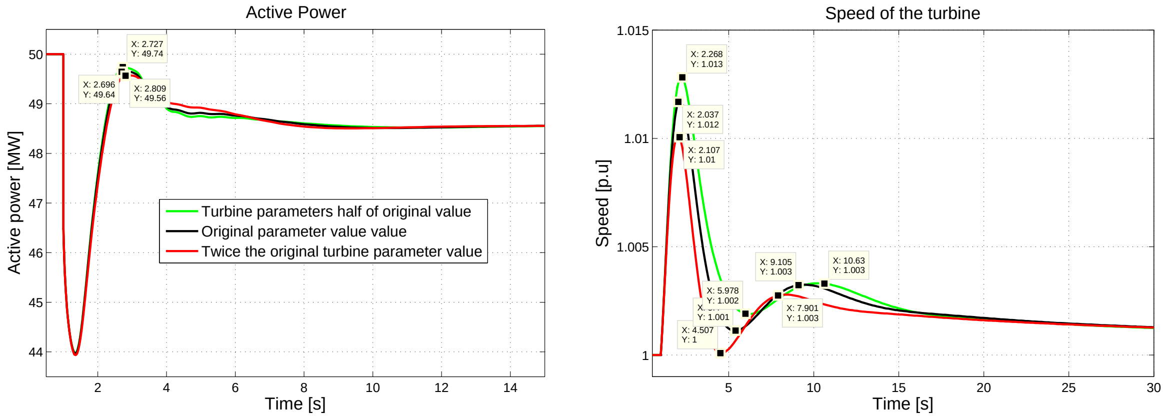


Figure 6.8: Two area system, Pressure and flow at the turbine, varying turbine parameters, Active power and speed

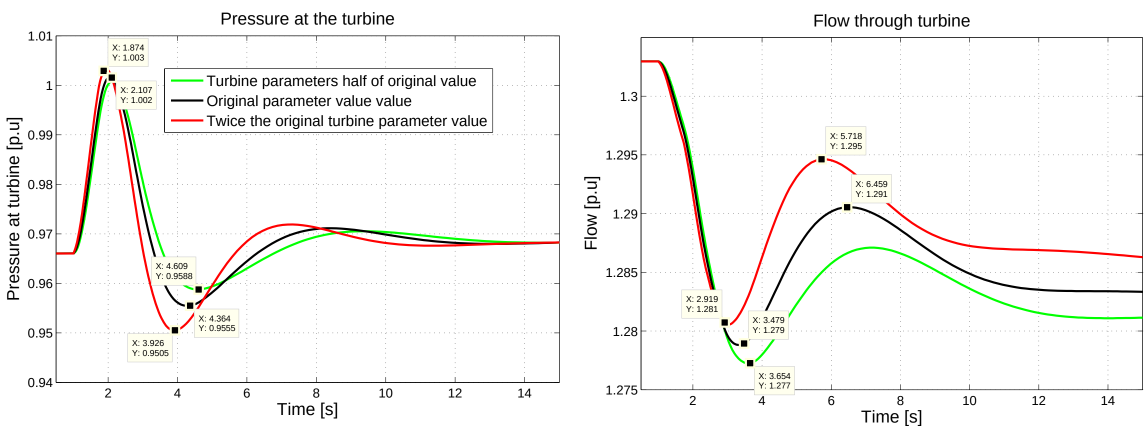


Figure 6.9: Two area system, Active power and speed, varying turbine parameters, Head and flow

## 6.4 System with enforced and sustained power swings

### 6.4.1 Description of the network and fault situation

The system consists of a generator connected to a transformer and a load. The load is in turn connected to an infinite bus that has sustained power oscillations, leading to power oscillations in the connected generator. The hydraulic models are implemented at the generator bus. Sinusoidal power oscillations are applied to the infinite bus with frequencies 0.5, 0.69, 1.38 and 2 Hz. The amplitude value of the power oscillations can be found in appendix A. The power oscillations at the infinite bus were created by using a .dsl file. The dsl file was originally created by the software developers.

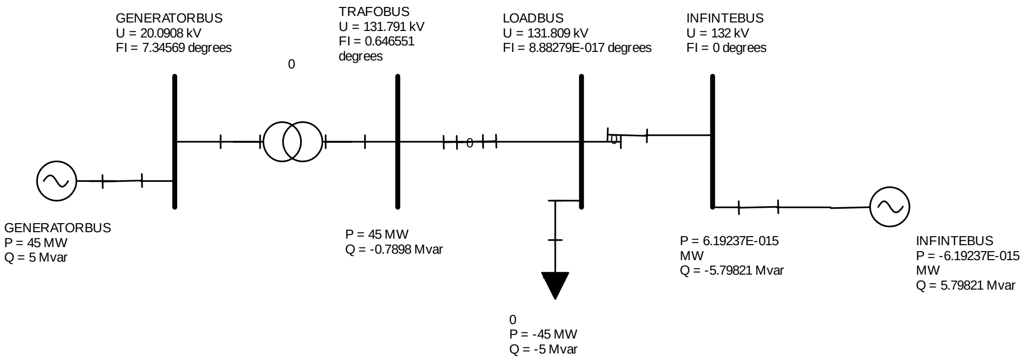


Figure 6.10: System with sustained power oscillations,Single line diagram

### 6.4.2 Simulation results

The resulting pressure and flow for the power oscillations are shown in figure 6.11 - 6.14. Note that for the curve showing the resulting flow from the turbine parameters model were been edited. 0.4 p.u was subtracted for better representation, for all simulated frequencies. The resulting power delivered to the network for power oscillations with frequencies 1.38 and 2 HZ are shown in figure 6.15.

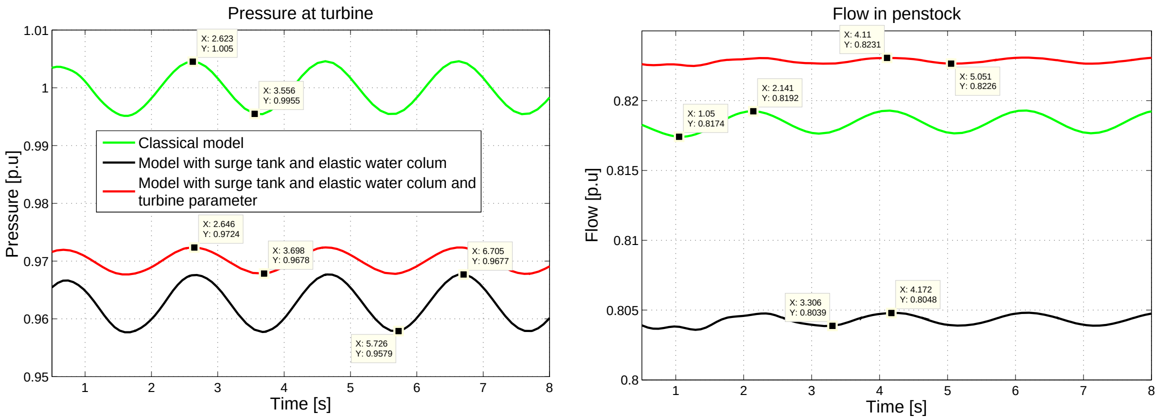


Figure 6.11: System with power oscillations, Pressure and flow at the turbine, Power oscillations 0.5Hz

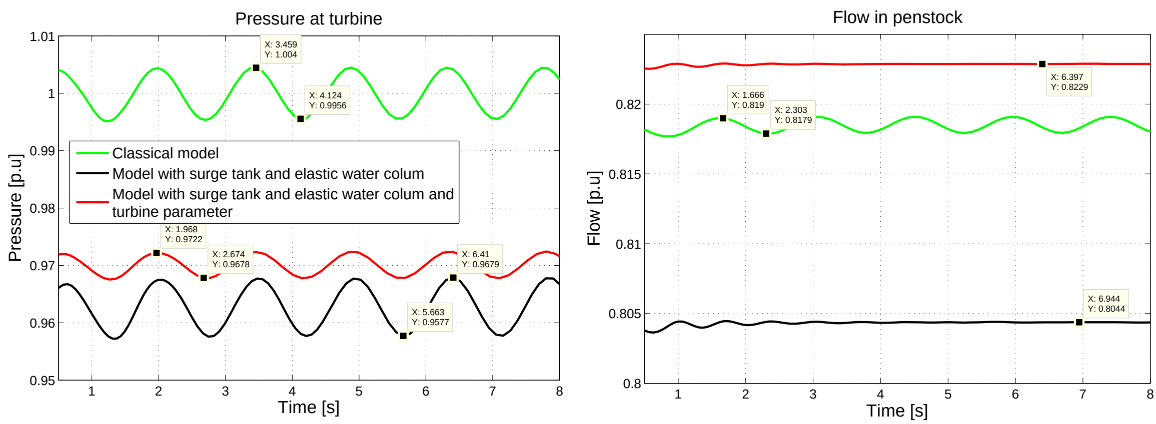


Figure 6.12: System with power oscillations, Pressure and flow at the turbine, Power oscillations 0.69Hz

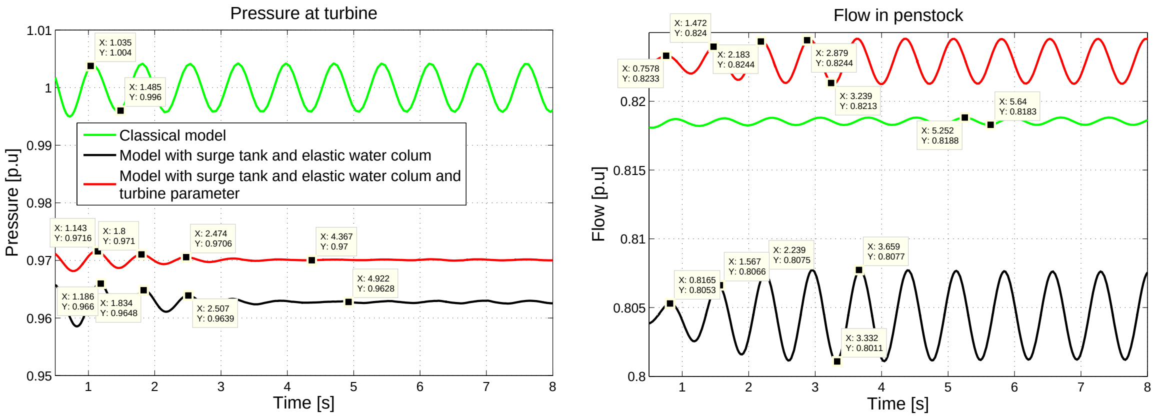


Figure 6.13: System with power oscillations, Pressure and flow at the turbine, Power oscillations 1.38Hz

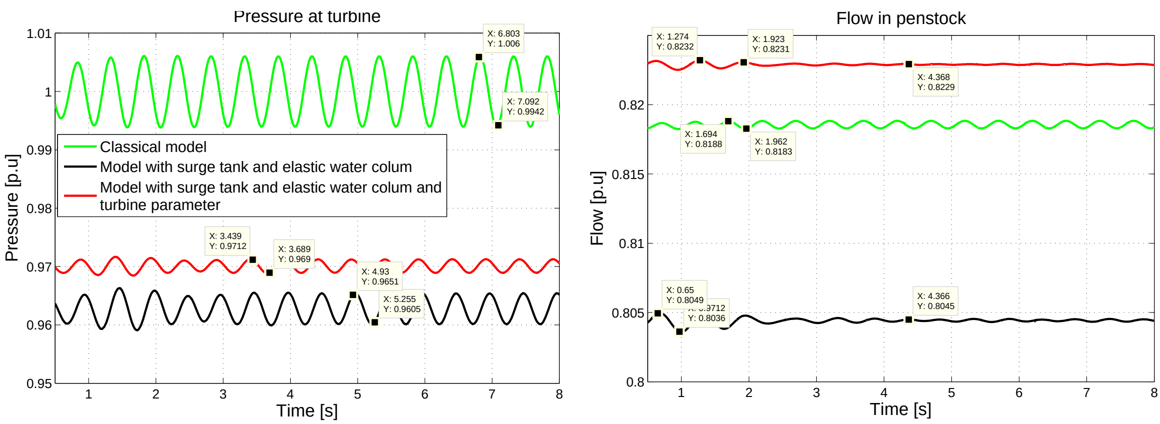


Figure 6.14: System with power oscillations, Pressure and flow at the turbine, Power oscillations 2Hz

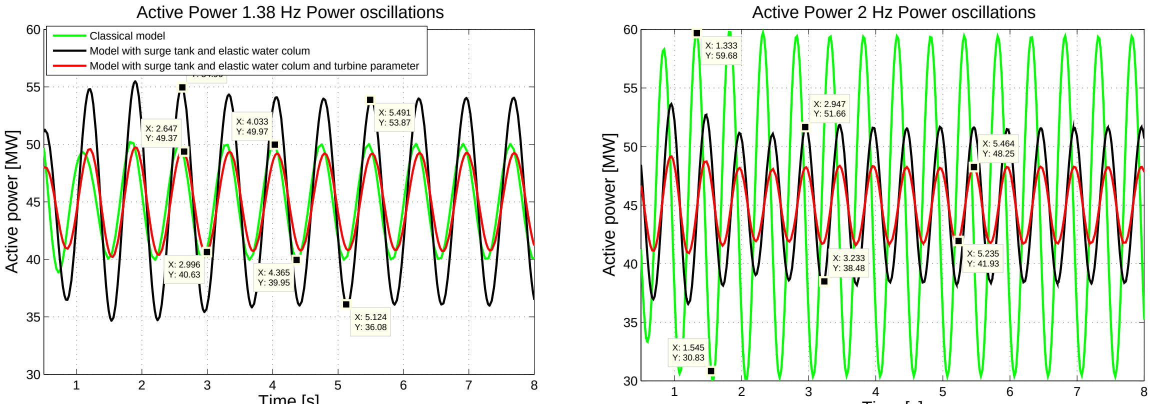


Figure 6.15: System with power oscillations, Active power, 1.38 and 2Hz

# Chapter 7

# Discussion

## 7.1 Performance of the hydraulic models

Figure 7.1 shows the magnitude response of the hydraulic models used in the simulations with hydraulic models implemented in power systems.

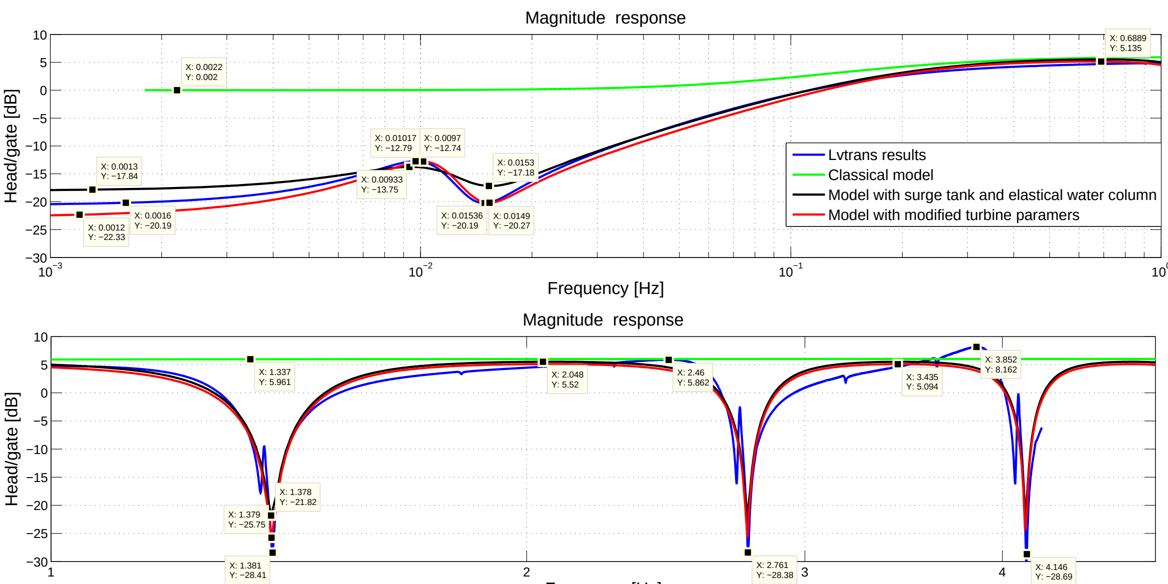


Figure 7.1: LVtrans simulation results $10^{-3}$ -1Hz

By examining figure 7.1 there are two distinctive points at the curves at around 0.01 Hz and at 1.38 HZ. The hydro power plant simulated has a penstock length of 435 meters and a pressure wave speed of 1200 m/s , giving a natural frequency of the water hammer effect of 0.69Hz, or a half period frequency of 1.38 Hz. The natural period of the surge tank is calculated to 92,95 seconds or 0.0107Hz, see equation 2.11.

The curves for Lvtrans, the model with surge tank and elastic water column and the model with turbine parameters have peaks at 0.0097, 0.00933 and 0.01017 Hz with magnitudes of -12.74, -13.75 and -12.79 db respectively.

The model with turbine parameters has been simulated with different parameter sets. The curve shapes for the different parameter sets in the frequency range from 0.001 to 0.1 Hz have similar curve shapes, although the turbine parameter sets are not proportional. All the curves deviate from the LVtrans simulation. Figure 5.6 shows the turbine model with an improved parameter set as well as changed friction coefficients. By investigating the curve shape of the turbine model it appears that the turbine parameter model deviation from LVtrans has been reduced.

The curves for Lvtrans, the model with surge tank and elastic water column and the model with turbine parameters have dips at 1.381, 1.379 and 1,378 Hz with magnitudes of -21.82, 25.75 and -28.41 db respectively. All three curves have dips at approximately 2.76 and 4.14 corresponding to 4 and 6 times the half period frequency of the water hammer effect. The maximum values of the model with surge tank and elastic water column appears at around 0.68, 2.04 and 3.4 Hz, corresponding to 1, 3 and 4 times the natural frequency of the water hammer effect. The magnitude value is 5.135, 5.52 and 5.094db. The Lvtrans has peak values at 0.69, 2.459 and 3.852Hz with magnitudes of 5.135, 5.852 and 8.192 db.

The classical model has a value of 0.0002 db in the frequency range from $10^{-3} - 2 * 10^-3$ and gradually increases to 5.961db at 0.69Hz.

The phase response of the models has also been simulated. The deviation between Lvtrans and the model with surge tank and elastic water column and the model with turbine is 14.1 and 2.7 degrees at 0.012Hz. For both the models the deviation increases with increasing frequency and the deviation is most significant at 1.38, 2.76 and 4.14 Hz corresponding to 2,4 and 6 times the natural frequency of the water hammer effect.

## 7.2 Limitations and further work with the hydraulic models

In the Simpow simulations the perturbation amplitude was set to 0.01 per unit. Other perturbation amplitudes have not been tested. The size of the perturbation may influence the simulation result.

Further, the numerical methods used in Simpow and LVtrans have not been investigated. It is known that LVtrans uses the method of characteristics. The numerical methods of Simpow are not known. The numerical calculations of the two programs may differ, giving systematic errors in the results. Lvtrans is believed to give accurate results. Verification of the LVtrans results against actual measurements has not been carried out, possibly leading to errors in the developed turbine parameter model.

It has been shown that the value of the friction factor has a significant impact on the simulation results for the model with turbine parameter in the frequencies below $3*10^{-2}$ Hz. A method for finding correct friction factors for use in Simpow should hence be developed.

The turbine parameter model requires accurate turbine parameters. A method for calculation of these parameters should be investigated further, preferably an automatic method, since the numerical value of the turbine parameters change according to the operation point.

## 7.3 Performance of the hydraulic models implemented in power system configurations

The active power, the speed of the turbine and the pressure at the turbine were investigated for the infinite bus single line system for a 3 phase to ground fault. The simulation for active power shows that post fault the delivered power is 62.88MW and 62.86MW with and without governor respectively, equal for all the models. For the model with turbine parameter the oscillations after the fault last for 7s with governor and 5s without governor. The classical model and the model with elastic water column oscillate for 7.5 seconds in both cases. The maximum deviation between the power delivered from the different models is 1.16MW and 1.04MW with and without governor, the classical model delivering the highest power, turbine parameter models delivers the lowest power.

The speed variation is maximum 0.7% of the nominal value. The damping of the speed variation takes 7 second for all models with the governor connected. When the governor is disconnected the damping takes 5.0 seconds for the turbine parameter mode and longer than 8 seconds for the two other models.

The initial pressure at the turbine is highest for the classical model and lowest for the model with turbine parameters with the governor connected. The progress of the classical model follows an exponential damped sinusoidal course. Initially the two other models show a similar course, but in the period from 1.8 seconds to 3 seconds they follow a more irregular course before entering a similar course as for the classical model. When the governor is disconnected, the classical model and the model with surge tank and elastic water column show no response, while the model with turbine parameters clearly shows a response.

Moreover, Kundurs two area system was investigated. The resulting active power and speed for different inertia constants were investigated. In addition different values of the turbine parameters $\partial q/\partial x$ and $\partial m/\partial x$ were simulated.

The results from active power shows that the model with surge tank and elastic water column delivers the highest power and the turbine parameter the lowest for both values of the inertia constant. The maximum variation is 0.66MW with H=2 and 0.38MW with H=4. The classical model has the largest speed deviation with the inertia constant= 2MVA/MW the damping ratio is largest for speed and power for the model with turbine parameters.

Investigation of how the variation in the two turbine parameters related to speed was studied. The speed variation after disconnection of the line is around 1% of the nominal speed. The damping of the speed variation is highest when the turbine parameters have been doubled, lowest when they have been reduced. The active power shows only minor variations by changed turbine parameters. Pressure at the turbine is affected most when the turbine parameters are doubled, least when they are halfed.

Pressure and flow variations as a result of power oscillations were investigated for selected frequencies. Pressure and flow variations are proportional to the applied sinusoidal power oscillations for all frequencies in the classical model. Pressure variations are proportional to the power oscillations for 0.5, 0.69, and 2 Hz for the two other models. Flow variations are proportional for 0.5, 1.38 and 2Hz. Pressure variation at 1.38 HZ for the turbine parameter model and the model with surge tank and elastic water column are initial proportional to the sinusoidal variation in power, decays and becomes very little affected by the power oscillations. Likewise the flow is only marginally affected by the power oscillations at 0.69Hz.

Simulations of the power delivered from the generator at 1.38 and 2 HZ show that the model with surge tank and elastic water column gives about 3.5 MW more than the two other models. The 2Hz simulations show that the classical model gives about 11.4MW more than the turbine parameter model and 8MW more than the model with surge tank and elastic water column which in turn gives 3.5MW more than the model with turbine parameters.

## 7.4 Limitations and further work with for the hydraulic models implemented in power systems

As stated in the theory part, turbine parameters are only valid for one specific operation point. During the course of the simulation the operation point of the turbine has changed without changing the turbine parameters, putatively leading to errors in the results. The case with sustained power oscillation has utilized a .dsl file that produces power oscillations, originating from the developers of Simpow. The .dsl file has not been thoroughly studied and may impose unintended effects on the power system. Albeit effort has been made to develop realistic scenarios with realistic input data, some of the input data for the electrical configuration might not necessarily reflect actual power system configurations accurately. The effect of the type and input values of governors and voltage regulators was not studied in detail, and could influence the simulation results.

Several recommendations for further work could be outlined. Primarily a study of the possible interaction between an electrical network, the hydraulic model and the synchronous generator may be of interest. The consequence of not changing turbine parameters during changed operation point should also be investigated. In addition, only a limited number of power system configurations has been investigated and it could be of interest to examine how the different models behave under different network configurations.

# Chapter 8

# Conclusion

Three different hydraulic models were successfully implemented in Simpow and a frequency scan in the range from $10^{-3} - 5$ Hz was performed and the results were compared with results from LVtrans simulations.

The magnitude response for the classical model overestimate the magnitude response for the whole frequency range, except for the frequencies from 3.6 to 4 Hz, corresponding to 5.58 times the natural frequency of the water hammer effect. As expected the model fails to show the effect of the surge tank and the elastic water column as it is not included in the model.

The comparison between the LVtrans results and the model with turbine parameter and the model with surge tank and elastic water column show that there is significant deviation between the simulation results in the frequency range from $10^{-3}$ to $3*10^{-2}$ Hz when steady state friction values are used. The improved parameter set with reduced friction factors gives improved similarity between the LVvtrans results and results for the model with turbine parameter. This suggests that the friction factor significantly affects the magnitude response in this frequency range.

The effect of the elastic water column, or water hammer effect was examined. The results show that both the model with elastic water column and the model with turbine parameters are able to accurately display the natural period and half period of the water hammer effect. The magnitude response is in line with the Lvtrans result for frequency lower than one half period of the water hammer effect. At higher frequencies the deviation between LVtrans and the models increases.

The phase response of the models was also studied. The classical model is not able to accurately represent the phase response. The deviation between LVtrans and the model with surge tank and elastic water colum increases near multiplums of the half frequency of the water hammer effect. The improved turbine parameter set reduces the deviation. It should be pointed out that the Simpow adds 360 degrees to the phase response when the phase response drops below -180degrees.

Different parameter sets for the model with turbine parameters were investigated. The results show that different values of turbine parameters and friction factors influence the simulation results. The correspondence between LVtrans simulation and model with turbine parameters may improve by applying a different set of input values.

The three different hydraulic models have successfully been implemented in three power systems. The active power, the speed, the pressure and the flow variations were investigated.

The difference in active power delivered from the generator where the different hydraulic models are implemented vary. In the infinite bus the variations are 2.57 % of the initial power delivered, Similar the deviation in Kundur two area system model is 1.32%. For the system with sustaind power oscialltions the resulting are larger than for the two other electric systems. In general the model with turbine parameters are better damped than the two other models.

Additionally, speed variations were studied. The model with turbine parameters gives the smallest deviations from the nominal speed. The model damp speed deviations better than the two other models.

The simulation results show that the initial pressure at the turbine is lowest for the turbine parameter model when the systems are in steady state. When the governor is disconnected only the model with turbine parameters are able to show a pressure variation resulting from a variation in speed. For power oscillations with frequencies equal to the half period frequency of the water hammer effect,1.38Hz, the model with surge tank and elastic water column and the turbine parameter model show very little response. For frequencies equal to the water hammer effect freqency, 0.69Hz, the variation in flow is also small for the two models. This suggest that power oscillations in the electrical network can affect the pressure and flow in the conduit system and that the interaction depends on the characteristics of the hydraulic systems.

Three different hydraulic models have been implemented in three different power system configuartions. Simulation with the different models display good agreement between the models for active power and speed, except for the system with sustained power system oscillations. Moreover, the model with turbine parameters is generally better damped then the two other models. Finally, the turbine model requires accurate turbine parameters and frictional values, and it has proved time consuming to determine such values.

# Appendix A

# Power oscillations from generator

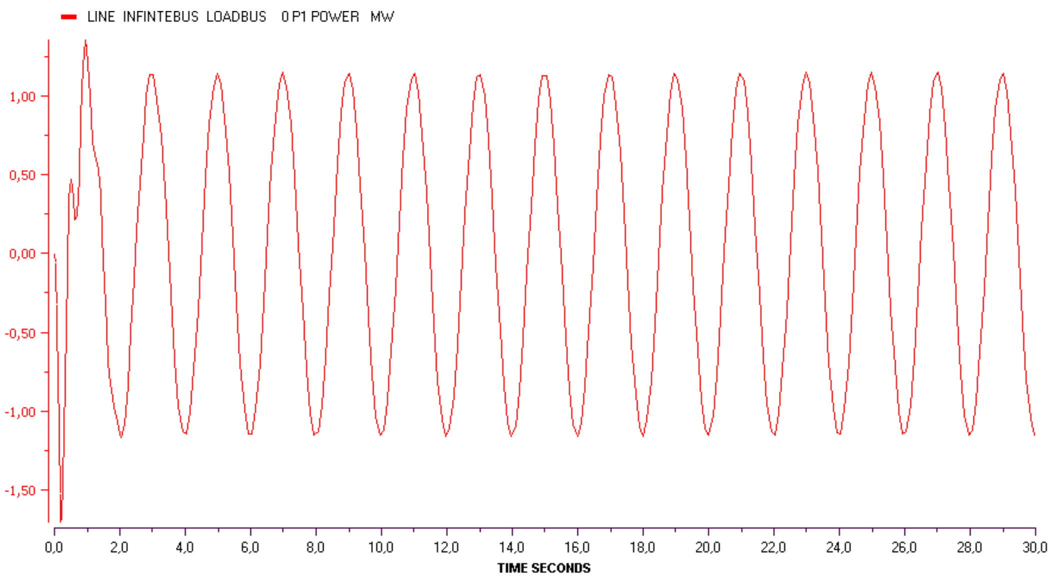


Figure A.1: Power swing from inifite bus 0.5Hz

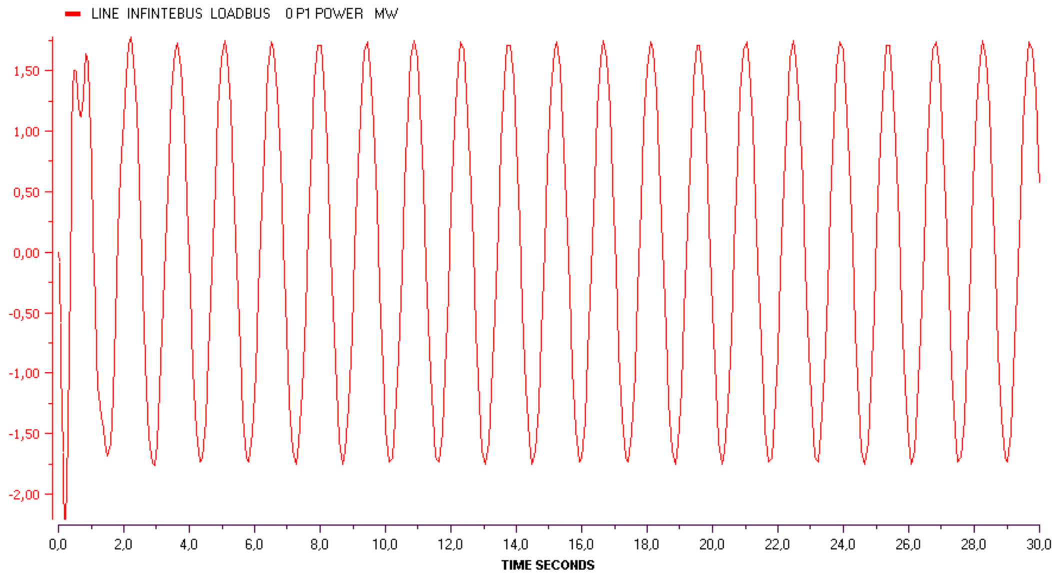


Figure A.2: Power swing from inifite bus 0.69Hz

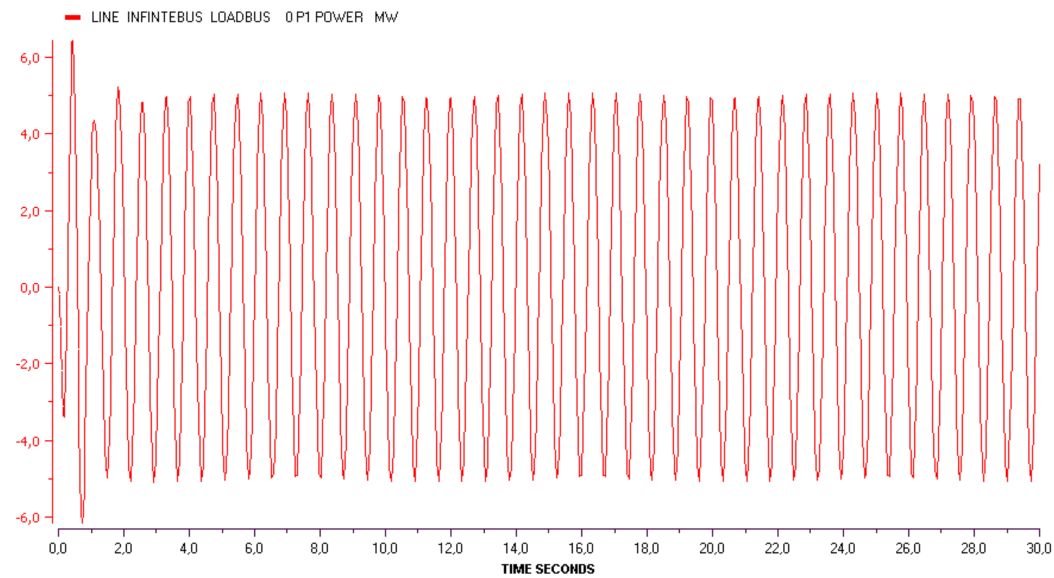


Figure A.3: Power swing from inifite bus 1.38 Hz

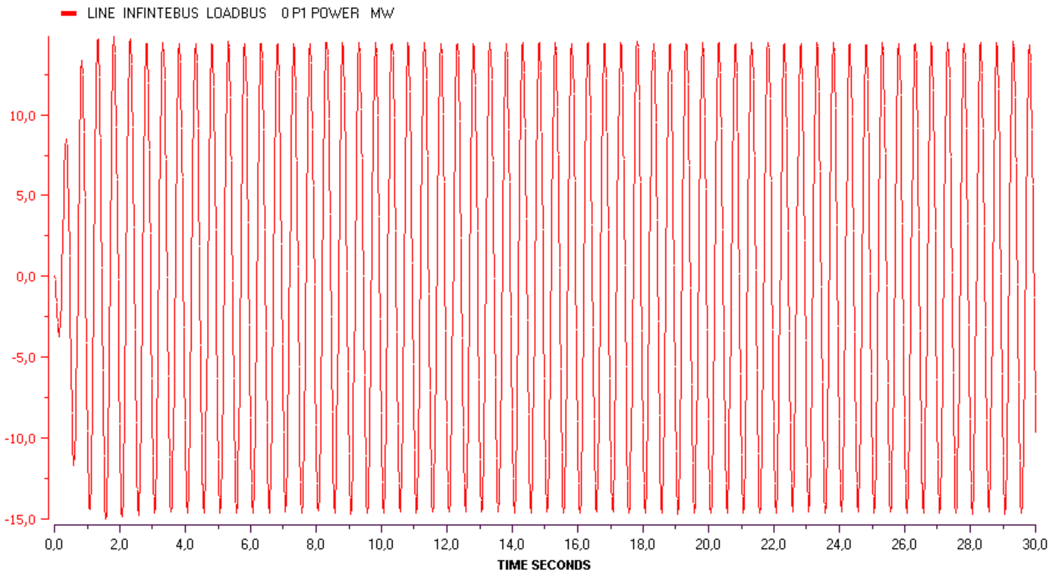


Figure A.4: Power swing from inifite bus 2Hz

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