



# CHALMERS

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## The Potential for Frequency Control in Paper Mills

With Particular Reference to Iggesund Mill

*Master's thesis in Energy and Systems Engineering*

PIERRE HULT



MASTER'S THESIS IN ENERGY AND SYSTEMS ENGINEERING

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PIERRE HULT

Department of Heat and Power  
*Division of Energy and Environment*  
CHALMERS UNIVERSITY OF TECHNOLOGY

Gothenburg, Sweden 2015

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PIERRE HULT

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Master's thesis ENMX04  
ISSN 1652-8557  
Department of Heat and Power  
Division of Energy and Environment  
Chalmers University of Technology  
SE-412 96 Gothenburg  
Sweden  
Telephone: +46 (0)31-772 1000

Cover:  
Picture of the Iggesund pulp and paper mill

Chalmers Reproservice  
Gothenburg, Sweden 2015

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## ABSTRACT

As more intermittent energy resources are entering the energy system the need for frequency regulation increases rapidly. At present, most of the frequency regulation is taken care of by hydropower. It is though becoming more difficult for the hydropower to take care of frequency control alone, especially at night when much of the hydropower is out. The costs of the frequency control can then be relatively high. The possibility for CHP plants and paper mills to make financial gain through participation is then of interest to investigate.

In this thesis the possibility for paper mills and especially Iggesund Paper Mill to participate in the Primary frequency regulation has been looked at, with respect to process and control changes. Paper mills usually consist of a recovery boiler and in some cases additional boilers with relatively slow response and one or more steam turbines to generate power. The problem is to meet the fast power response to the grid required for participation while still satisfying the steam demand for the steam consuming processes in the paper making. A simplified model of Iggesund mill has been modelled using the software Dymola. A graphical interface that easily build and simulate large complex systems with many fixed equations.

Different control methods and their effect on the system as a whole have been analysed. If the control system is fairly redesigned and the oil fuel inputs are maximized in one or more of the boilers during frequency deviations, Iggesund mill can participate to some extent. Maximizing the oil inputs is however not desirable in the long term as the aim is to be independent on fossil fuel. The damage and stress it would do to the boilers is questionable. Throttling of steam past the turbine and redistributing for ramp up seems to be the easiest and cheapest way to control the plant in participation.

Keywords: Frequency control, Pulp and Paper mills, Steam network, Dymola, Time delay, Modeling

## ACKNOWLEDGEMENTS

I want to thank Solvina and all the employers that made this thesis work possible; Peter Dahlström, Mikael Eng and Bengt Johansson for the background research. I want to thank my examiner Klas Andersson at Chalmers University of Technology and my supervisor Mattias Johansson for their collaboration and feedback. Special thanks to Mattias Johansson and Henrik Granberg at Solvina for their help and support.



# NOMENCLATURE

## Abbreviations:

PFR	Primary Frequency Regulation
FCR	Frequency Containment Reserves
FCR-N	Frequency Containment Reserves for Normal operating band
FCR-D	Frequency Containment Reserves for Disturbances
FRR	Frequency Restoration Reserv
ENTSO-E	The European Network of Transmission System Operators for Electricity
TSO	Transmission System operator
SvK	Svenska Kraftnät, network operator in Sweden
LP/MP/HP	Low-/Medium-/High-Pressure
P11	Oil Boiler in Iggesund
P12	Oil- and Bark Boiler in Iggesund
SP	Recovery Boiler in Iggesund
SST800	Steam Turbine in Iggesund
GB	Gas Boiler in Iggesund
PI	Pressure Indicator
FI	Flow Indicator
MCR	Model Predictive Control
CHP	Combined Heat and Power
FWH	Feed Water Heating

## Concepts and definitions:

Setpoint	The default or desired target value of any parameter used in the control systems.
Time constant	The time it takes for the output signal to reach 63% of its final value.
Frequency	A measure of how many times the alternating current in the electric power grid changes its direction every second. The nominal value is 50 Hz.
Deadband	An interval of which the frequency control is not responsive. The measurement is updated only when it has reached a specified value from the last measurement.
Turbine control range	The interval of change of rotational speed on the shaft or power output of the generator from the set point value that is allowed to be adjusted in order to maintain a stable system frequency.
Frequency sensitivity	The minimum frequency deviation in the net that will result in a change of active power output in the generator, to recover the nominal frequency.
Network	The connection of power distributors and consumers.
Turbine- or Boiler trip	The unit is taken out of operation.
Inertia	The resistance to change the rate of motion.
Deadtime	The time before a change in the setpoint will make a change in the actual output.

Common variables:

$p$	Pressure	[Pa]
$\rho$	Density	[kg/m <sup>3</sup> ]
$L$	Length	[m]
$v$	Velocity	[m/s]
$V$	Volume	[m <sup>3</sup> ]
$\dot{V}$	Volumetric flow rate	[m <sup>3</sup> /s]
$k_v$	Valve coefficient (European standard)	[-]
$C_v$	Valve coefficient (American standard)	[-]
$\dot{m}$	Mass flow rate	[kg/s]
$m$	Mass	[kg]
$F$	Force	[kg m/s <sup>2</sup> ]
$h$	Enthalpy	[kJ/kg]
$s$	Entropy	[kJ/(kg · °C)]
$f$	Frequency	[Hz]
$T$	temperature	[K or °C]
$\alpha$	The opening of a valve	[-]



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# 1 Introduction

This project will be carried out at Solvina AB in cooperation with Chalmers University of Technology.

*Solvina is an engineering consulting firm with extensive experience in power plant dynamics, power grids, process, dynamic simulation, and simulation development. They are active in Sweden and based in Gothenburg, but they also have international assignments.*

## 1.1 Background

In a grid there has to be a constant balance between production and consumption so that the frequency does not deviate from its nominal frequency of 50 Hz. Additional future expansion of renewable electricity generation, for example from wind, will imply major challenges to this balance. Today in Sweden, the balance is mainly maintained by hydropower plants adjusting their output and thus counteracting deviations. Studies have shown that different process industries may have the possibility to participate in the balancing, but more detailed studies of Swedish process industries are needed. These kinds of studies have not been relevant before since the cost of participating has been assumed to be too high in comparison to hydropower. The demand for frequency regulation ( $\pm 100\text{mHz}$ ) now though follows a clear up-going trend, as does the price and the interest for it. [1]

To be able to supply primary frequency regulation, there has to be room for increasing or decreasing the electricity production from a normal operating mode in accordance to the ENTSO-E rules, which sets demands on the response and power output over time.

In pulp and paper mills black liquor and bark are produced as bi-products. These are combusted in different recovery boilers. The steam is primarily used for the processes in the paper mill and secondarily used to produce electricity in a steam turbine.

Pulp and paper mills have good possibilities to participate in the frequency regulation, but the response of the boilers alone are too slow. Process changes may have to be done in order to increase the generation and/or availability of accumulated steam to the turbine, through manipulation and control.

Up to date, studies of the effect of a changed mass flow through the turbine has been done. Closer examination of other process changes and/or more extensive modelling and simulations is of interest.

## 1.2 Objective

In the thesis, different process and control engineering solutions will be evaluated, to determine if Iggesund pulp and paper mill can participate in the frequency regulation. The examination will consider what kind of technical limitations that might exist and possible solutions. These solutions will be analysed through dynamic process simulations and compared to find the best solution, and/or other possibilities in terms of function and feasibility. The most simple, short term solutions will be looked at primarily.

## 1.3 Delimitations

- Examine one specific pulp and paper mill (Iggesund Mill).

## 2 Theory

In this chapter the theory to support the modeling and the result is presented. This covers briefly the software used and its solver. The theory behind the essential process equipment and the control system theory behind.

### 2.1 Modelica Language

Modelica is a free open language developed to model large complex systems, expressed in algebraic and differential equations. It was first introduced in 1997 with the objective to form a unified language for the modelling of complex physical systems. The Modelica association is a non-profit organization with members from all over the world and simulation experts developing the standard Modelica Library. An open source Modelica library is to date available containing some 1280 models and 910 functions from different fields.

In Modelica, equations building up a model or the models of a system are calculated without consideration of the actual computational order. All equations are calculated and iterated at once. The Modelica compiler will in this way determine which variables that are given or set and which should be solved. As long as there is equal number of equations as unknown variables, it does not matter where in the system these variables are, or in what order the equations are written. The language can though be very sensitive to initial conditions and the dynamics of the equations. It must be made sure that each individual component works fine and that the system is expanded model by model.

Using the Modelica language, the user can write individual system components with its detail encapsulated in the model. Each model can then transfer properties and variables through connectors specified in the model, as long as the connectors are compatible. Models can be stored and individual libraries can be made and used in different applications, together with other libraries and models. The language in itself is not built for a special engineering field, and can therefore be used to model any kind of processes or system. [2]

#### 2.1.1 Dymola

Dymola is a simulation environment built upon the Modelica language. A graphical interface where Modelica models can be used, constructed, put together and be simulated. It is developed and distributed by Dynasim AB in Lund, Sweden. Apart from proprietary packages, Dymola use exporting features to import the standard Modelica library and use it as a base package. The graphical interface makes it easy to build large complex systems by connecting smaller models representing different kind of processes instead of writing all the equations in one big model. The Modelica standard Library that is included has models describing all basic mathematical and physical functions, constants, tables and units. [3, 4]

#### 2.1.2 Solvers

In the Dymola interface the user have a large number of solver algorithms to choose from, suitable for different simulations. If there are few events occurring in the simulation, variable step size algorithms are the most efficient, otherwise single-step. The choice of algorithm should depend on the CPU time, stability and accuracy of the obtained simulation with different algorithms. The default solver used in this thesis work is the Dassl solver, which is a variable step size algorithm. Apart from the solver the user also has the possibility to change the iteration step size, the maximum error and the tolerance of the simulation. Increasing the tolerances will fasten up the simulation to the cost of accuracy. A big disadvantage is that this solver needs very close initial state guess values to not diverge at the first initialization step of the simulation. If there exist strong non-linear behaviour in the model, the model is even more unstable. The initial values should therefore be defined as close to steady state as possible. [5]

### 2.1.3 Simulation

When a simulation is carried out, Dymola translates the Modelica language to C-code with all the necessary information to perform the simulation including all possible features. The C-code is stored in a file Dynasim “*Dynamic model simulator*”.

### 2.1.4 Media Library

The Modelica media package is a sub-library of the standard library. It contains high precision models and properties of numerous substance fluids. It allows the calculation of all the important fluid properties, including partial derivatives of thermodynamic functions. Simple or multiple substances as well as one or multi-phased substances are available. If one of the combinations of the following state variable inputs are given (p,T, (x)), (p,h, (x)), (d,T, (x)) or (p,s, (x)) a specific thermodynamic state is obtained.

Where:

- T = temperature [K or °C]
- p = pressure [Pa]
- (x) = dryness factor or quality (mixtures) [-]
- s = entropy [kJ/(kg · °C)]
- h = enthalpy [kJ/kg]

The media package can then be used to compute all the other properties of the media from the thermodynamic state.

In this thesis only the properties of water in different phases has been used, that is the (IAPWS/IF97) models. These models uses the properties of water defined in IAPWS IF97 “*The International Association for the Properties of Water and Steam*”. [6, 7, 8]

### 2.1.5 The Steam Power Library

This library has been completely developed by Solvina alone and consists of different components that may be used in all kind of power generation, done from steam. E.g. nuclear power plants, CHP and paper mills. Since all the components are made in-house some may lack the extensive user guide information that the standard libraries of Dymola have.

## 2.2 Frequency Regulation

In the grid, a balance has to be maintained between electricity generation and consumption in order to keep the frequency at a desired value. In the northern countries, the grid is connected in such a way that it can be seen as a big island operation, meaning that the generators act as if they were mounted on the same shaft. At any time when the electricity consumption exceeds the production, proportional to the mechanical work, the shaft will begin to rotate slower and the frequency will decrease. In the opposite way, when the production of electricity exceeds the consumption the frequency will increase and the shaft rotate faster. [1]

When the frequency deviations become large, some components may be damaged. If the system frequency would coincide with the resonant frequency of the turbine blade or turbine shafts, the plant would have to be shut down to avoid severe damage. In addition, transformers and generators are tuned and dimensioned to operate at a normal frequency of 50 Hz. Too big deviations and the windings may become overheated and the unit may be damaged. Steam turbines are exceptionally sensitive and should not under any circumstances operate at frequencies below 47.5 Hz, since it may lead to harmful vibrations in the mechanical components. An exception is hydropower plants that can handle frequencies down to 45 Hz without being harmed. Moreover,

the majority of the devices and all kind of electrical devices connected to the power system is designed to work at a nominal frequency of 50 Hz and will, if not get damaged; have its function decreased in case of frequency deviation. [9]

Over a long period of time, it is also important that the number of 50 Hz cycles will be exactly the nominal number. This is because a lot of devices such as clocks uses the main frequency as a reference, which means that some clocks or devices can go relatively wrong over a few hours but be corrected over days.

To balance the frequency, mechanical work needs to be applied or removed in power generating turbines connected to the grid. This is called frequency- regulation or control. Frequency control can also be achieved through other types of technologies like batteries and fuel cells. For now these technologies are either too expensive or are missing a working infrastructure, and in this report frequency control will only be considered by means of power generating turbines.

Depending on the process equipment and the system characteristics of a plant, different suitable control structures are implemented to control the turbine. The control system will react on the rotational speed and regulate the steam supply thereafter, but if the control is too fast it may damage the turbine and/or the system. A ramp function will be implemented to limit the response.

The ramp function is measured as a proportionality between the frequency deviation and the corresponding power change. One way to measure it is to divide the power change in megawatt by the frequency deviation in hertz:

$$ControlPower = \frac{\Delta P [MW]}{\Delta f [Hz]} \quad (2.1)$$

Equation 2.1 gives a function of how many megawatts of Power that will be increased by the turbine for every hertz of deviation.

Another way used to measure FCR is by dividing the frequency deviation in percent of 50 Hz by the percent of the rated power used to achieve that.

$$Droop = \frac{\Delta f [\%]}{\Delta P [\%]} \quad (2.2)$$

In this way you get a measure of how much the frequency must deviate in order for the plant to change its power output to the maximum of rated frequency control power. That is, the frequency change (in percent) of 50 Hz that results in a turbine opening shift of 100%, called droop.

The droop is normally used to determine each power station's value of participation in the frequency regulation. The distribution of "regulatory action". This value may vary from 0 to 100%, where one means that the generator or the power station strives to independently maintain the frequency at nominal value. 100% means that it has no participation at all and runs on a constant power output. In large systems like the Nordic network, a reasonable droop value could be 4%, which means that a frequency deviation of 4% results in a 100% change of power output. This droop value will vary a lot depending on the power facilities ability to respond to large and rapid changes. A water turbine can respond relatively quickly while a nuclear power station cannot, and the droop setting will then be set in accordance.

In terms of time delay in the system it's important to understand that in a turbine shaft and the coupled generator there is a natural inertia against frequency regulation. There is a small amount of stored energy in the rotating mass of the turbines and generators. This stabilizes the system, meaning that there is not an instant frequency change in the system when the load changes. The control structure of the turbine will play the biggest role in the time delay, ranging from seconds to minutes to full proportional response that depends on the implemented droop. The most used control structure is a PI-controller which suit both small and big networks, with both a precise and smooth response.

### 2.2.1 Primary Frequency Regulation

Primary frequency regulation responds to sudden, rapid load changes and is implemented automatically to counteract the deviations in the grid including variations from intermittent energy sources like solar and wind.

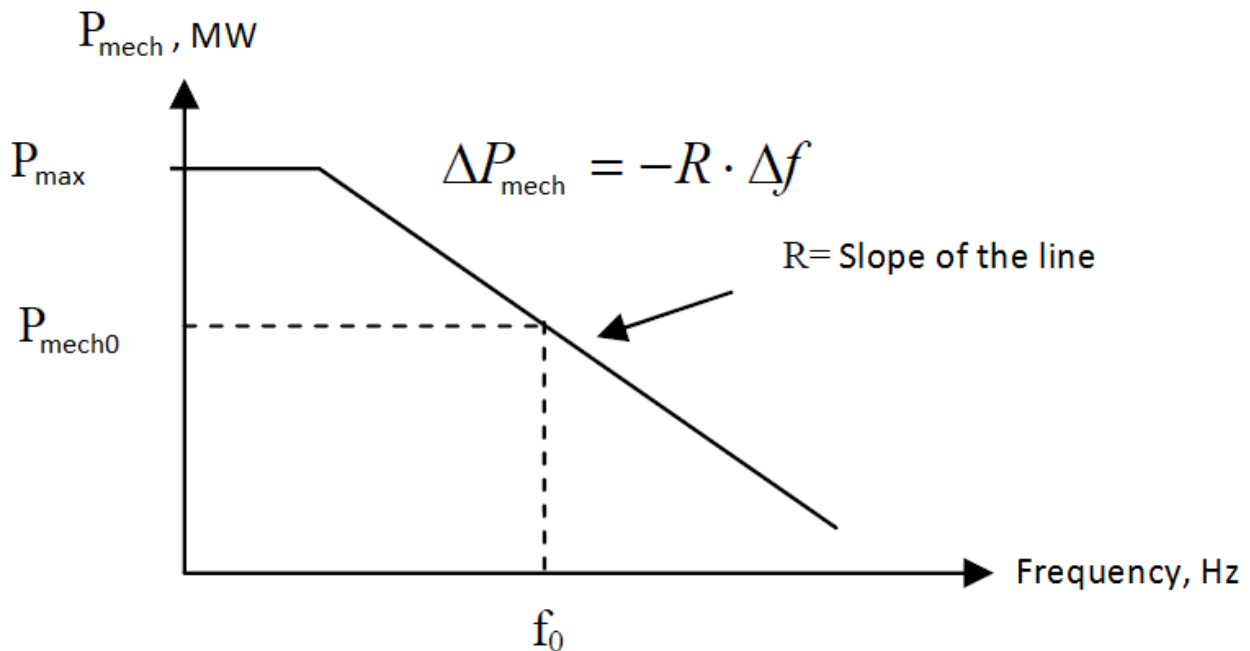


Figure 2.1: A graphic presentation of  $R =$  control power, representing the slope of the line.

If there is a sudden frequency deviation in the system, plants responsible for the PFR will in a matter of seconds increase or decrease their power production. How much and how fast they change will depend on the implemented control structure and chosen droop, as mentioned in section 2.2.

The PFR includes both control for small and large deviations which are declared as FCR-N and FCR-D respectively. In this report FCR-N will only be considered, which is the smallest variations (within  $\pm 0.1$  Hz of 50 Hz). This kind of regulation puts certain requirements on response time as well as retained response time on the turbine and the generator.

### Requirements for FCR-N

ENTSO-E has established a table of requirements on the plant for participation in FCR-N:

Table 2.1: The requirements for participation in the primary frequency regulation, FCR-N derived from ENTSO-E (2013).

1.	<b>Frequency sensitivity</b>	Max 10 ... 30 mHz
2.	<b>Deadband</b>	0 ... 500 mHz
3.	<b>Droop</b>	2 ... 12%
4.	<b>Turbine control range</b>	$\pm 1.5 \dots 10\%$ of electrical power (from the setpoint value)
5.	<b>Response</b>	Initiated within 2 seconds of frequency deviations
6.	<b>Full response</b>	Within 30 seconds after the frequency deviations
7.	<b>Retained response</b>	For 15 ... 30 minutes, both for the remaining frequency deviations and repeated frequency step

The first three requirements are only dependent on the control structure and are easily fulfilled. The remaining requirements are of more interest and dependent on the structure of the processes in the plant. Initially it sets requirements on the turbine and secondly on the processes affecting how fast accumulated steam can be redistributed and produced in the system. [10]

### 2.2.2 Secondary Frequency Regulation

If there are remaining frequency deviations after implementation of PFR, SFR will be implemented with a couple of minutes delay to return the frequency back to 50 Hz. This regulation will be infinite until other remaining frequency deviations occur. When the SFR is put into action and the frequency is returned, the power generating units output for PFR returns to its setpoint and thus the regulating power of PFR is restored. SFR also takes care of the overall balance throughout the day or in e.g. case of sudden shut down of a certain power plant.

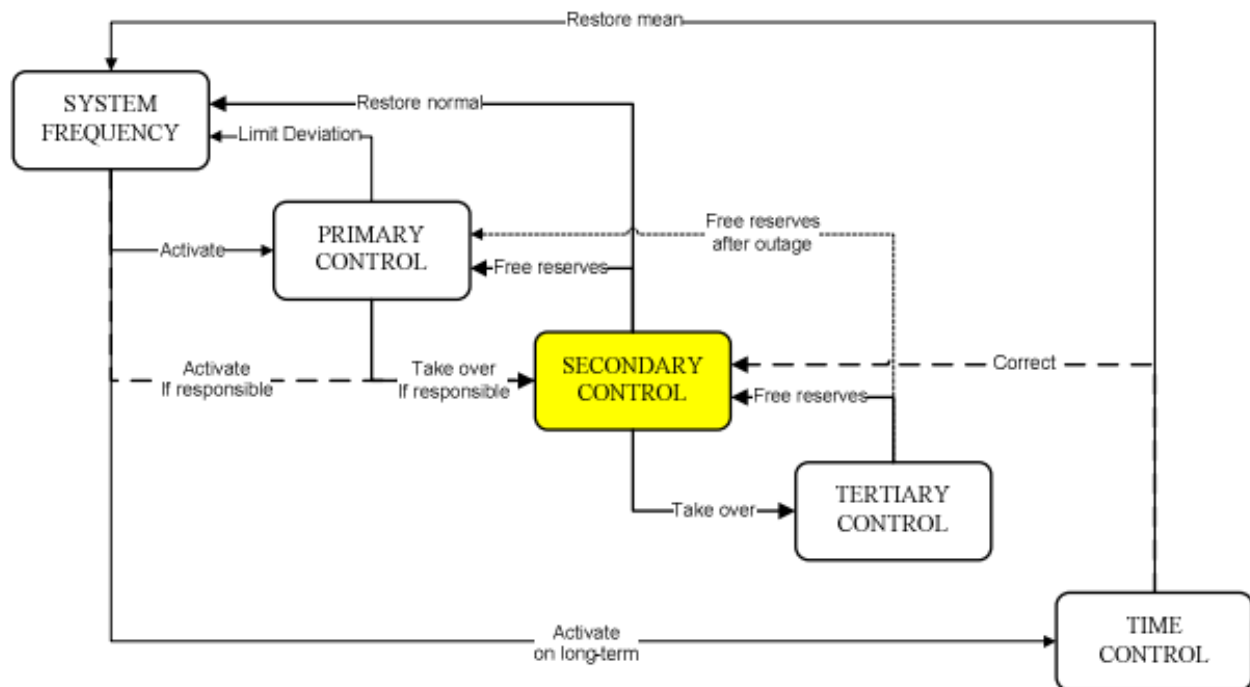


Figure 2.2: A basic control Scheme for FCR. [11]

## 2.3 Pulp and Paper Mills

Pulp and paper manufacturing is a complex procedure, that contains a lot of processes and can be done in a number of different ways, either mechanically, chemically or thermo- mechanically.

In the pulp mill wood chips are converted into fibreboards which then are transferred to the paper mill. In this process there are bark and wood chip residuals that are reused as fuel in the paper mill's boilers which primarily produces steam for the processes in the paper mill and secondarily they produce power, if a turbine is installed. The main goal of the pulp mill is to liberate fibres from the wood. This could be done either mechanically or chemically or thermo- mechanically mentioned above. This is done chemically in Iggesund mill and the process is also called delignification. In chemical pulping only about half of the wood is converted and the rest dissolved, hence the need for recovery. The chemicals used to liberate the fibres and the dissolved wood particles, are separated and the residue is known as black liquor. The most common used delignification method is called the Kraft process, in which the delignification is done by use of white liquor, that when recovered also contains dissolved wood particles and lignin from where it get its darker colour, hence the name black liquor.

The pulp extracted from the pulp mill will need to be bleached to get the desired white colour, how much will depend on the composition of materials and chemicals used. The pulp is finally washed before it is sent to the paper mill. [13, 14]

In the paper mill, the pulp together with a number of additives is put into a paper machine where the paper is produced through a number of steps. Press section, Drying section, calendaring, coating and reeling.



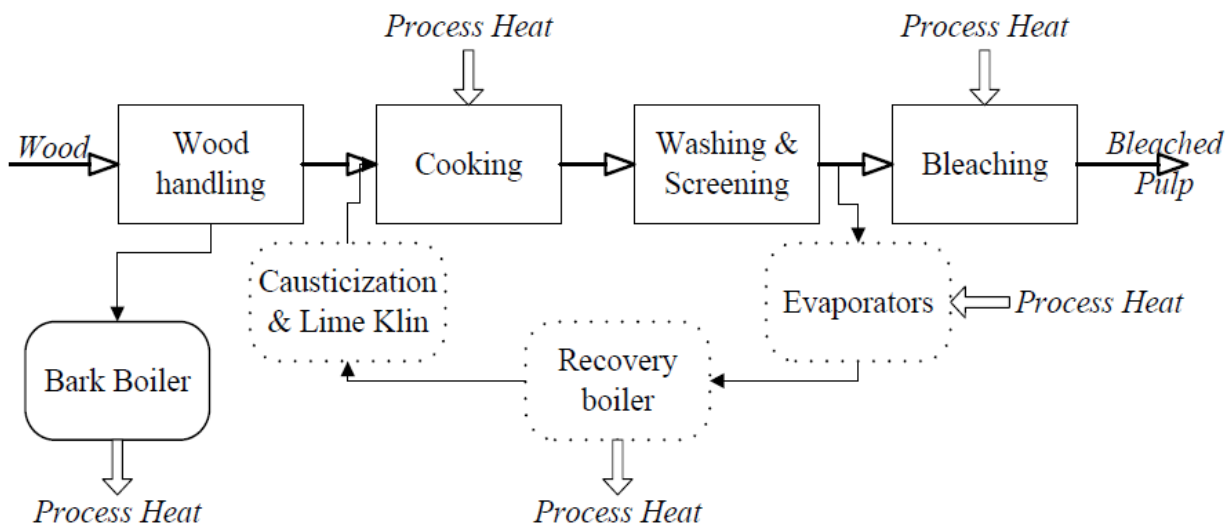


Figure 2.3: A block presentation of the Kraft process, with dashed blocks representing the recovery cycle. [12].

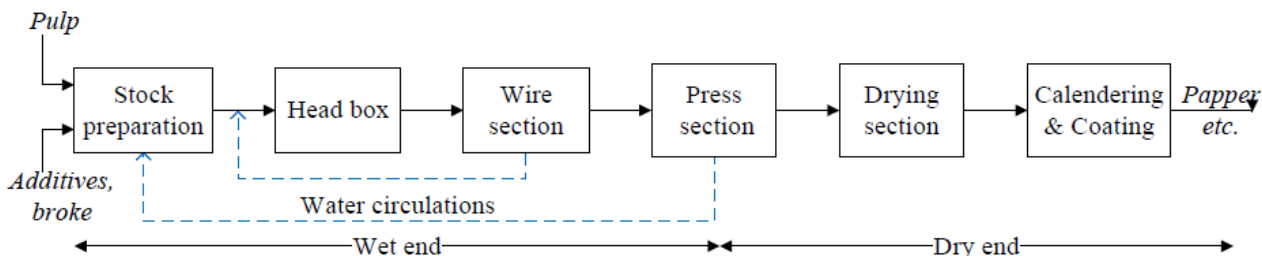


Figure 2.4: A block presentation of a paper machine's processes. [15]

## 2.4 Steam Nets

In a pulp and paper mill there is a constant need of energy, in this case, steam of diverse states to drive the different processes. There is also a need to recover or reuse the chemicals used in the process, and get rid of by-products such as bark from the paper making process as mentioned in the pulp and paper mill section. Boilers are used to extract the energy in the by-products and to separate the chemicals to be reused in the process. The needed process steam is covered by the recovery boiler and in the case of Iggesund with additional oil- and bark boilers. The chemical energy bound in the fuel is converted to thermal energy in superheated steam and extracted and used as heat to drive the cooking section, the bleaching section, the evaporation of black liquor and the drying section in the paper machine. The superheated steam coming out of the process is often at a much higher pressure than needed in the process. The pressure is lowered through valves, accumulators and/or turbine stages to the different pressure levels and mass flow rates needed in the process. In turbines electricity can be produced as a by-product and either be used in the process or be sold.

### 2.4.1 Steam Turbines

A steam turbine is by definition a machine that converts or transfers thermal energy from the dynamic action of flowing pressurized steam on spinning blades, to the energy of mechanical work on a shaft, by expansion to a lower pressure, also called head. In steam turbines the medium cannot be considered as an ideal gas, properties at a specific state then have to be taken out of a Mollier chart or a steam table. [16]

The work output of a single extraction pressure steam turbine can be determined by multiplying the steam

mass flow rate with the change of enthalpy over the stage:

$$W = \dot{m} \cdot (h_1 - h_2) \quad (2.3)$$

In equation 2.3:

- $\dot{m}$  = Steam mass flow rate [kg/s]
- $h_1$  = The specific enthalpies at inlet [kJ/kg]
- $h_2$  = The specific enthalpies at outlet [kJ/kg]

The enthalpy change ( $h_1-h_2$ ) can be calculated using an expression for the isentropic efficiency:

$$\eta_{iso} = \frac{h_1 - h_2}{h_1 - h_{2iso}} \quad (2.4)$$

In equation 2.4,  $h_{2iso}$  is the specific enthalpy of the steam outlet considering isentropic conditions. The enthalpy can then be calculated with known temperature and pressure of the steam at the specific state.

The massflow can be calculated using Stadola's turbine equation [17]:

$$\dot{m} = C_t \cdot \frac{p_{in}}{\sqrt{T_{in}}} \cdot \sqrt{1 - \left(\frac{P_{out}}{P_{in}}\right)^2} \quad (2.5)$$

Where Stadola's constant ( $C_t$ ) is a unique property of the specific steam turbine. It is a measure of the effective flow area through the turbine and its value can be found empirically. Stadola's constant is a function of the following time dependent parameters:

$$C_t = f(\dot{m}, P_1, P_2, T_1) \quad (2.6)$$

Stadola's constant can also be expressed as a function of the following fixed mechanical parameters and properties.

$$C_t = \frac{\alpha \cdot f_{sp}}{\sqrt{z \cdot R}} \quad (2.7)$$

Where:

- $\alpha$  = flow coefficient [-]
- $f_{sp}$  = turbine constriction cross-section [-]
- $z$  = number of stages [-]
- $R$  = molar gas constant [ $m^2 kg s^{-2} K^{-1} mol^{-1}$ ]

In the model, once Stadola's constant is set, the pressure drop over the turbine will be altered by the steam massflow that comes in. This is done by control of steam valves.

To summon up the turbine work output can be expressed as a function of the following five parameters:

$$W_{turbine} = f(c_t, P_1, P_2, T_1, N_{iso}) \quad (2.8)$$

The power output will then be a product of the generator efficiency.

Going back to the definition of Stadola's turbine equation, a long known relationship for any point in the expansion in a stage is: [18]

$$\phi = \frac{\dot{m}_i}{\sqrt{T_i P_i}} = Constant \quad (2.9)$$

Where:

- $\phi$  = mass flow coefficient [-]
- $\dot{m}_i$  = mass flow to the next stage group [kg/s]
- $P_i$  = inlet bowl or shell pressure [Pa]
- $T_i$  = absolute temperature at the specific point [K]
- i = subscript denoting any point in the expansion [-]

Since the steam is overheated through the whole passage of the turbine one can assume that the ideal gas law applies, and equation 2.9 can be changed into:

$$\phi_1 = \frac{\dot{m}_i \cdot \sqrt{T_i}}{P_i} = Constant \quad (2.10)$$

This gives a different, yet widely used expression for the mass flow coefficient. Another known relationship or law is the law of the ellipse, also called Stadola's cone law. [18]

$$\phi_2 = \sqrt{1 - \left(\frac{P_{out}}{P_{in}}\right)^2} \quad (2.11)$$

This law is based upon the relationship between the mass flow, the inlet pressure and outlet pressure of a turbine stage that Stadola found experimentally. The name of the law comes from the fact that when these parameters are presented in a Cartesian coordinate system, with the inlet pressure fixed on the x-axis. The outlet pressure and mass flow forms an arc of an ellipse between the y- and z-axis.

Combining equation 2.10 and 2.11 and setting a constant  $C_t$ , equivalent to the difference in massflow coefficients in both equations one get the following relationship presented earlier, known as Stadola's massflow equation.

$$\dot{m} = C_t \frac{p_{in}}{\sqrt{T_{in}}} \sqrt{1 - \left(\frac{P_{out}}{P_{in}}\right)^2} \quad (2.12)$$

This equation is based on a 100 years old experimental data and although the turbine design is almost unchanged over the years the equation still only relies on small pressure differences. The working pressure difference in turbines is now much higher than it was then, and therefore the equation is getting somewhat outdated. When high pressure drops occurs in the turbine stage the flow often becomes choked and the relationship between mass flow and pressure then does not apply. Once the flow becomes choked, the mass flow remains constant regardless of pressure difference and it is only dependent on the pressure at the inlet. For the model to be accurate, the mass flow would have to be calculated at the very point where the flow gets choked. On the other hand, it requires a lot of information about the turbine working condition to know when the flow is choked or not. When using Stadola's turbine equations in Dymola, this will be a large source of uncertainty.

Also, when the turbine is working off-design, the isentropic efficiency is not compensated for and considered to be constant. This will give a higher work output than it would in reality, for small massflows. [3]

### 2.4.2 Boilers

The definition of a boiler is a mechanical component that transfers the energy chemically bound in fuel, electrical energy or energy stored in atoms to steam of a desired state.

Steam pressure is regulated by either increasing the fuel input or the steam mass flow rate. If as much steam is released as is formed, the pressure of the steam remains constant, if not, the pressure will change proportional to the change of the released steam flow. The same analogy applies for the fuel and steam.

A basic design of a boiler consists of equipment for both the air-flue gas side and fuel side and for the water and steam side. Except the vessel and combustion chamber the most crucial parts or most important to model are those on the steam side. Feed water pumps, steam drum, an economizer-, evaporator- and a super heater section. [19]

The feed water enters the economizer in the flue gas exit of the boiler furnace and is preheated at close to saturation temperature. In some cases the feed water is preheated using steam bleeds from the turbine. The feed water then flows to the steam drum and then to the evaporator part of the boiler furnace. This is closest to the flame and the processes where most heat is absorbed to evaporate the water. The saturated steam then goes back to the steam drum which acts both as a reservoir and as a phase-separator for the steam and water mixture, where water is separated from the steam. The density difference between steam and water naturally helps the separation of water. Dry steam is then led from the top of the drum to the superheater section to be further heated to its design temperature. Saturated water in the bottom of the tank is fed back to the feed water tank. Air is often preheated in an air preheater section at the end of the flues in the furnace of the boiler. See figure 2.5 for a simplified illustration.

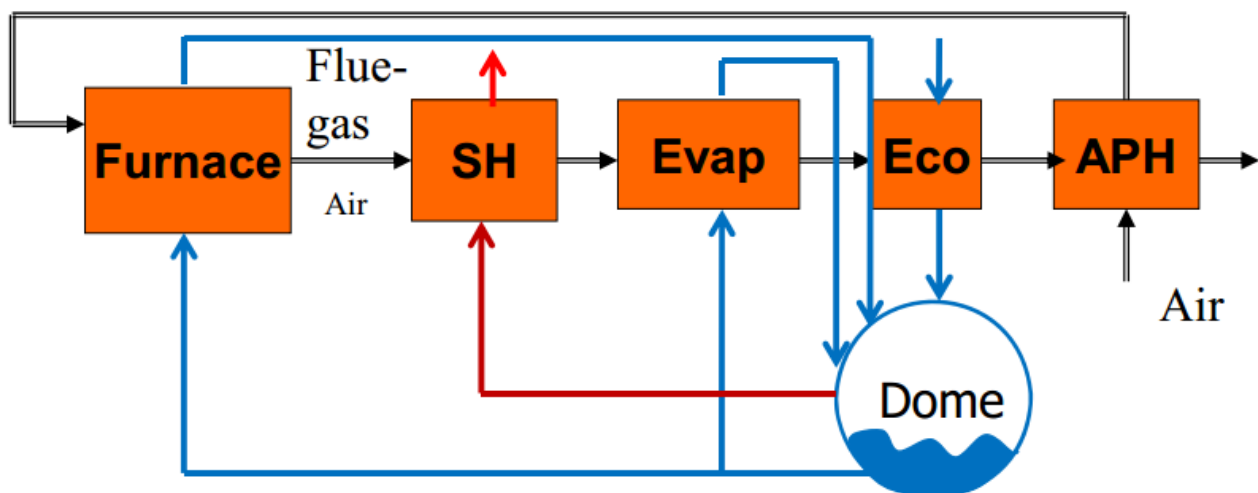


Figure 2.5: *The structure of a simple boiler.* [19]

The water and the tubes act as a coolant and the heat absorbed by the coolant must meet the heat generated through the combustion in order for the boiler to function as it should. A too high temperature will damage the equipment and too low will not make the combustion process self-sustaining. In the drum there must be both steam and water. Only water in the drum would indicate that more fuel should be injected or less water fed in, and vice versa for only steam.

What is of interest here is the time it takes for the boiler to transform the feed water to superheated steam of desired state, the time constant of the process. This is dependent on the design of the boiler and most importantly the fuel that is used.

### 2.4.3 Fuel

The boilers used in pulp and paper mills are at least one recovery boiler where the black liquor is combusted and often at least one boiler to combust bark.

## Oil

Oil used as fuel consists of varying compositions of hydrocarbons which are vaporized and separated upon heating. The amount of heat needed and at what temperature depends on the composition of the hydrocarbons. For the oil to combust properly it must be perfectly mixed with a good preheated air supply, and then rapidly heated to high temperatures. At these high temperatures splitted products can burn fast and a short hot flame is achieved. To accomplish this so called atomization of oil, special oil burners are used to mix the oil and combustion air. At the right conditions the oil droplets burns very rapidly, and can thus allow high space loading. Burning oil in the boiler will give a relatively fast response, and will in that manner be the preferred fuel for frequency regulation. [19]

## Bark

When the moisture content is high bark, chips and sawdust are preferably burned on a plane or sloping grate. The moisture content of the fuel is crucial for both the choice of equipment and the dynamics of the process. The bark will however burn relatively slowly, and thus give a slow response to the boiler in case of a fuel ramp up.

When bark has got a lower heating-value than 7.3 MJ/kg, additive fuel must be used. In this case oil.

To allow a much faster regulation of the boiler, which would be of interest for frequency regulation, the bark could be dried and pulverized in another instance before entering the boiler, increasing the efficiency, combustion temperature and combustion time. [19]

## Black Liquor

Black liquor is the name of the residue of the used liquor in the pulping process. This is burned in large recovery boilers and converted into new cooking chemicals and energy. This residue consists more or less of pulping chemicals, lignin and water (20-40%). Over the years, to increase the performance of recovery boiler the water content of the fuel has decreased which also sets new standards on the design of the boiler and spraying practices.

Black liquor has almost got the same heating value as bark, chips and sawdust mixture, but with a higher combustion rate. [20]

### 2.4.4 Steam Accumulators

A steam accumulator acts as a storage device for steam. Normally it is filled half way by water when not in operation. Steam from e.g. a boiler at desired pressure is then blown in from the bottom. Some of the steam will be condensed and used to heat up the water. Equilibrium will eventually be reached between the steam and the water, at a specific temperature and pressure, a specific state. If more steam is injected more water will be condensed and a higher order of state of the steam will be reached. That is a higher pressure and temperature. The accumulator tank is normally considered fully loaded when three quarters of the tank is filled with water. When steam is discharged from the top of the tank, the pressure will gradually go down along with temperature and more water will boil off and the tank will eventually have to be refilled with steam again.

The purpose of the steam accumulator is mainly to counteract pressure deviations in the medium- and low pressure net. That is, make sure that the accumulated steam needed in the processes remains constant and the system to be less sensitive to disturbances. If there is access of steam in the MP- or HP- net the accumulator will be filled up with steam and if there is a deficit of steam in the LP-net the accumulator will release steam to the LP-net. If the tank is full and there is still an excess of steam, the steam will be released to the atmosphere from the LP-net through a blow off valve.

### 2.4.5 Steam Saturators

After each valve that expands the steam down from the high pressure net to the medium- or low pressure net there has to be a steam saturator to cool the steam. Superheated steam would damage the pipes or any other equipment and processes that work with much lower temperatures. Saturating the steam is also done for better heat transfer. When the steam condenses, very high heat transfer coefficients are achieved. The steam saturator works in a way that it sprays a certain amount of water on to the steam until the desired steam temperature is reached. The steam coming out from the saturator has the enthalpy of the desired state with a lower temperature than the incoming and a higher mass flow. The steam temperature must be above the saturation temperature.

### 2.4.6 Valves

Valve models basically calculate the flow rate through the valve as a function of the upstream and downstream pressure. They are used to regulate the desired mass flows in the system. Closing a valve will decrease the flow through it and build up a pressure at the valve opening according to equation 2.13:

$$\dot{m} = C_v \cdot k \cdot \alpha \cdot \sqrt{\rho \cdot \Delta p} \quad (2.13)$$

Where:

- $\alpha$  = the opening of the valve [-]
- $k$  = a conversion factor [-]
- $\rho$  = density [ $kg/m^3$ ]
- $\Delta p$  = pressure difference [Pa]

The  $C_v$  -value, also known as the valve flow coefficient is used as a parameter to indicate the capacity of the valve when it is fully open, and will be a function of the parameters presented above. The corresponding European measure is a  $K_v$ -value and the difference is only by the definition of units.

With the right control structure, if mass flow goes down due to less fuel input, the valve will be shut accordingly to keep the desired pressure or pressure drop. The same goes for if the mass flow is increased. For a good control, the valve opening should be about half open during normal operation.

### 2.4.7 Pipes

Pipes are used to transfer water and steam between processes in the system and will account for a certain time delay and accumulation.

## 2.5 Control Systems

The most common type of control in the process industry is feedback control. A set point signal is sent into a controller together with the actual measured value in the process. The difference between the actual measure and the set point forms an error and the error is then used to regulate the system. When the actual value in the system corresponds to the set point signal, there will not be any error and no regulation. The structure or design of the controller will depend on the dynamics of the process and the desired control. A block presentation of this is given in figure 2.6.

A common way to calibrate the controller is to introduce a step response into the set point signal and then observe the dynamic output signal of the process over the period of time. The controller is then tuned to get the desired output dynamics. This procedure will be iterated until satisfaction. A frequently used controller in the process industry is a PI-controller. It is cheap to buy and fairly easy to implement and tune. It is

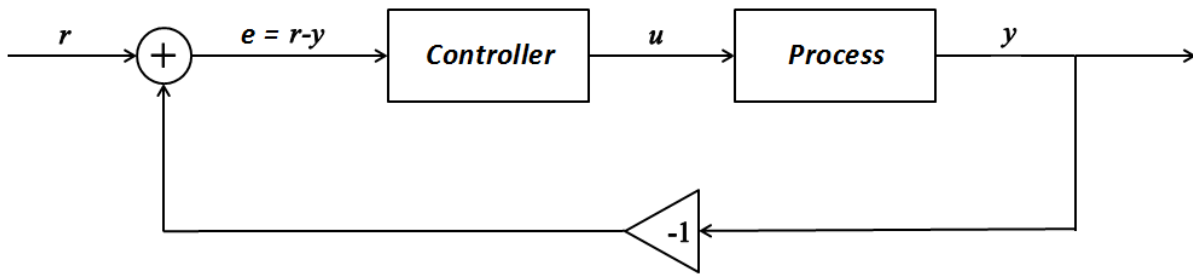


Figure 2.6: An overview of the feedback control process.  $r$  is the reference signal, and  $y$  the real measure.  $e$  is the error and  $u$  the control value.

preferred to have several PI-controllers controlling different processes than one bigger and more complex one. The abbreviation PI comes from the fact that the controller consists of two different parts. P stands for the proportionate part and I for the integrating part. [21]

In theory it can be expressed through equation 2.14:

$$u(t) = \underbrace{K_p e(t)}_P + \underbrace{\frac{K_p}{T_i} \int^t e(s) ds}_I \quad (2.14)$$

Using Laplace transformation the transfer function between the error and the control signal will be according to equation 2.15:

$$F_{PI}(s) = \frac{u(s)}{e(s)} = K_p \left( 1 + \frac{1}{T_i s} \right) \quad (2.15)$$

The proportional part represents the gain and will minimize the control error but depending on the size of the gain, there will always be an error. The bigger the  $K$ -value is the bigger the gain is and the smaller the control error. Conversely, as  $K$  gets bigger the risk of stability issues increases. The integrating part is essential to more or less eliminate the error but to the cost of further decreased stability margins. A derivative part can be used to increase to stability margins but to a cost of increased influence of measurement errors giving a somewhat noisy output signal. This can however be filtered away. A block presentation of a PI controller can be seen in figure 2.7. [21]

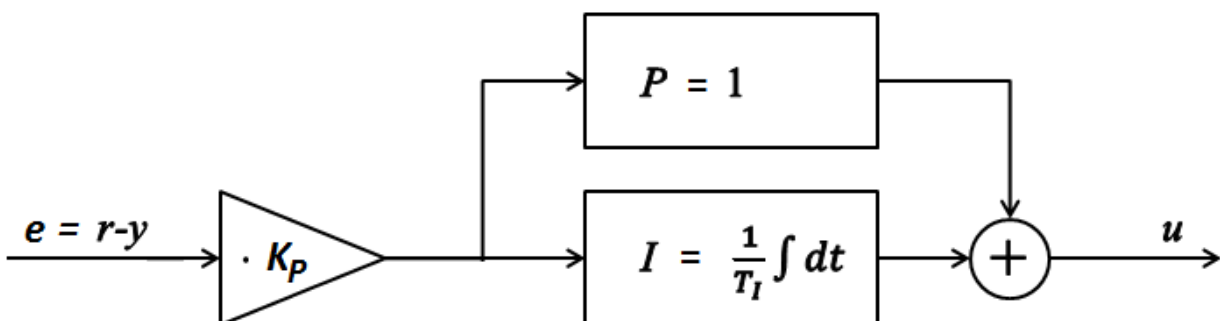


Figure 2.7: The structure of a basic PI controller where  $e$  is the error and  $u$  is the control signal.

### 2.5.1 The Lambda Tuning Method

The control of the plant covered in this thesis work, will mainly be done with PI - controllers.

An easy way to tune a PI - controller is to guess the values of gain and integral parameters, see how the process responds with different settings and tune thereafter. However, this is not the best method if you are looking for an optimal tuning. It is also rather difficult to implement and to get good results. A recommended method for tuning of PI - controllers is the so-called Lambda method.

Using the Lambda method, you study how the controller (in manual mode of operation) reacts to various changes in the control signal, for example with a ramp function. A relationship between the static gain of the process, the gain of the controller, and the time constant is set, and changed by the change of a parameter  $\kappa$ . The controller is then tuned by trial and error with different values of  $\kappa$  until desired a response is achieved.

The method can be summarized into one simple formula:

$$K_c = \frac{T}{K \cdot (L + \lambda)} \quad (2.16)$$

Where:

$$\lambda = \kappa \cdot T \quad (2.17)$$

$$K = \frac{\Delta y}{\Delta u} = \frac{\Delta y \cdot (Y_{max} - Y_{min})}{\Delta u \cdot (U_{max} - U_{min})} \cdot 100 \quad (2.18)$$

Where:

- $\lambda$  = Lambda
- $\kappa$  = Lambda factor
- Y = Setpoint
- U = Process output
- L = Deadtime
- T = Time constant
- K = The gain of the process

$K_c$  equals the static gain of the controller and  $\lambda$  is a measure of how fast the regulation will be.

K is initially set using equation 2.18. The time constant T of the process after the first step response is set equal to the time constant parameter of the integrator block,  $T_i$ . The lambda constant  $\kappa$  and indirect  $\lambda$  is finally set as an arbitrary value depending on the characteristics of the controller that you want.

- $\kappa < 1$  = Aggressive
- $\kappa = 1$  = Neutral
- $\kappa > 1$  = Defensive

The advantage of having an aggressive controller, a low  $\kappa$  value, is that the control error is eliminated faster. The process may however become unstable at large variations in gain and deadtime, if the plant is operating at other conditions than the regulation it is tuned for. Thus, poor robustness can lead to self-oscillation.

If for ease a  $\kappa$  of one is set, thus a neutral controller and zero deadtime is assumed, equation 2.16 can be simplified to:

$$K_c = \frac{1}{K} = \frac{\Delta u}{\Delta y} \quad (2.19)$$



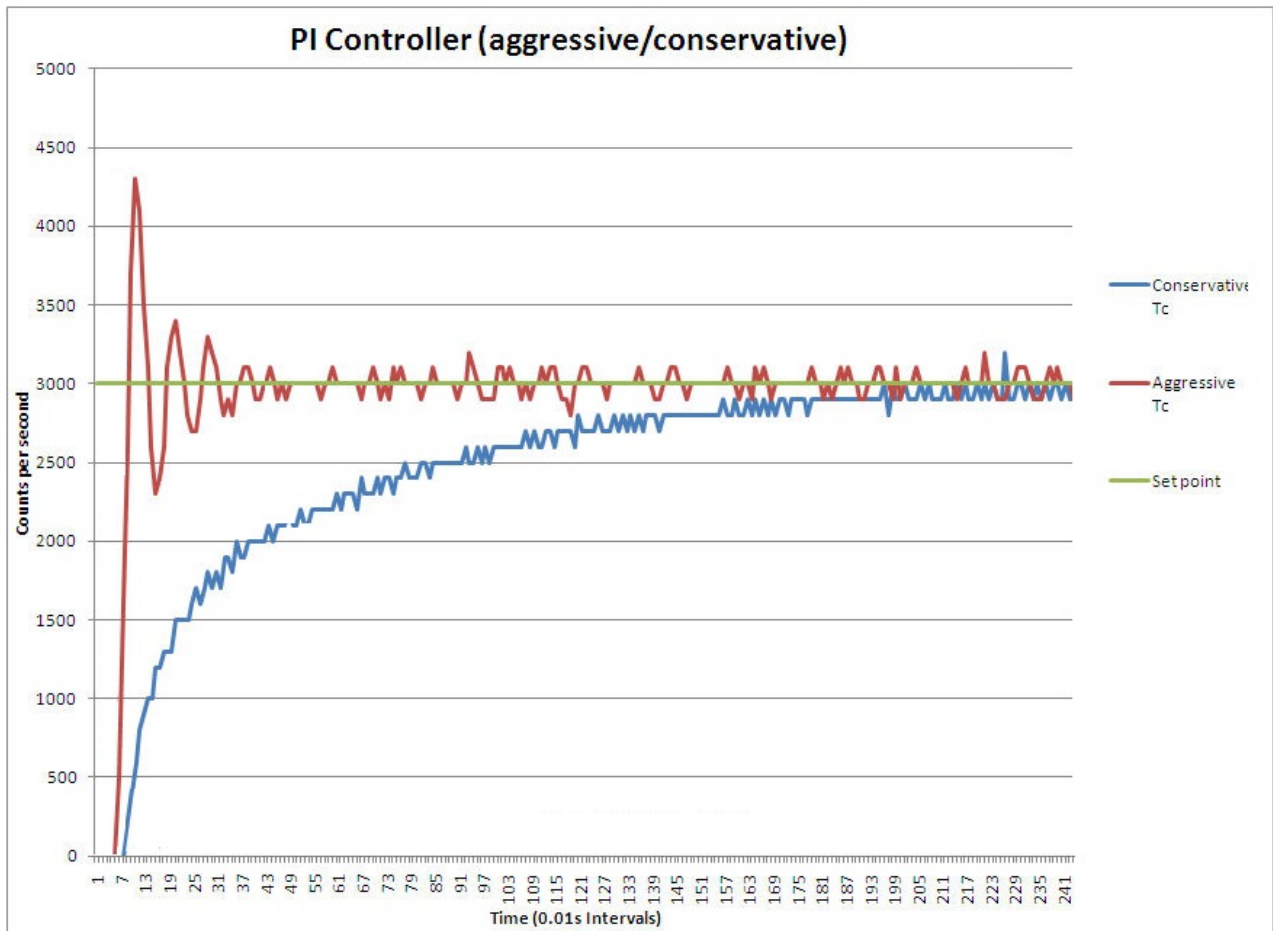


Figure 2.8: *The difference between an aggressive and a defensive controller.*

A recommended start value for  $\kappa$  is 1.5, or between 1-2. If it turns out that this value does not lead to any stability problems in any of the possible operating modes, one can test to reduce  $\kappa$  and get a faster structure. [22]

### 2.5.2 Feed-Forward

If a process parameter regulated by a specific controller is fluctuating to the extent that tuning the controller more aggressively makes it unstable, and less aggressively makes the response too slow, a feed-forward structure can be implemented. This means that information from a known disturbance is used as an extra measure to control a specific sequence in the regulated process itself. An additional feed-forward term from the external process is added to the control signal in the feedback loop, to fasten up the internal process before it gets out of hand. [22]

### 2.5.3 Cascade Control

If a process is controlled by a single controller, a master controller and a response in a specific part of the process needs to be fastened up, e.g. a part with relatively large disturbances, cascade control can then be a good option to implement. The output value of the master controller is then sent to be the set point value in a slave controller. This means that you have an internal feedback loop over a faster subsystem inside the system. If there is an error or disturbance in the subsystem, the inner loop and slave controller measure and corrects for this error much faster than what the outer loop would have done. Cascade control requires additional measurement equipment and controllers but will increase the response time of the specific system. [21]

### 2.5.4 Split-Range Control

As the name suggests, a split range control means that the output of a controller is divided and scaled into two or more control objects. The splitter defines in which order and to what magnitude the controlled objects should be controlled as the main controller changes its output from 0 to 100%. Ideally, you want to achieve a constant gain through the whole working range regardless of which object that is controlled. This is done by scaling.

As an example, to reduce the pressure in the MP-net at Iggesund Mill, it could either be done by opening of a steam accumulator valve, a valve to throttle the excess steam down to the LP-net or to release it through a blow-off valve. To utilize the steam fully the split has in principle been sequenced as can be seen in figure 2.9. To achieve a constant gain or a linearly proportional response between mass flow and controller output, the slope of the lines has to be less steep for the bigger valves than the smaller, and vice versa.

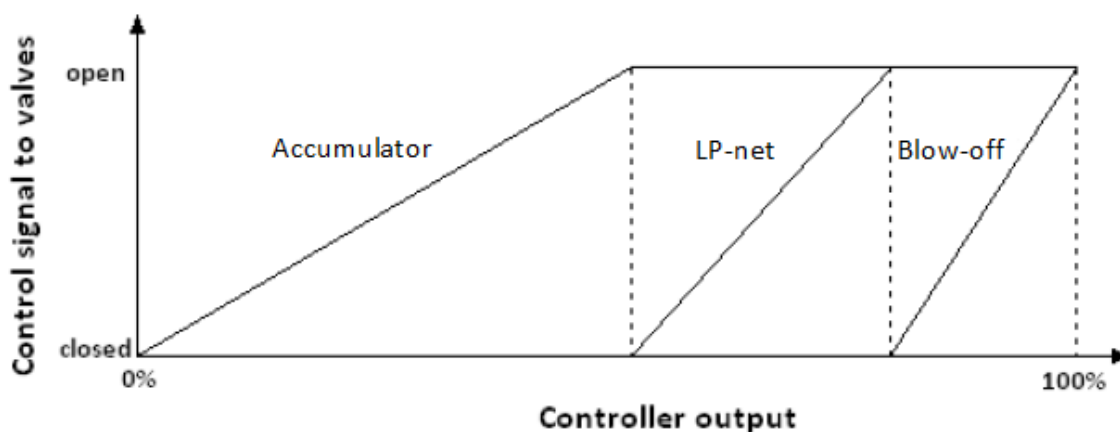


Figure 2.9: A principle split range control of valves.

This method is commonly used in systems with more than one actuator, being active in the same control range. The advantage of this type of control is that a single controller can be used to control several actuators in the system.

## 2.6 Concepts for Frequency Control

There are multiple ways of designing the control structure for optimal frequency control. Looking at figure 2.10 one can get a good estimate of the number of parameters to control that in different ways are a function of the power output of the turbine. The most convenient way to control the power output of the turbine is by controlling the inlet pressure of the turbine through throttling. [23]

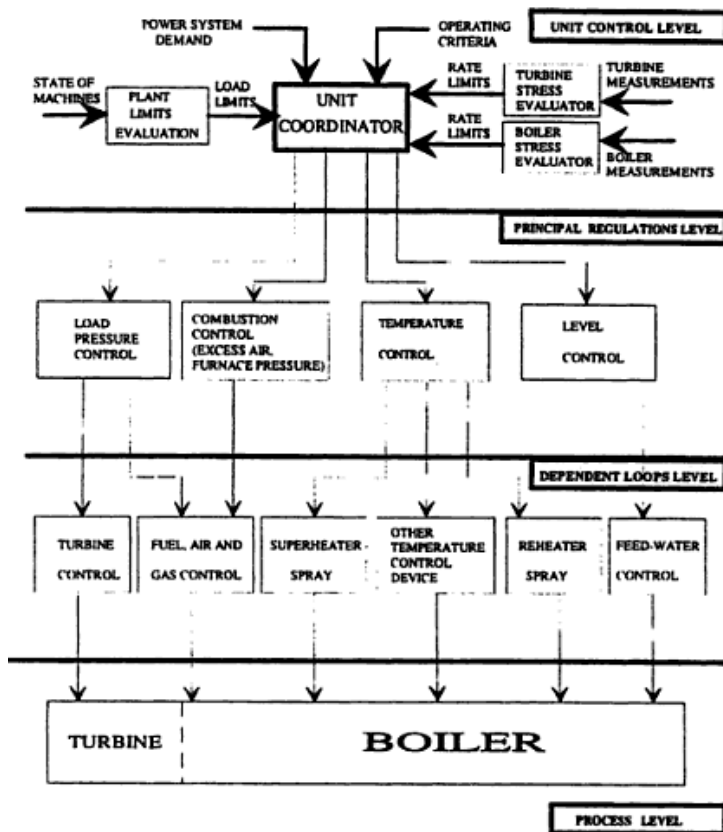


Figure 2.10: A hierarchical overview of a conceptual power plant control. [23]

power can be very fast depending on the chosen control structure and the time constants of the actuators. If it's going to work, it has to match the time constants in the system to supply accumulated steam.

There will be a maximal preferred amplitude of temperature fluctuations during load variations in the turbine to limit material stress due to creep or fatigue. That is, when the steam in to the turbine is throttled the temperature will slightly change; to which direction will depend on if the valve is closed or opened up. Also, a maximal deviation of throttle pressure and drum water level is needed to avoid potential dangerous conditions in the boiler for the largest deviations, e.g. during load rejection at which no steam can be blown off. Also the valve cannot be closed instantaneously in case of steam hammer. One could control the pressure directly in the boiler by feeding it with the signal out of the governor, but since the energy stored in the evaporators and drum are relatively high, and that the pressure is a function of that amount of energy there will be very large undesired time constants.

The high pressure header may fluctuate within about 5% of the set point pressure, avoiding boiler-and turbine trips or damage. In this way there is a very narrow span of which the storage capacities of accumulated steam in the steam drum and pipes can be utilized when there is a fast power ramp up demand during frequency regulation. [23]

The load levelling equipment to work with is the turbine stages, the steam accumulator tank, the feed water tanks, the boilers and the blow off valves. Depending on the operation of Iggesund, there are different ways of controlling the power output. The power can always be ramped down through instant redistribution of steam and throttling. Excess steam will first be filling the steam accumulator and secondly used for pre-water-heating and at last be blown away. Ideas of possible changes of operation can be summarized as:

The plant and steam net system must automatically respond to a frequency deviation in the electrical grid and correct it within the time requirements for FCR. The frequency deviation can easily be translated to an analogous power change of the turbine, or rated circular momentum of the shaft. That can in turn be translated to a related change in accumulated steam flow change through the turbine or a changed pressure ratio over the turbine. But there has to be limitations to the ramp load variations, response time, and participation band of the power system. To account for this, a governor will have the actual frequency in the grid as an input recalculated through the chosen droop and maximum control power participation to a new desired power output. Rearranging the equation 2.2 explained in section 2.2 gives the change of power needed.

$$P_{new} = \frac{\Delta f [\%]}{Droop [\%]} + P_{ref} \quad (2.20)$$

The new desired output from equation 2.20 will form an error together with the actual power output, going through a PI-controller and then fed into the valve controlling the steam flow and pressure into the turbine. This way of controlling the

### 2.6.1 Ramp-Up

The boilers are operating on part load and all the steam goes through the turbine. The turbine's inlet and outlet pressure could then be altered to match the desired power output, and the steam accumulator will be used in power ramp downs.

To ramp down the load, steam is redirected from the gas turbine to the reducing valves, or by taking out more steam in the higher pressure stages, decreasing the power output. To ramp up the power, the delay will firstly be dependent on the time constant of the boiler. That is how fast the oil burns and how fast the feed water is turned into accumulated steam, and secondly how much accumulated steam that can be utilized from the steam drums and pipes without risking turbine or boiler damage or trips. The ramp up speed of the actuator has to be limited so that the pressure after the boiler stays within a 5% margin from the set point value. Some of the accumulated steam in the pipes and steam drum of the boiler will be utilized and the valve will then gradually open as the oil is burned in the boiler. The extra steam will be taken out at 3 bar, getting as much power out as possible where the rest is used for water-pre-heating, district heating or lastly be blown away.

### 2.6.2 Throttle

The boilers operate at a constant load to match the processes steam demand and some steam is reduced past the turbine directly to the steam consumers, and the rest of the steam goes through the turbine. The steam redistribution between the turbine and the amount reduced past it is then altered to match the needed power output. When there is a ramp up of power, the steam accumulator will be used to cover the deficit of steam for the consumers.

This alternative is much more convenient than the "Ramp-Up" one. If one wants to ramp up the power the steam is simply just redirected through the throttling valves and put through the turbine instead, immediately increasing the power output.

### 2.6.3 Overload

The boilers are put to work on overload, supplying a constant overproduction of steam blown away that can be instantly used to increase the power output.

This alternative could be an extension of the "Ramp-Up" alternative, where a sufficient amount of steam will be constantly blown away to cover up for possible power ramp ups, not possible in that specific operating condition.

### 2.6.4 Retrofit

Installing a new steam accumulator on the 110 bar steam net to supply steam during a power ramp up, or extending the condensing stage on the turbine, for a possible 25 MW power increase with increased steam expansion.

With an additional steam accumulator steam will be redirected and both steam accumulators will be charged or drained to cover for ramp ups and downs. The fuel input could then be altered depending on the water level in the accumulators. With a condensing stage there will always be possible to condense 3 bar steam and get a rapid increase in power production. The steam deficit in the 3 bar low pressure net will be compensated by the already installed steam accumulator.

### 2.6.5 Steam-Bleeds

Steam bleeds could be shut in order to instantly increase the power output to some extent. During power ramp ups, it might be convenient to throttle some of the steam bleeds on the turbine for the pre-water-heating and use it to instantly produce power. The time constant of the boilers will increase due to the lower feed water temperature but initial response of the turbine is then much faster. This is something that can be implemented in any operating change.

## 3 Iggesund Mill

In this chapter the specific control and the most important specific equipment of Iggesund mill is explained shortly.

### 3.1 Boilers

In the Iggesund mill there are three different boilers, one recovery, one pure oil boiler and one bark and oil boiler. The recovery boiler works at much higher pressure and temperature, and uses black liquor as primary fuel. During operation of the paper mill, used chemicals are sent to the recovery boiler and recovered in the char bed. The energy from the lignin transferred to the flue gases is recovered in heat exchangers and utilized as steam or electricity in the same way as a normal boiler. The recovery boiler also has the possibility of burning oil.

As a part of Iggesund's new policy the use of fossil fuels should be limited as much as possible. Oil is also in terms of energy per money the more expensive of the two fuels. All the produced black liquor will be burned in the recovery boiler on full load all the time, since the cost of the chemicals recovered is much more expensive than the actual energy extracted. In terms of frequency regulation, input of more oil when the frequency has to be ramped up is still of much interest. Oil also burns much faster than bark, so in terms of response, this is the better option.

The pure oil boiler (P11) is not in use during normal operation. It has got a steam generating capacity of 170 ton/hour (47.22 kg/s) at 485°C and 63 bar. The time from start of oil throttling to full flow is about 10-20 seconds and a full steam flow is achieved after about 109 seconds. The feedwater temperature is 140° C. The oil and bark boiler (P12) burns bark and secondly oil, during normal operation. For this boiler, a full steam flow is achieved after about 72-82 seconds. The steam generation capacity is 150 ton/hour (41.67 kg/s) at 63 bar. This flow can either be achieved from burning both oil and bark, or oil alone. The steam flow capacity from burning bark alone is 51 tons/hour (14.17 kg/s) constituting a bark flow of 22860 kg/h (6.35 kg/s). The remaining 27.5 kg/s would then have to be covered by oil, constituting an oil fuel flow of 7500 kg/h (2.083 kg/s).

The recovery boiler is for now over-dimensioned and ready for further expansion, accomplished by additional heating surface area and a flue gas section. The operating conditions are at present 515°C and 105 bar. The feedwater temperature is 160°C and the steam generating capacity is 383 tons/hour (106.39 kg/s) and 703 t/h = (195.28 kg/s) with expansion.

### 3.2 Turbine

A single casing turbine with centre admission that is suitable for condensing or back-pressure operation. It has got a power capacity of up to 250 MW, an inlet pressure of up to 170 bar and 565°C steam can be achieved. It has two controlled extraction pressures and seven bleeds of various pressure levels as can be seen in figure 3.1. [24]

In the plant the steam is reduced down to 3.5 bar, with an inlet pressure of 105 bar and 515°C steam. There are 6 bleeds which together form 4 extractions with different pressures. There are no reheats between the stages as power production is not its primary purpose.

In the plant it has got a maximum capacity of 75 MW and a future possibility of a 25 MW increase in a condensation stage. The turbine is supplied with a free shaft end to be complemented with a condensation stage. The generator is also designed to withstand both back pressure and condensation. [25]

### 3.3 Gas Boiler

One small gas boiler is installed to burn process gases of a constant load, generating steam at a pressure of around 13 bar.

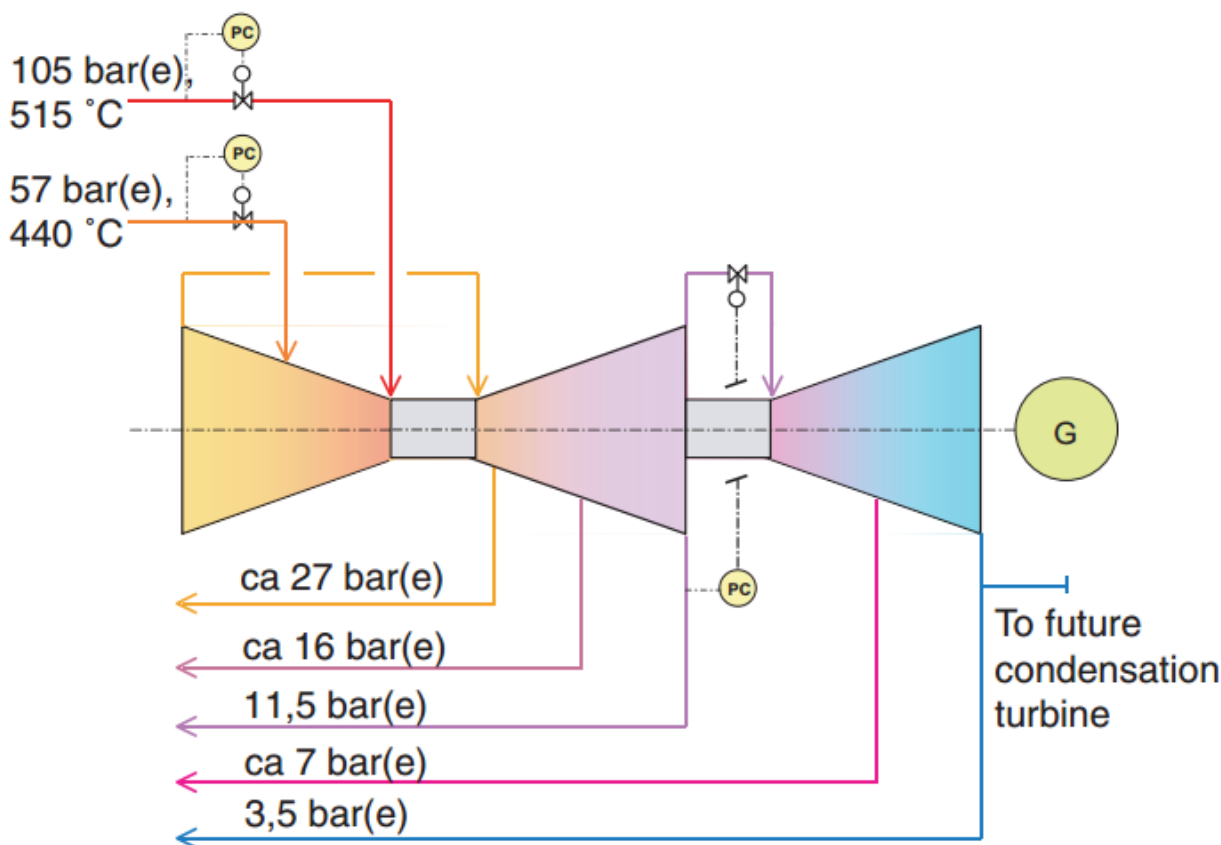


Figure 3.1: The structure of the SST800 turbine. [25]

### 3.4 Steam Accumulator

In the Iggesund mill there is one steam accumulator with a capacity of 22 tons of steam.

### 3.5 Control System

In the actual plant each steam network has a main controller which responds to pressure changes in the steam network. The output signal from the controller is separated into the available control objects (valves or restrictions) by the respective network’s split range function. The size of the control object is adjusted in the split range function so that a similar change of the master regulator’s output signal gives the same change in mass flow to the network regardless of the control object that is concerned. The control objects usually consist of a group of valves which fulfil the same function. As an example, the reducing valves between a higher pressure net to a lower consists of several valves that work as one unit, one big valve. After a signal is sent to a control object of this kind, the signal is divided up on the valves according to the split range function’s priorities. In cases where two steam networks are affected by the same control object, the conflict is handled by allowing one of the networks to control an imaginary control object which set a maximum or minimum limitation of the real control object.

Fundamentally, the control system strives for energy optimization while keeping the net pressures stable. This basically means, maximizing the amount of steam passing through the turbine, and storing the steam left over in the accumulator tank. [26]

## 4 Modeling

In this chapter the models used to build the Iggesund system model will be explained, how they are coupled and the equations behind.

### 4.1 Iggesund Model

Below is a picture of the conceptual model in Dymola that was used for verification.

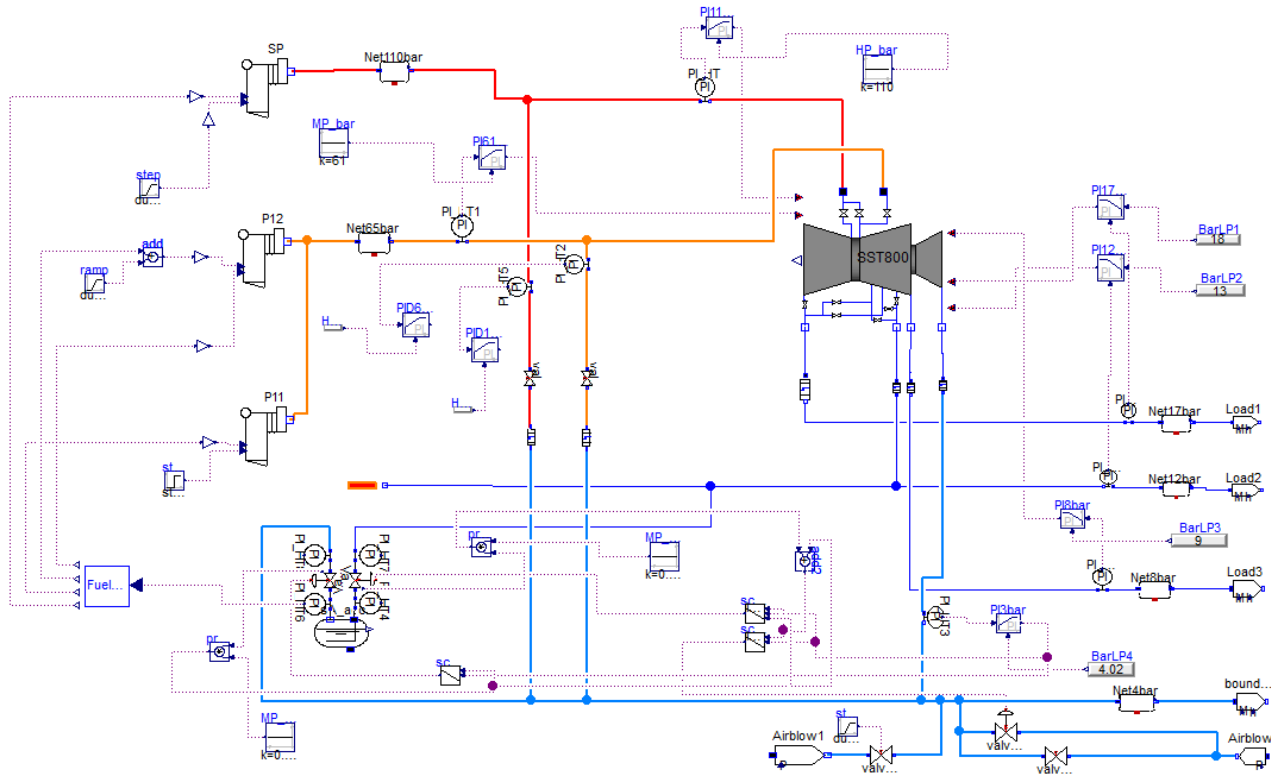


Figure 4.1: An overview of the model used for verification.

### 4.2 Boundary Flows

*Boundary flows* from the *Steam power library* can be used to model processes that the user wants to be set at a fixed value. In these *Boundary flows* different combinations of parameters can be specified such as pressure, enthalpy, temperature and mass flow rate. *Boundary flows* work in a way that if there are no connectors connected, the set point parameters will be used, otherwise the parameters of the in- and output. The pressure and enthalpy is always set according to the connectors. *Boundary flows* will handle both negative and positive inflow and outflow, meaning that if there could be a negative mass flow into the boundary. Flows taken from unconnected connectors are modelled to be infinite.

#### 4.2.1 Pulp and Paper Mill Processes

These processes are assumed to be operating at constant load during nominal paper production. The frequency regulation should be able to be implemented without effecting the paper production, given that this is the main purpose of the plant.

These processes have then been modelled as constant load black boxes to limit the amount of simulation time, using different *boundary flow* (4.2) models from the *steam power library*.

### 4.2.2 Airblow

The accumulated steam in all pressure-stages in the system is coupled to a steam airblow-valve, which releases steam at overpressure.

This valve is modeled as a *boundary flow* (4.2) model from the *steam power library* with a fixed outlet pressure and enthalpy at atmospheric conditions.

## 4.3 Pipes

Pipes are modelled with a *closed tank* from the steam power library of Dymola, described in subsection subsection 4.3.1. This will give the system a realistic dynamic behaviour, with time delay but without pressure drop.

### 4.3.1 Closed Tank

The *closed tank* from the *Steam Power Library* is modelled to have constant volume, filled with a media of fluctuating density. This will imitate the volume and mass of the accumulated steam in the pipes. It is for simplicity modelled so that the pressure in the tank is the same as the outlet pressure.

In the *closed tank* you can define any initial condition, pressure and temperature or enthalpy. A state of the steam corresponding to a specific density, filling up the defined volume of the tank corresponding to a specific accumulated mass of steam.

The initial condition is then calculated through equation 4.6 or 4.2 using the media library mentioned in subsection 2.1.4 depending on, if the initial temperature or enthalpy is set by the user:

$$h(\text{start}) = \text{Modelica.Media.Water.IF97}(\text{P}_{\text{start}}, \text{T}_{\text{start}}) \quad (4.1)$$

else:

$$\text{T}(\text{start}) = \text{Modelica.Media.Water.IF97}(\text{P}_{\text{start}}, h_{\text{start}}) \quad (4.2)$$

The dynamic behaviour of the tank can be expressed by the following basic equations:

$$\text{Volyme} = \text{constant} \Rightarrow \dot{V}_{in} = \dot{V}_{out} \quad (4.3)$$

$$m = \rho(P, h) \cdot V \Rightarrow \frac{dm}{dt} = \frac{d(V \cdot \rho(P, h))}{dt} \Rightarrow \dot{m} = V \cdot \dot{\rho}(P, h) \quad (4.4)$$

$$\frac{dU}{dt} = \dot{h}_{in} + \dot{h}_{out} + \text{Heatport} \cdot \dot{Q}_{in} \quad (4.5)$$

$$\frac{dm}{dt} = \dot{m}_{in} + \dot{m}_{out} \quad (4.6)$$

If the massflow in is lower than out according to equation 4.6, temperature or pressure will decrease, and so will the density according to equation 4.4.

If the *closed tank* is at steady state and connected to a steam turbine and a higher power output from the turbine is wanted, the need for accumulated steam massflow is increased, and will be drained from the steam



pipes connected to the turbine. There will then be a negative mass balance over the tank in accordance to equation 4.4 until the needed massflow is met by either steam from a steam drum in a boiler or by steam from a steam accumulator tank. All of these tanks will later be refilled by the increased steam generation from the boiler that in contrast has a modelled transient behaviour. For how long the increased power will be met will be a function of the time constant of the boiler and the total accumulated steam in the system.

The pressure of the steam will as a consequence of the negative mass balance decrease in the tank, this is the only way to decrease the density in the tank since the temperature is constant and fixed from the boiler. If the tank becomes empty, the density becomes infinitely low and the simulation will stop, or when the pressure and/or density is so low that it violates any possible state of the steam. The opposite is of course true for a reduced power of the turbine.

## 4.4 Valves

*SteamValves* from the *Steam Power Library* have been used to model the valves. These are especially designed for water steam flows and have a little more complex structure of equations than described in the theory chapter in subsection 2.4.6. The massflow through the valves can be described using equation 4.7:

$$\dot{m} = C_v \cdot N_6 \cdot \text{openpart} \cdot Fp \cdot Y \cdot \sqrt{x \cdot P_1 \cdot \rho} \quad (4.7)$$

In equation 4.7  $N_6$  corresponds to  $k$ , a conversion factor to give the massflow in kilogram per second. *Openpart* corresponds to  $\alpha$ , the position of the valve. *Fp* is a piping geometry factor accounting for pressure drops due to the pipe fitting.  $Y$  is the expansion factor as can be seen in equation 4.8,  $P_1$  is the inlet pressure and  $x$  is another expansion factor seen in equation 4.10.

$$Y = 1 - \frac{x}{3 \cdot Fk \cdot xT} \quad (4.8)$$

$$Fp = \frac{\kappa}{1.4} \quad (4.9)$$

$$x = \frac{P_1 - P_2}{P_1} \quad (4.10)$$

There are several ways to describe the  $C_v$ -value of a valve, to put it shortly, it is the difference between the conversion factor and accuracy of each individual parameter. When  $C_v$  is calculated manually, nominal measures are used, described in equation 4.11.

$$C_v = \frac{\dot{m}}{N_6 \cdot Fp \cdot Y \cdot \sqrt{x_{nominal} \cdot P_{1nominal} \cdot \rho_{nominal}}} \quad (4.11)$$

Due to the fact that the steam flow is compressible, the density will not be constant. When the velocity of the steam goes above the speed of sound, choked flow will occur. When this occurs, the massflow over the valve cannot further increase regardless of how much the pressure drop increase. This is also considered in the model and equation 4.7 will be modified when this occurs. The dynamic behaviour of the valve will then depend on the  $C_v$ -value, the valve characteristics in equation 4.11.

The valves are modelled so that the steam expansion process is adiabatic, the enthalpy and enthalpy flow remains constant whereas the exact temperature of the new state is calculated to match the new desired pressure, without any losses. The initial conditions of the steam are calculated the same way as the *closed tanks* in subsection 4.3.1 using the medium library and the initial temperature or enthalpy specified by the user.

### 4.4.1 Actuators

To the specific valve modelled there is a possibility to connect an actuator from the *Steam Power Library* described in subsection 2.1.5 to control the position of the valve ( $\alpha$ ). The actuator will receive a signal between zero and one from the controller, where one is a fully opened valve. The actuator will introduce a realistic dynamic behaviour of an actual actuator, setting the time to shut and open the valve. That is, the response time of the actuator.

## 4.5 Steam Saturators

The process is assumed to have zero pressure drops and to be adiabatic. This can be summarized into equation 4.12 and 4.13:

$$h_{out} = \text{Modelica.Media.Water.IF97}(P_{out}, T_{set}) \quad (4.12)$$

The enthalpy of the steam coming out is calculated using the Modelica media library (2.1.4) with the desired outgoing temperature and outlet pressure, depending on the processes followed by the saturator.

$$\dot{m}_{water,in} = \dot{m}_{steam,in} \cdot \left( h_{steam,out} - \left( \frac{\frac{P_{steam,in}}{\dot{m}_{steam,in}}}{h_{water,in} - h_{steam,out}} \right) \right) \quad (4.13)$$

Where:

- P = enthalpy flow [kW]
- $\dot{m}$  = massflow rate [kg/s]
- h = enthalpy [kJ/kg]

The water flow in equation 4.13 is calculated so that the exact desired temperature is reached. The outgoing steam mixture will have a higher enthalpy flow than the steam flow into the saturator according to equation 4.15, with the added enthalpy flow of the water in equation 4.14.

$$P_{water,in} = h_{water} \cdot \dot{m}_{water} \quad (4.14)$$

Where  $h_{water}$  is set by default to 400 [J/kg].

$$P_{steam,out} = P_{steam,in} + P_{water,out} = h_{steam,out} \cdot \dot{m}_{steam,out} \quad (4.15)$$

The water will by default be drained from an infinite water reservoir.

## 4.6 Steam Accumulator

The steam accumulator module is based upon a *Separator tank* which is identical to a *closed tank* imitating the dynamic behaviour of the pipes, described in section 4.3.1. The *Separator tank* is limited to only work in the 2-phase region, through a true or false statement.

Apart from initial conditions such as pressure, enthalpy or temperature, height and volume of the tank can be fixed. It is important to specify a realistic initial condition of the medium inside the *Separator tank* somewhere

in the 2-phase region. Depending on the outlet and inlet pressure to the *Separator tank* there will be a natural flow of media out or into the *Separator tank*.

The fraction of water and steam is calculated using equation 4.16.

$$V_{mix} \cdot \rho_{mix} = V_{liquid} \cdot \rho_{liquid} + (V_{mix} - V_{liquid}) \cdot \rho_{vapour} \quad (4.16)$$

Where:

$V$  = Volume [m<sup>3</sup>]

$\rho$  = Density [kg/m<sup>3</sup>]

$h_{liquid} = Modelica.Media.Water.IF97.h_l(p)$

$h_{vapour} = Modelica.Media.Water.IF97.h_v(p)$

$d_{liquid} = Modelica.Media.Water.IF97.\rho_l(p)$

$d_{vapour} = Modelica.Media.Water.IF97.\rho_v(p)$

The thermodynamic state of the 2-phase medium is used together with the media library described in subsection (2.1.4) to get the desired properties needed in equation 4.16.

The steam accumulator in the Iggesund mill described in chapter 3 has been modelled with a volume of 400 m<sup>3</sup> and a maximum water content of 56%, corresponding to 22 tons of steam.

## 4.7 Steam Turbine

### 4.7.1 SST800

A model of the actual SST-800 steam turbine in the plant described in chapter 3 is created using the *Steam Power Library* described in subsection 2.1.5.

The turbine has been modelled using 6 single *Stadola turbine stages* described in this section, *Closed tanks* and *ValveSteam* from the *Steam Power Library* described in subsection 4.3.1 and 4.4 respectively. The function of the turbines are based upon Stadola's turbine equation described in subsection 2.4.1, which relate the steam massflow rate through the turbine to the steam conditions at the turbine inlet and outlet.

To set Stadola's constant for each turbine, the system has to run with an arbitrarily value of the Stadola's constant and then be trimmed to match the full load working conditions of the actual plant. Part load calculations will then only be a good estimate of the actual conditions due to simplifications made in Stadola's model. Since there are 6 bleeds summing up to 4 different pressures, the outlet pressure of each turbine does not have to be absolute, and could be altered with Stadola's equation to give the optimal working condition.

In the actual model you are free to specify Stadola's constant as just mentioned as well as the efficiency, mentioned in the theory chapter, subsection 2.4.1. In addition, start values can be specified which should be accurate in order for the iteration to be true.

The model will autocorrect the pressure to be a minimum of 650 [Pa], the enthalpy to be maximum of 8000 [kJ/kg] and minimize the pressure drop to 0.6%.

$$W_{turbine} = \dot{m} \cdot \Delta h \quad (4.17)$$

Equation 4.17 will calculate the power output based upon the massflow rate through the turbine and the enthalpy drop over it.

$$S_{in} = Modelica.Media.Water.IF97(h_{in}, P_{in}) \quad (4.18)$$

$$h_{is} = Modelica.Media.Water.IF97(S_{in}, P_{out}) \quad (4.19)$$

The Modelica media library (2.1.4) is used in equation 4.18 with the pressure and enthalpy of the incoming steam to get its specific entropy. The same goes for equation 4.19, instead with the entropy and outgoing pressure as variables to get the enthalpy of the outgoing steam, assuming isentropic expansion.

$$\Delta h = (h(P_{in}, T_{in}) - h_{is}(P_{out}, S_{in})) \cdot \eta_{is} \quad (4.20)$$

Through equation 4.20 the enthalpy drop is calculated as the difference between the enthalpy before and after the stage, with isentropic expansion, multiplied by the isentropic efficiency.

$$h_{in} + h_{out} - W_{out} = 0 \quad (4.21)$$

$$\dot{m}_{in} + \dot{m}_{out} = 0 \quad (4.22)$$

Energy and mass balances are calculated through equation 4.21 and 4.22.

To operate the turbine at part load the pressure before and/or after the stages has to be altered which will change the massflow and enthalpy drop over the turbine. According to Stadola's equation (2.5) the massflow will have a non-linear relationship to the inlet pressure and the pressure ratio. If the inlet pressure is high, the outlet pressure will not have much influence on the massflow rate, and the opposite is true for a low inlet pressure. The combination of inlet and outlet pressure can have an exponential effect on the power output.

To decrease the power by lowering the massflow and keeping the enthalpy fixed, the steam state is just shifted to the right in a Mollier chart. Thereby decreasing both inlet and outlet pressure, keeping the same pressure ratio but decreasing the massflow rate only dependent on the change of inlet pressure, according to Stadola's equation 4.20. The relationship between massflow rate and power output will then basically be linear.

Keeping a constant pressure ratio/drop, and increasing the inlet pressure will give a linear relationship to the power output. A change of the outlet or inlet pressure, will have an exponential effect on the power output. In this thesis, the power output of the turbine will be controlled by controlling the inlet pressure mentioned in section 2.6, since the outlet pressure of the external four turbine stages has to be constant in order for the processes to function properly. The massflow rate of the six internal turbine stages may be changed in order to match different operating conditions.

## 4.8 Boilers

Although the boilers are somewhat different in the Iggesund Mill described in chapter 3 the same model is used for all the three units found in the *Steam Power Library* described in subsection 2.1.5.

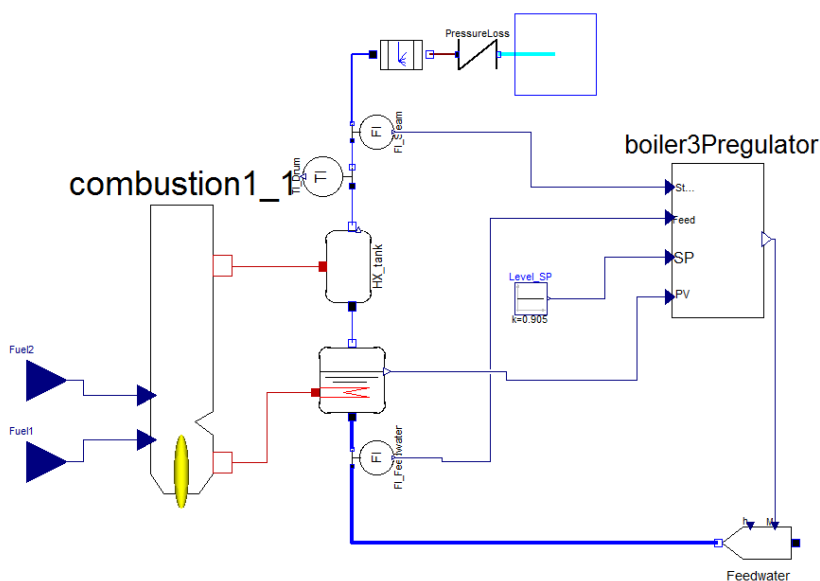


Figure 4.2: The structure of a the boiler model used from the steam power library.

drum. This will only work with one fuel fed.

The heatflows out of the combustor then enter the superheater and drum. The superheater is modelled as a *closed tank* described in subsection 4.3.1, and the drum as a *steam accumulator* described in section 4.6. Now both tanks use the heatflow input to model the heat transfer in the tubes in the boiler. That is the heat transfer from the combustion process to the feedwater. The final superheated steam coming out of the superheater is cooled down by a *steam condenser* described in section 4.5 to the desired temperature, the design condition for the specific boiler, set as a fix parameter.

The feedwater flow into the drum is controlled by a controller unit from the *Steam Power Library* called *Boiler3Pregulator*. This controller unit has the feedwater flow into the drum, the steamflow out of the superheater and the level of liquid water in the boiler as inputs. The output is then the desired feedwater flow modelled as water coming from an infinite reservoir, using a *Boundary flow model* described in subsection 4.2.

In the actual model, the level of the steam drum will not be considered more than that the simulation will stop if the water level is too high or too low. The temperature control in the boiler will be fixed and corrected by the use of a steam saturator described in section 4.12. The combustion control will not be considered, referring to the control of excess of oxygen or furnace pressure. It is only modelled through fuel input and perfect conversion combined with an induced time delay relative to actual measures of the specific boilers, mentioned above.

Normally the pressure level of a boiler is controlled through MCR where the pressure bias will be set on different levels depending on the load level of the boiler. This way the efficiency will be better considering that less

Figure 4.2 shows the structure and the components the boiler is made of.

Figure 4.3 illustrates the combustion chamber alone. Both fuels are fed into a gain function respectively representing the heating value of the fuel. Each signal is then divided and fed into two different lag filters, one for the super heater section and one for the drum section. Neither of the other sub models in the boiler model has any time delay, that is, no transient behaviour. The lag filters will then represent the time it takes for the energy content of the specific fuel to be transferred to the feedwater and come out as superheated steam at a desired state.

After the lag filters, the signals for both fuels are added up to get a final heatflow for the superheater and the

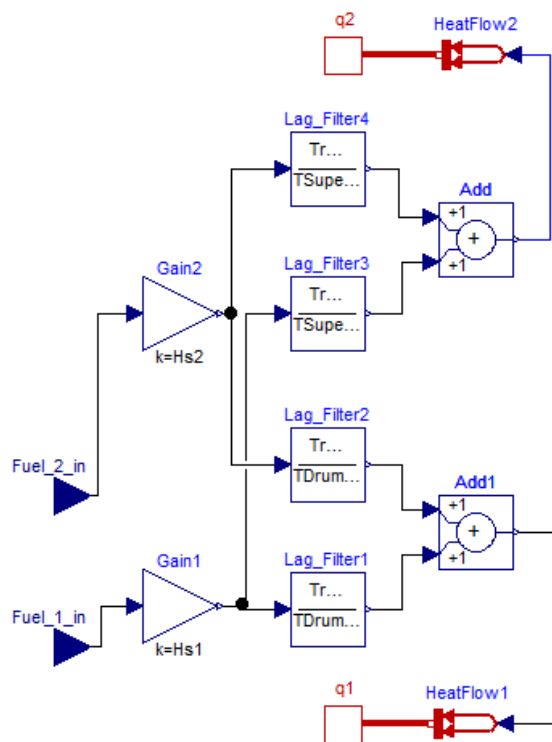


Figure 4.3: The internal structure of the combustion chamber.

throttling will be needed in the turbine. However, in the model the pressure of the boiler is controlled so that a constant pressure is held independently of the actual load.

## 4.9 Gas Boiler

The gas boiler is modelled with a *boundary flow* and *SteamValve* from the *Steam Power Library* described in subsection 4.2 and 4.4 respectively. The *boundary flow* model has got a fixed output pressure of 20bar and an enthalpy of 2800 kJ/kg, a state very close to saturated steam. The *SteamValve* is set to have a constant  $C_V$ -value and a constant valve opening ( $\alpha$ ) giving a constant slightly superheated steam massflow.

## 4.10 Control

As mentioned in the theory chapter, section 2.5. The control of the plant is mainly done through different sets of PI – controllers which are easy to tune with the Lambda tuning method. Each net except the low pressure net is controlled with one PI-controller that controls one single valve with the actual pressure in the net as a measurement and the wanted as set point. By throttling the correct pressure is then being held.

The figure

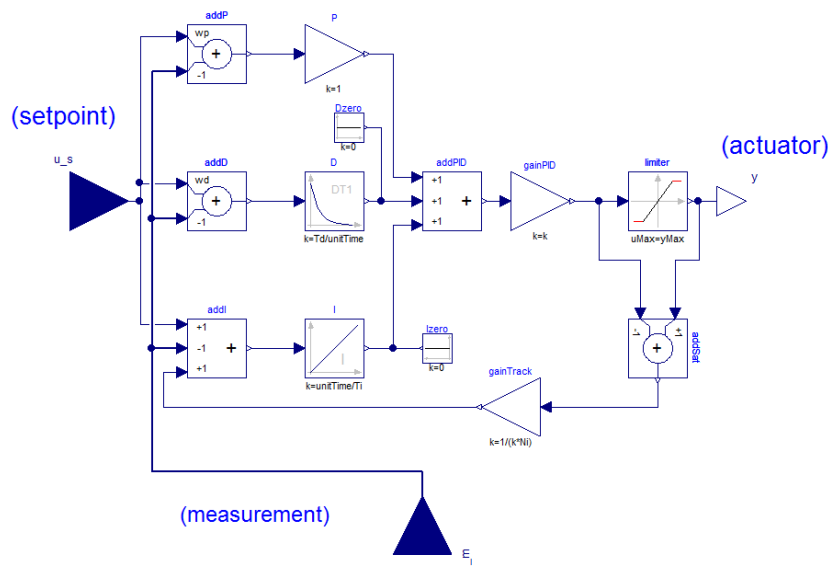


Figure 4.4: The structure of a *limPID*-controller in the original *Dymola* package.

### 4.10.1 Tuning

To improve operation, each controller has to be tuned to the specific operating condition.

To do this, the process was run and a step function was performed to see how long it takes for the reading to implement its change. That would be the time constant ( $T$ ). How long the deadtime ( $L$ ) is, was also noted. The processing gain ( $K$ ) was measured according to equation 2.18.

For a valve, that would be:

$$K = \frac{\Delta P}{\Delta \alpha} \tag{4.23}$$

Here it is important to understand that the valve authority, which is the relationship between the change in pressure and valve opening, is assumed to be linear.

The low pressure net has the lowest priority, meaning that it is only this net (under normal operating conditions) that will have an excess or a deficit of steam. This net will control the steam accumulator and the boiler throttle as in the real plant, although very simplistic.

The steam accumulator is controlled so that when there is overpressure on the 3 bar net, it first closes the outlet to the steam accumulator and then the inlet is gradually opened to a maximum and finally the blow-off valve is opened. The reverse occurs at underpressure.

This is realized through a mini-split range function (2.5.4) that is built upon three scale functions. The scale of the functions is such that they compensate for that the valves at the inlet and outlet of the accumulator tank and free blowing valve are of different sizes. If a valve is double in size compared to two other, the opening should be between 0-50% of the controller output for the big valve and between 0-25% for the other two.

Since the pressure in the accumulator decrease or increase as it is being filled or drained of steam, the steam flow through the valves will change thereafter. To compensate for this, the scale functions will change their range of scale according to coefficients in the valve massflow equation 4.7:

$$\text{Compensation} = Fp \cdot Y \cdot \sqrt{x \cdot P_1 \cdot \rho} \quad (4.24)$$

In this way the massflow through the accumulator will increase or decrease linearly independent of the pressure in the tank.

Worth to mention is that the pressure on the low pressure net will oscillate around 4.02 bars which is the setpoint, meaning that the power output will also follow these oscillations. The decrease or increase of enthalpy and exergy in the steam coming out of the turbine on the low pressure net is directly proportional to the power output. It is therefore extra important that the controller on the lower pressure net is finely tuned for the specific operating case.

### 4.10.2 Accumulator

In the accumulator the pressure can vary between the pressure on the nets at the outlet and inlet, which are 4 and 13 bar, respectively. The initial pressure state of the accumulator is 8 bar and that is also the set point of the accumulator. As mentioned before, the black liquor in the recovery boiler is always run on full load and if the pressure goes below 10 bars the bark input on the P12 boiler is slowly maximized to keep the pressure at 10 bars. At normal working condition the bark input is on full load. When the pressure goes below 8bars the oil fuel input on the P12 boiler starts to increase. This is the fuel input on the margin and has a lower time constant than both black liquor and bark. If the pressure somehow would decrease further below 7- and 6 bar, the oil input on the P11 boiler would start to increase and lastly the oil input on the recovery boiler. This is illustrated in the figure 4.5.

### 4.10.3 Frequency Control

In the real model, in terms of steam net control, the accumulator tank is used as an indicator for whether there is an excess or deficit of steam in the system mentioned in section 3.5. If the pressure in the tank is higher than the initial state, the fuel inputs to the boilers are decreased and if the pressure is higher they are increased. In this way the accumulator balances disturbances in the system, no steam is wasted and the pressure in the nets remain constant. But since the accumulator only releases steam to the lower pressure net, it is always the low pressure net that either has got a deficit or surplus of steam and pressure, depending on control structure on the low pressure net.

As mentioned in section 2.6 there are different ways of controlling the frequency through process changes. It can be concluded that the specific boilers at Iggesund mill are designed with relatively small steam drums. The pressure and the temperatures are high meaning that special steel and thick walls are required. Along with safety measure these are very expensive. To install bigger steam drums or an additional steam accumulator on the high pressure net is therefore not feasible, neither is the option to supply an over production of steam. The option to retrofit and put the boilers on overload is therefore not looked at.

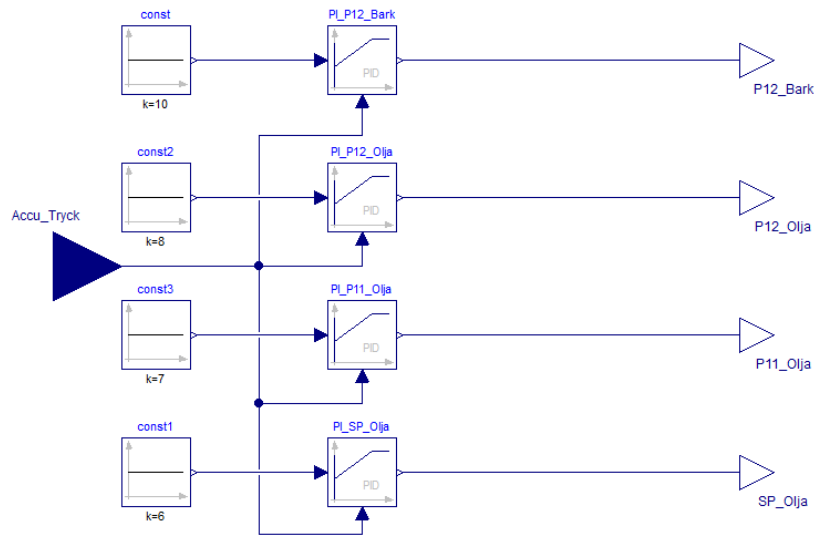


Figure 4.5: The control of the fuel inputs through the accumulator modelled in Dymola.

The concept to shut steam bleeds for FWH and instead expand it in the turbine is not possible since there are no bleeds for FWH. LP-steam will be and is used for this as long as there is no condensing stage.

The operating conditions of Iggesund, may vary over the year. Depending on the price for electricity it can be feasible to throttle the steam directly from the HP net down to the MP- and LP-net and the steam consumers. Going past the turbine since the value of accumulated steam then is of higher value than that of electricity. The paper production is planned to increase in the near future, increasing the demand of accumulated steam to the processes, giving less space for ramp up in the boilers. [1]

P11, the complete oil boiler mentioned in chapter 3 is seldom used at all since Iggesund's policy is to go towards a complete renewable operation, meaning that the oil is to be phased out. Also there is a relatively small oil-input to the bark and oil boiler, P12. Nevertheless, the possibility to ramp up the oil fuel input can be desirable in case of frequency regulation, along with the option to throttle steam past the turbine.

Depending on the operation of Iggesund, there are then two preferable ways of controlling the power. In both cases the following control has been implemented:

A PI-controller that has the power from the turbine as a measurement and the specified power as a set point (which will be dependent on the frequency). If the power is too low then the PI-controller will measure a fault and send out a higher signal between 0 and 1, which in turn indicates the degree of opening of the valve to the turbine on the 61-bar network. When the turbine valve opens up, the pressure on the net will decrease if the new steam demand is not met by the specific boilers. Another controller has then been added to the 61-bar net with a pressure set point 5% lower than that of the actual set point (61 bar). The outputs of both of these controllers are then fed into a min-function, only letting the smallest number pass. In this way the accumulated steam in the drums and pipes will be utilized maximally and the power will increase as fast as possible, without letting the pressure decrease more than 5% in line with the recommendations mentioned in chapter 2.6, yet avoiding boiler- and turbine trips or damage. If the pressure on the other hand would go above 61.5 bar a throttling valve will open to release steam to the lower pressure nets.

#### 4.10.4 Ramp-Up

In case of ramp up (mentioned in the theory chapter 2.6.1) the control system is designed with an additional, very aggressive controller, which measures the pressure on the net, with 61bar as set point. When the pressure decreases in the net, in terms of frequency control, the controller will get an error and almost immediately maximize the oil input to the P12 bark- and oil boiler. A working structure with the P11 oil boiler being used has also been modelled. The initial fuel control via the accumulator is then not controlled.



To illustrate an example of how this works, let's imagine a sudden frequency drop corresponding to a specific power increase. The set point of the controller connected to the turbine will increase, the controller will then have an error sending out a signal to open up the turbine inlet valve. A ramp function is built into the controller limiting the output signal to reach its final needed value and opening of the valve, over the course of 30 seconds. This is the maximum time to reach full power response according to ENTSO-E requirements mentioned in subsection 2.2.1. At first, to cover up for the extra steam needed, the steam drums and pipes will be drained and the pressure in the net will decrease. When the pressure in the net decrease the other controllers, controlling the fuel input will maximize the oil input to one or more oil boilers. As more fuel burns in the boilers, more accumulated steam goes to the turbine and less steam has to be drained from the drums and pipes. After a certain time the extra steam needed will be met by the fuel input alone and the drums and pipes will not be drained anymore. Instead the pressure will increase back to the default of 61 bar as the steam production increase further and steam is accumulated in the tanks and pipes. When the set point of 61 bars is reached the oil input will be lowered and any steam above 61.5 bars will be throttled to the nets below and stored in the accumulator until the system has stabilized. Se figure 4.6 for the complete system:

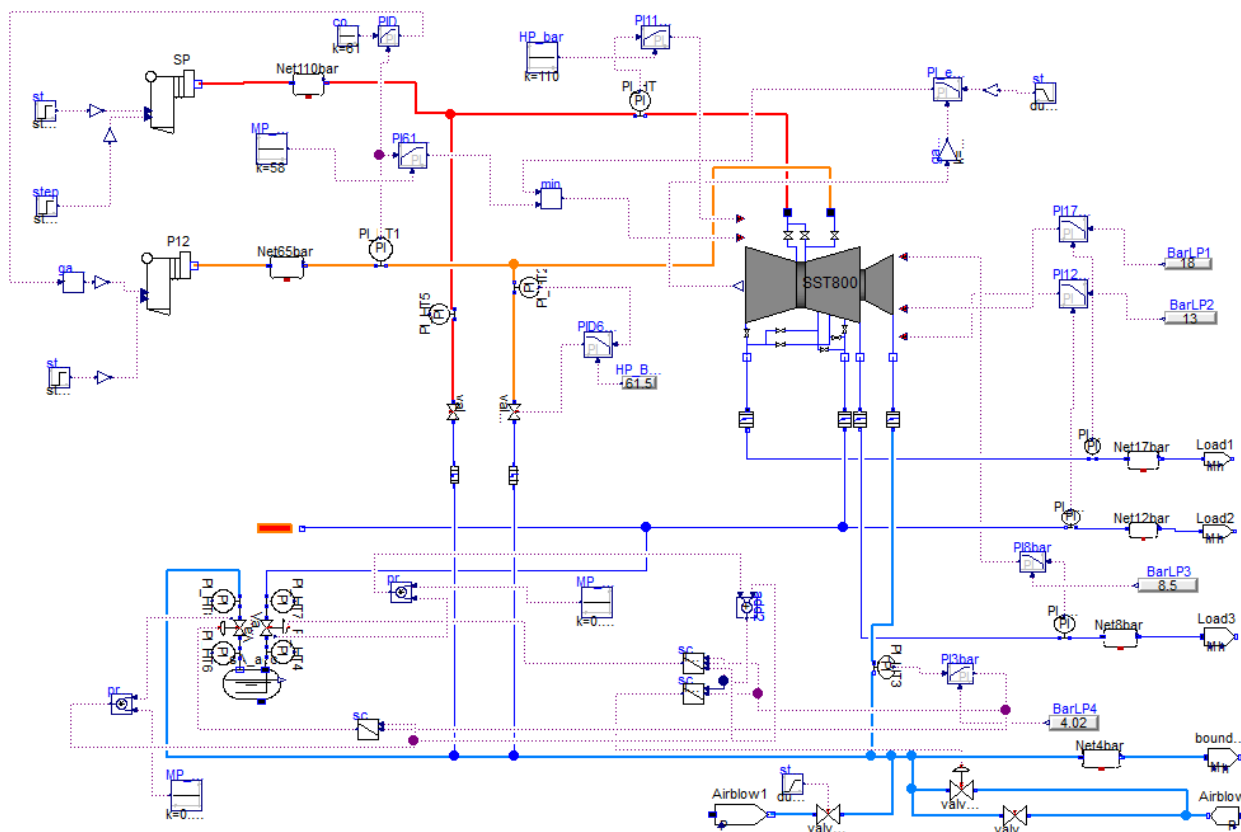


Figure 4.6: The total system modelled in Dymola in case of "Ramp Up".

### 4.10.5 Throttle

If the plant is to run with throttling mode (mentioned in the theory chapter 2.6.2), the same control structure can be used as in the actual plant. The exact same amount of steam will then be throttled past the turbine corresponding to the amount of power it would produce going through the turbine, which you want to participate for. But when throttling steam past the turbine, steam saturators (mentioned in section 4.5) are needed to cool the steam down to the desired temperature in the low pressure net by injecting saturated water at that pressure. This will increase the steam flow by approximately 25%. When the steam goes through the turbine no cooling is needed at all. The increased steam flow will be stored in the accumulator and the fuel inputs will be lowered until the exact amount of steam is given to the steam consumers. This means that about 80% of the steam that you initially throttle past the turbine will be the throttled and available for ramp up as the system stabilizes. With this kind of control, the steam will be utilized to its maximum, but the accumulator fuel control must not be affected by any form of disturbances.

The accumulator fuel control structure could be taken out of control and let the excess steam fill the accumulator and thereafter be blown away. This is not very good for energy optimization, though. See figure 4.7 for the complete system:

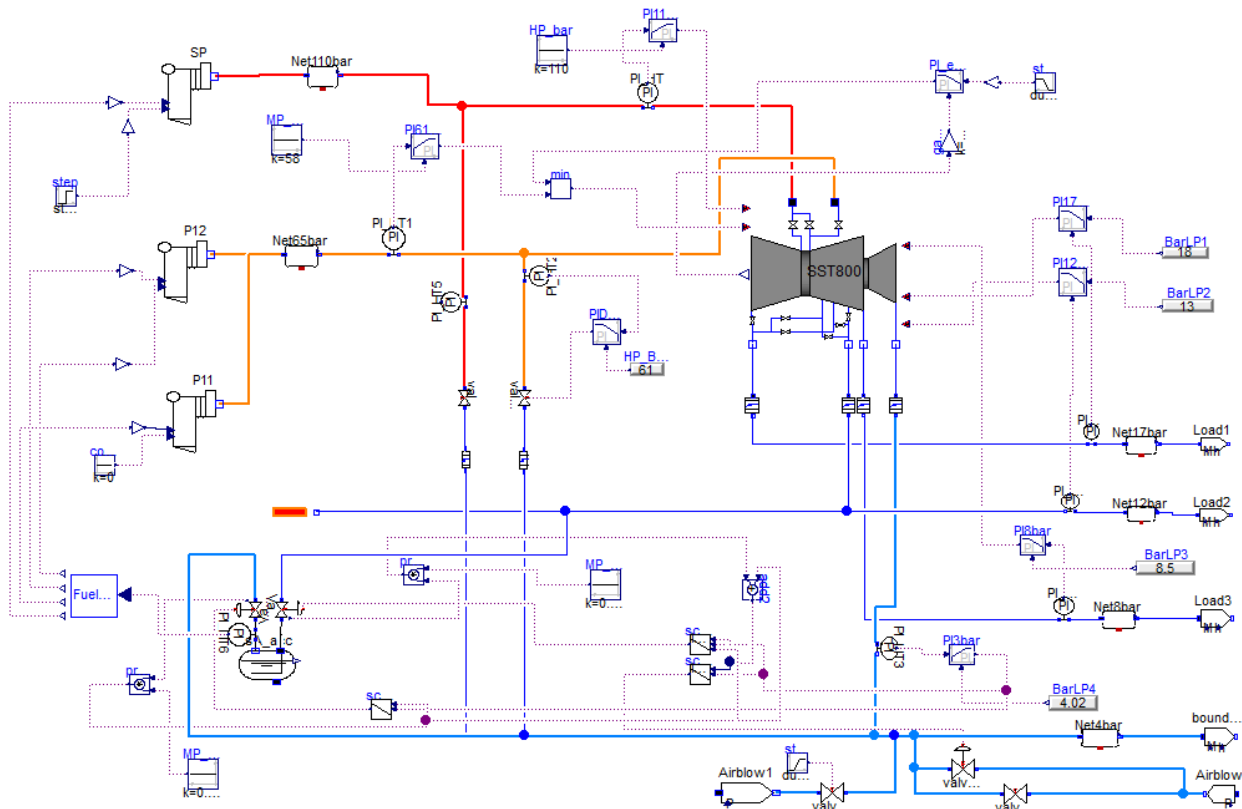


Figure 4.7: The total system modelled in Dymola in case of "Throttling".

## 4.11 Verification

The overall model used for verification is modelled for a normal operating condition. With the exact same massflows from the recovery- and bark boiler, and the same pressures and mass flows to each net in the system as in the real case. This is realized by a constant burning of bark and black liquor. In case of maximal oil ramp ups, a minimum of  $0.5 \text{ m}^3/\text{s}$  oil has to be burned in the oil boiler to keep it operating, decreasing the amount of bark burned by a corresponding amount of energy.

A simulation was done with the extensive model and with the simplified. At a specific time, long enough for pressures, power and so on to have evened out for both models, a ramp up of oil was done in the P12 oil boiler in both models. The most important dynamic responses were observed and plotted against each other for both models.

The response of the accumulator was a little bit slower in the simplified model than the extensive one. Nothing that will affect the frequency control itself, the pressure in the accumulator will just take more time to even out.

## 5 Results

In this chapter the results from the project are presented. That includes the simulation and optimization of the two most favourable concepts for frequency regulation of Iggesund mill, mentioned in sections 2.6 and 4.10.3 and some important plots from the steamdrums and pipes, found in the appendix.

Unless otherwise stated, all plots and simulations are based on an average working condition of 355000 tons of paper per day, at steady state. In all cases, full response has been modelled to be at 30 seconds which is the primary requirement for participation mentioned in 2.2.1. This would then indicate the maximum range of participation for the system during different operating conditions. All figures and plots with explanations concerning the results can be found in the appendix. When referring to a plot in a specific figure, plots are numbered in chronological order, from top to bottom.

In all cases the opening time for each valve used is accounted for in the system but since that time is in the range from 3-5 seconds and that the actual rate of opening is only up to a few percentage of the maximum value in the ramp ups, this time delay is negligible.

### 5.1 Steamdrums and Pipes

Regardless of the concept used for PFR it is interesting to study the steamdrums and pipes, to see their individual potential for PFR.

When a lower pressure is accepted, accumulated steam in the pipes and steam drums can be utilized to further increase the response mentioned in section 2.6. In this way, to lower the pressure in the tank and the steam drums there has to be a density gradient, decreasing the density in the tanks and thus decreasing the accumulated mass of steam in the volume of the system. How fast this is released, the mass flow rate out of the tank will be inversely proportional to the pressure difference between the inside and outside of the tank. If one wants to have a more linear mass flow response, the pressure drop can be altered by controlling the valve opening to the turbine constituting a constant power increase.

When the pressure is altered to drop in one of the drums by 3 bar over 30 seconds, the first few seconds, a little bit more steam leaves the drum than feed water comes in. With a negative mass balance flow out of 5 kg/s at the most, the system balances out quickly. The water level in the drum drops slightly. To summarize, the mass in the drum is decreased initially as more steam-mass leaves the drum than feed water comes in as the pressure decreases. The temperature in the tank decreases just slightly. The decreased temperature and pressure gives a state of lower enthalpy.

Through simulation it can be seen that a double tank volume would double the power over time at the same pressure drop. To double the pressure drop gives almost the same effect. In figure 7.2 the pressure setpoint in the net was instantaneously decreased by 3 bar, hence the power and massflow peak at about 6 seconds. However, the area under the graph in the first plot shows the amount of energy from the pipes and tanks at various volumes that with losses can be converted into electrical power.

The volume in the pipes between the P12 boiler and turbine is assumed to be  $10 \text{ m}^3$ . If the P11 boiler is to be included, a volume of  $22 \text{ m}^3$  is assumed. As can be seen in figure 7.2 the size of the tanks simulating the volume in the pipes does not make much of a difference compared to that of the size of the drums. When the pressure in the pipes is decreased by 3 bar the density of the steam is only decreased from  $19.2 \text{ kg/m}^3$  to  $18.2 \text{ kg/m}^3$ , about  $1 \text{ kg/m}^3$ . This means that pipes with a volume of  $10 \text{ m}^3$  can, as an example, release 10 kg of accumulated steam over time.

In figure 7.1 it can be seen that when the pressure set point in the net is decreased by 3 bar over about 16 seconds, the power contribution from the tanks and pipes will then peak at this very moment, and then start decreasing. Combined with the power contribution from burning oil, that decrease will be precisely met. A power increase of about 3.5 MW could then be met within 16 seconds and about 4 MW within about 30seconds, from the P12 boiler alone. If the same pressure drop in the net would be delayed by 14 seconds a maximal power increase of about 5.5 MW can be achieved within 30 seconds. There is also a curve where the pressure in the net was decreased over 30 seconds with a combined power increase of about 4.5 MW over 30 seconds.

## 5.2 Throttle

When throttling a specific amount of steam past the turbine, and use that to increase the power when needed, the power increase can be done “instantaneously” or with a fixed time delay, limiting the opening time of the inlet valve to the turbine. There is no other way to get a faster response than this, using a steam turbine to generate the mechanical work to the shaft, relating to a specific frequency in the system.

With this concept there is a loss of about 125% power generation of the amount participated for. However, fuel is saved for the lesser steam production. It is feasible if the compensation for participation per MW power combined with the compensation for extra sold or not sold electricity is higher than that of sold electricity missed out on.

In figure 7.6 the accumulator control structure is in operation. In this case the power is ramped up from 35 MW to 47 MW. As mentioned in subsection 4.10.3 when reserving steam for power ramp up with the accumulator control structure in use, the amount throttled will decrease as the fuel input decrease to match the process demand. When the ramp up then occurs, there will be a sudden deficit of steam to the consumers that the accumulator has to match, as can be seen in the second lowest plot. The pressure in the accumulator then decrease. As the pressure in the accumulator decreases the fuel inputs to the boilers are increased in line with the accumulator control structure mentioned in subsection 4.10.2.

In figure 7.7 it is seen that the steam saturator outflow after the turbine is basically the same as the inflow meaning that almost no water is injected. In the case of throttling there is a clear difference of how much steam there is after the steam condenser and before.

## 5.3 Ramp Up

A maximal ramp up of oil was done in both the P11 and P12 boilers to distinguish the maximal power response potential on the 65bar net. Black liquor cannot be ramped up since this is always burned on full load. The bark is too slow, and will cause more instability to the system than response. When P11 and P12 both are ramped up max at once, the turbine inlet valve on the 61-bar net will eventually maximize and the pressure will thereafter increase and the mass flow through the valve as a consequence of that. This means that you cannot fully utilize P12 and P11 for a fast ramp up. The power increase over time will not be as good as it could have been with a bigger valve. This is illustrated in figure 7.8.

In figure 7.8 the red line indicates the power increase from the turbine when both the P12 and P11 oil fuel inputs are maximized, at the same time as the pressure are set to decrease linearly by 3 bar over the course of 30 seconds with the use of a ramp function on the set point value to the PI-controller of that net. The valve into the turbine is enlarged, never to be fully opened. The maximal response is then as can be seen in the plot just a little bit more than 9 MW within 30 seconds, starting from the normal operating condition. That is a power increase of 17%. The pink line is when an increase of 9 MW over 30 seconds is specified in the system mentioned in subsection 4.10.4. The system designed for ramp up of oil. Here we can see that it works as intended. The power increase linearly by exactly 9 MW over 30 seconds to maintain that power in agreement with the requirements for frequency regulation. The blue and green lines indicate the response for both cases with the real valve. As can be seen in the plot, at 20 seconds, the power ramp up rate for both the maximum case and for the system is decreased. Here the valve to the turbine is fully opened. The pressure in the system before the turbine will therefore rise as can be seen in figure 7.9. When the pressure rises, the pressure drop across the valve increase and a larger flow may come through and thus the effect continues to increase, but at a slower rate.

Figure 7.9 is an extended plot of 7.8 that also shows the net pressure and the opening of the valve in the different cases. As an example, looking at the purple lines one can see that the power is maximized, the pressure utilized fully and the opening of the valve never maximized. Picture ?? is a close up of those pressure changes. Important to notice is that the valve connecting the turbine to the 12 bar net does not have the capacity to both meet the process steam demand at the 12 bar network and fill the accumulator tank at maximum speed. This can be seen in figure 7.4.

In figure 7.11 the ramp up system with only the P12 boiler installed has been used to ramp up the power. The red line indicates the power increase over time as the pressure is set to decrease linearly by 3 bar over the

course of 30 seconds with the use of a ramp function on the set point value to the PI-controller of that net. The pink line indicates that the power increase in addition to the power increase from a full oil fuel ramp up in the P12 boiler. Different ramp ups have then been tested to verify the control system. At 3.5 MW the accumulated steam in the pipes and drums is drained too fast to be able to maintain the power after 30 seconds, the same goes for the 3.2 MW case. 3- and 2 MW of ramp up works fine. The conclusion here is that with the use of P12 alone, from a normal operating condition the maximum amount of participation is 3 MW with the use of ramp up of oil alone. Figure 7.12 shows the opening of the valve at these specific cases.

To sum up, figure 7.13 with the most important process characteristics can be found in the appendix. Here the power is set to be increased by 3 MW with the use of the P12 alone.

In figure 7.13 and in the first plot the power is increased by 3 MW over 30 seconds. The pressure goes down and reaches a pressure drop of 4 bars after around 45 seconds. The pressure is then increased as excess steam starts to accumulate and the pressure is back after around 90 seconds. When the pressure starts to reach the set point value of 61 bar the fuel flow as can be seen in the plot below shuts. Since there is some time delay and time constants in the boiler, the fuel takes some time to burn. A small amount of steam will then firstly be throttled down and lastly dumped to limit the pressure of 61.5 bars. The accumulator is filled to the amount desired and is constant thereafter since the control of the accumulator has been taken away.

## 6 Conclusions

After research it was down to two different concepts for frequency regulation at Iggesund mill, which were modelled and tested. The potential for utilizing steam accumulated in steam drums and pipes was also tested, and as far as that goes it cannot alone be utilized for frequency regulation since it is emptied too fast, but it can be very helpful for the response at the concept of fuel ramp up.

In terms of the concept of fuel ramp ups in the boilers, according to specific data there is a time delay of 10-20 seconds for the oil in the burners to reach full flow rate, meaning that the power response would decrease substantially. The turbine has got inertia and there are other time constants that may contribute to a slower response, which is already tight. According to the participation requirements for PFR and equation 2.1 a droop of 2%-12% is accepted which corresponds to a minimal power increase or decrease of 3.33% - 20% of the actual power output which could be met, but only to a limited amount. If both P11 and P12 boilers would be used for ramp-ups, valves to the turbine and the accumulator will be limiting the response as they are maximized. Actual operation of the plant with this concept in use can be questioned, and as mentioned in section 3.1 Iggesund has the vision of operation, free from fossil fuel, which it would contradict.

As mentioned in the objective, the most simple and short term solutions were going to be looked at first. In this manner, to reserve power by reducing steam past the turbine seems to be to the best concept for frequency control, especially at Iggesund mill. There is no real drawback with this concept, as long as the amount of steam throttled and reserved for frequency regulation is being held, independent of operating condition and disturbances. With this concept the range of participation is not limited to more than the amount of steam that can be reduced past the turbine. In comparison to the fuel ramp up concept where the limit at best is down to the time constant of the boilers. If the compensation for frequency control and saved fuel is higher than that of sold electricity per MW reserved, this concept can be highly interesting.

### 6.1 Further Work

The more extensive model at Solvina could be expanded and renewed to work in the latest version of Dymola where frequency regulation should be implemented. The mechanical and thermal stress on the boilers and turbine could be looked at. It could also be of interest to look at the economics of different operating conditions, depending on the up going market for frequency control, if it is profitable or not.

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## 7 Appendix

In this chapter the figures and plots concerning the results from the project are presented. There is a short explanation of the plots beneath each figure.

### 7.1 Steamdrums and Pipes

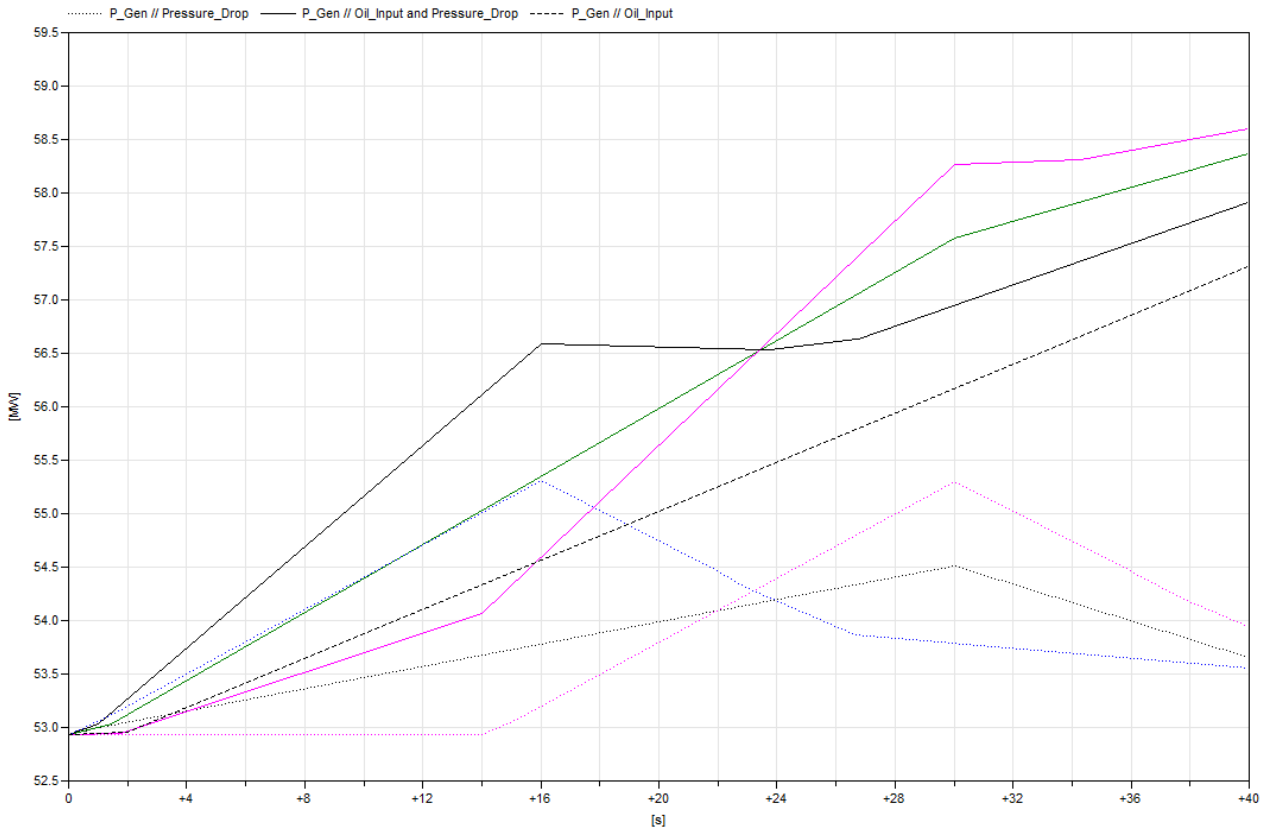


Figure 7.1: *The dynamic response of the pipes and drums at different rates.*

In figure 7.1 different responses are plotted as the accumulated steam in the drums and pipes are drained at different speeds in the P12 boiler, while it is burned at full oil load simultaneously. Dashed lines " $P_{Gen} // Pressure_{Drop}$ " indicate the power increase from a pressure drop in the net that start and peak at different times. The black dashed line in the middle " $P_{Gen} // Oil_{input}$ " is the oil ramp up alone and the full lines " $P_{Gen} // Oil_{Input}$  and  $Pressure_{Drop}$ " indicate both of these cases combined, at the same time.

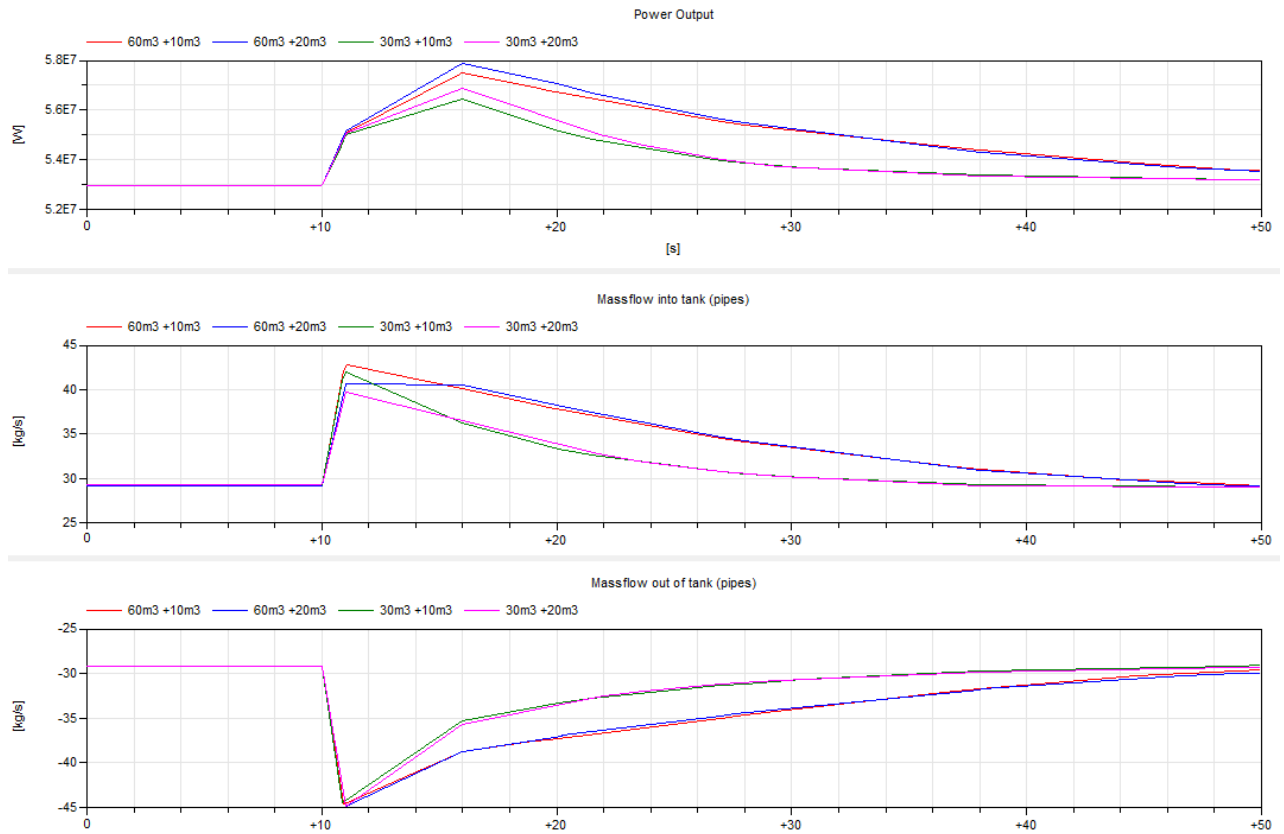


Figure 7.2: The dynamic response of the pipes and drums at different volumes.

In figure 7.2 the dynamic response of the pipes and drums has been tested and verified. Unlike figure 7.1, the accumulated steam in the drums and pipes are drained at the same speed but with different volumes. In the first plot four different size set ups of pipes and tanks has been tested. "30m3" and "60m3" refers to the volume of one or two steam drums, respectively. "10m3" and "20m3" refers to the volume of one or two pipelines connecting the boilers to the turbine. The second and third plot in figure 7.2 shows the inflow and outflow of steam in the drums and pipes.

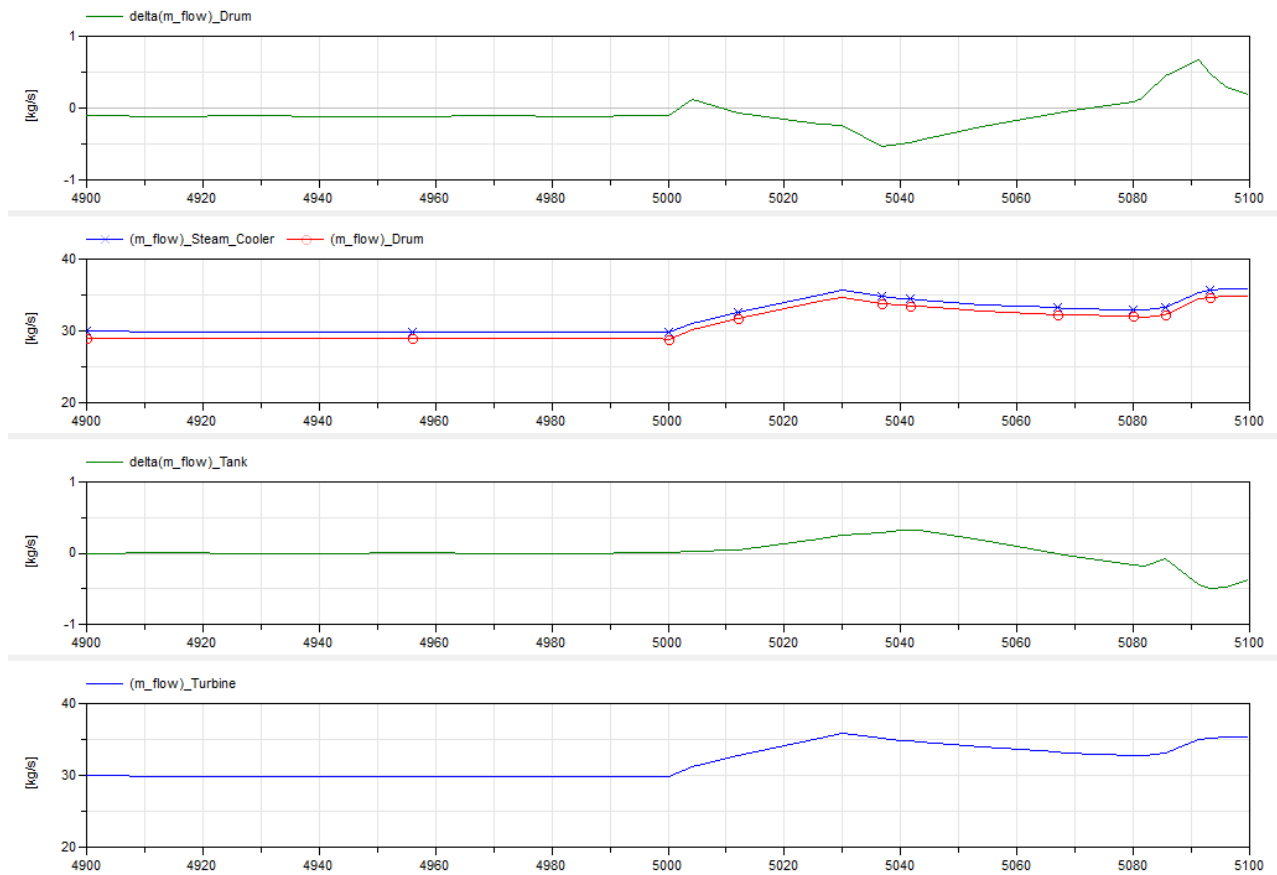


Figure 7.3: The dynamic response of drums and tanks during a 3 MW ramp up of the system with only P12 active.

In figure 7.3 the rate of change of flow into the pipes "delta(mflow)tank" and drum "delta(mflow)drum" can be seen in plot one and three. Plot two and four shows the massflow of steam into the drum, turbine and steam cooler.

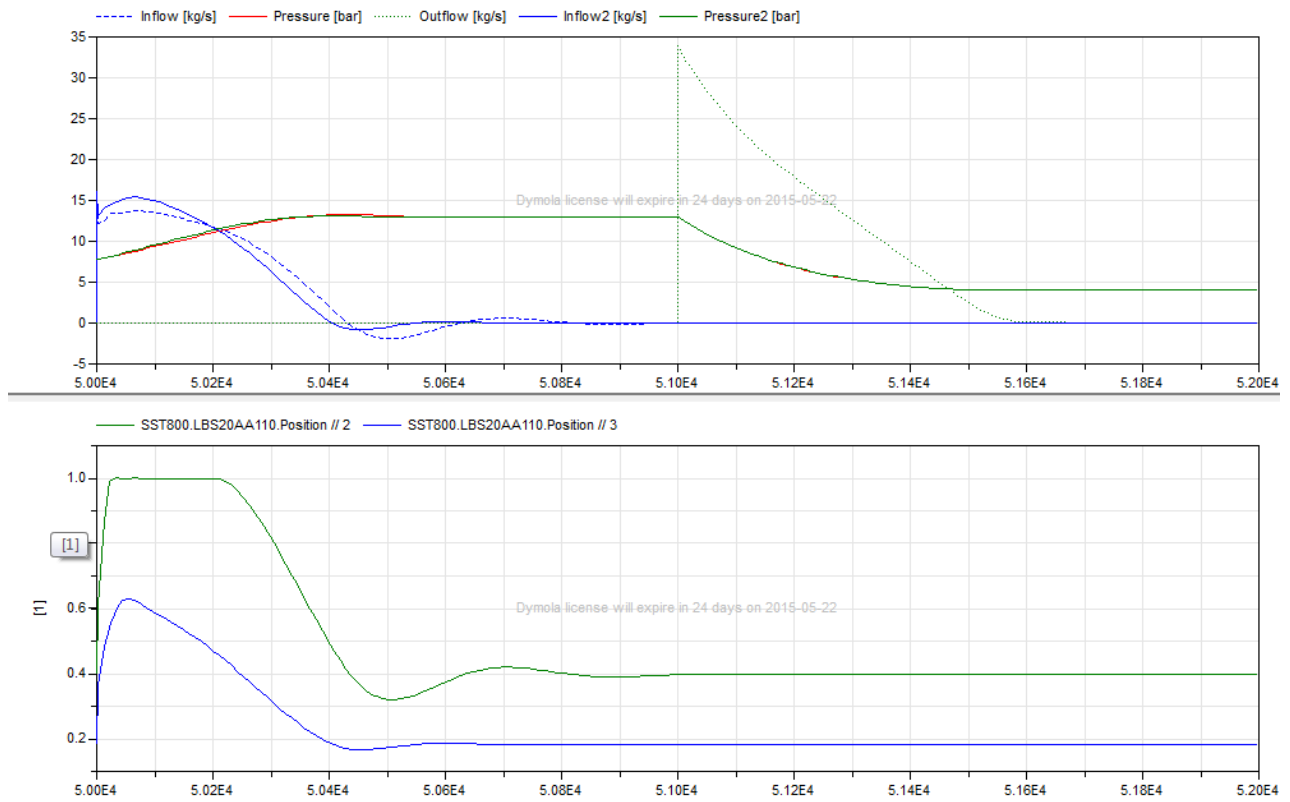


Figure 7.4: The steam massflow into the accumulator during "ramp-up" with two different sized bleed-valves from the turbine on the 12-bar net.

In figure 7.4 two different sizes of valves at the 12 bar bleed of the turbine are plotted during a "ramp-up". With the original valve size, the opening is maximized and the massflow through it is decreased compared to the bigger one, as can be seen in plot one and two respectively. "Inflow" refers to the steam-mass inflow to the accumulator, "Pressure" is the pressure in the steam accumulator tank and "Position" refers to the degree of opening of the valve.

## 7.2 Throttle

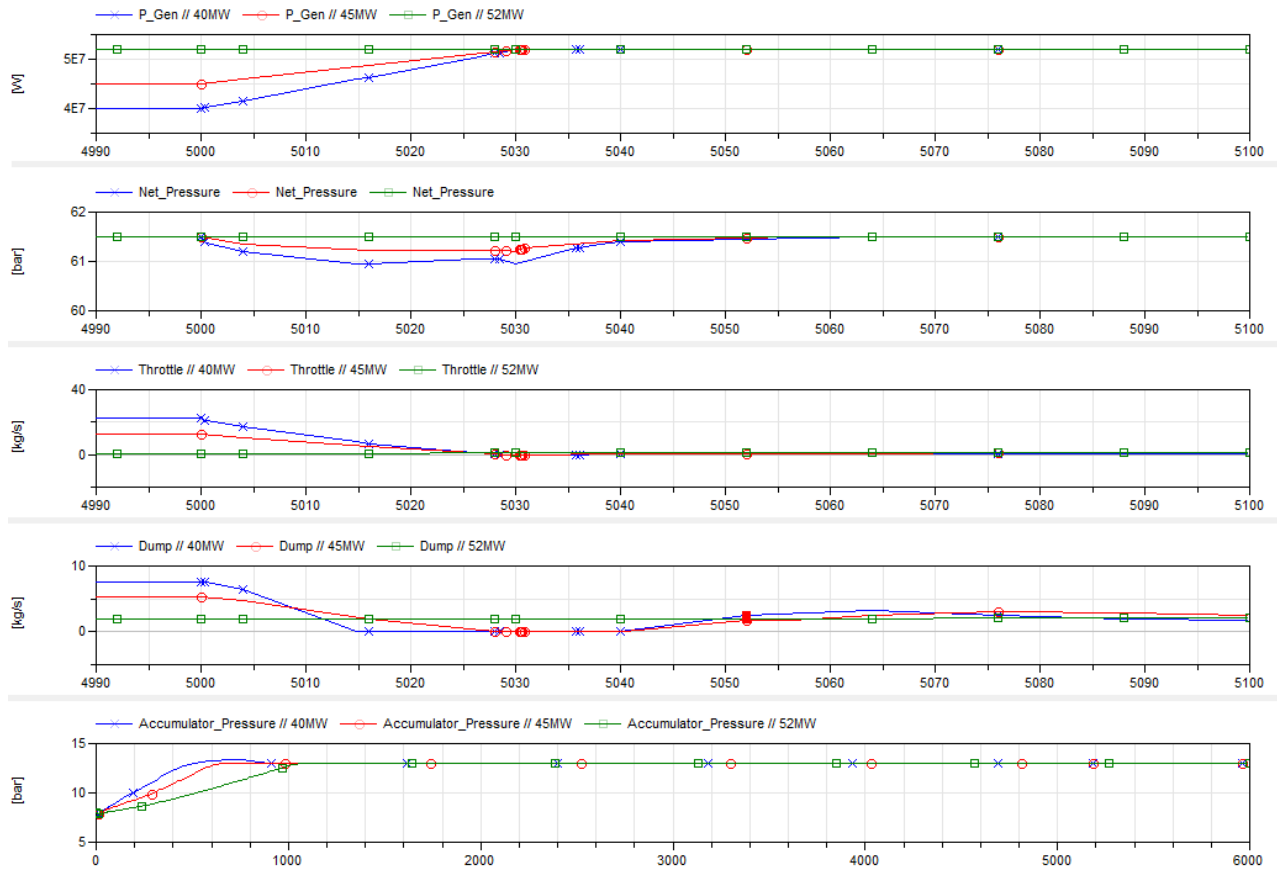


Figure 7.5: The dynamic responses of the "Throttle" concept at various power increases, with the accumulator control structure out of control.

In the figure 7.5 a constant power of 52 MW has been plotted against two cases where steam is throttled at 40 MW and 45 MW. " $P_{Gen}$ " in the first plot refers to the power generated for each case over time. " $Net_{pressure}$ " is the pressure in the net, and " $Throttle$ " the amount of steam directly reduced past the turbine to the consumers. " $Dump$ " refers to the amount of steam released to atmosphere at atmospheric conditions, unutilized. " $Accumulator_{pressure}$ " is the pressure inside the accumulator. The accumulator control structure is in all cases taken out of control, meaning that there will be excess steam which is "dumped" in all of these cases, as can be seen in the plot second lowest. In other words, exactly the same amount of steam is produced in all cases, the steam is just redistributed. When the power is increased the amount of steam "throttled" and "dumped" is decreased to later stabilize and be the same as the constant power case. The accumulator pressure is maximized, but could easily be limited.

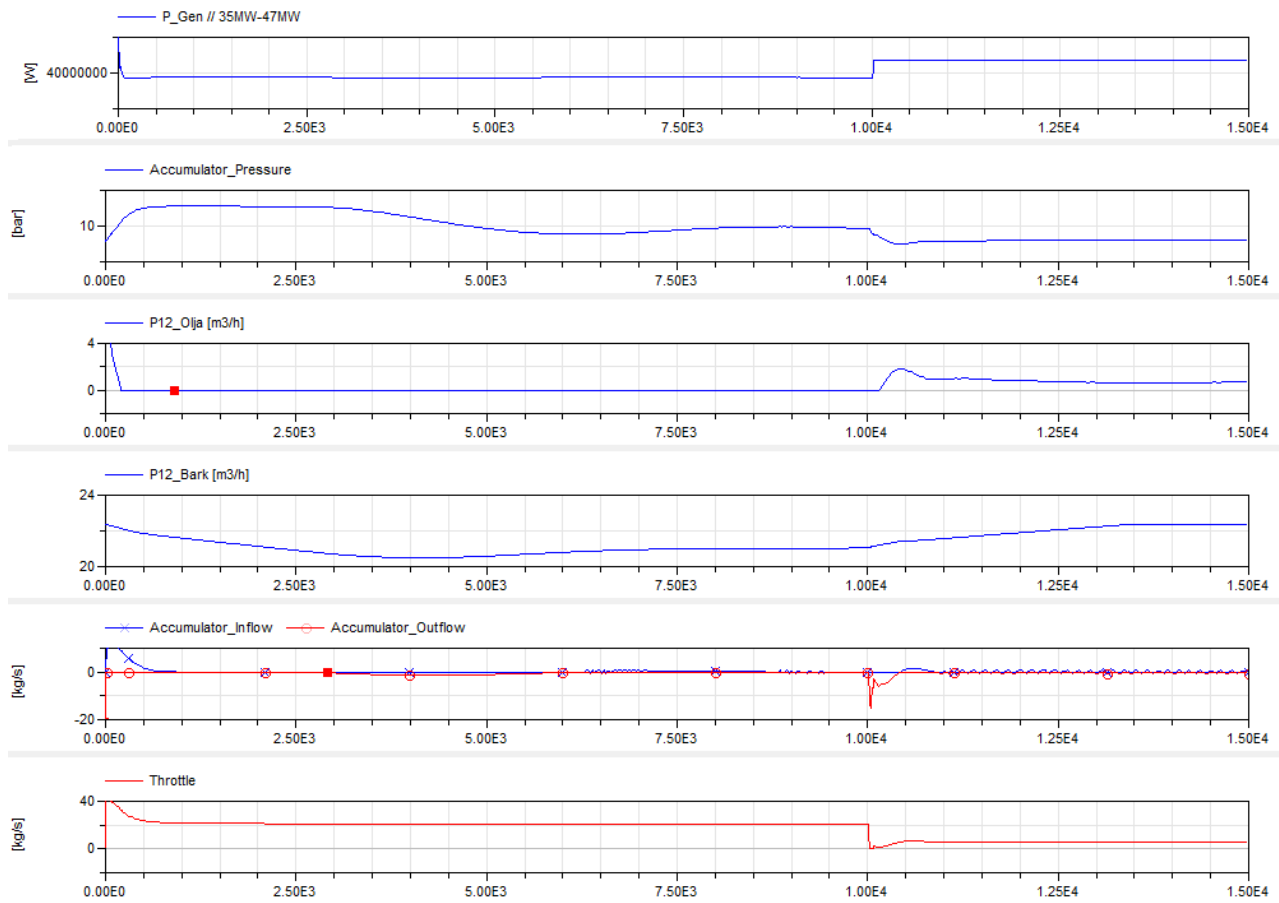


Figure 7.6: *The dynamic response of the "Throttle" concept at a specific power increase, with the accumulator control structure in control.*

In the figure 7.6 the accumulator control structure is, unlike in figure 7.5, in operation. In this single case the power is ramped up from 35 MW to 47 MW. As mentioned in subsection 4.10.3, when reserving steam for power ramp up with the accumulator control structure in use, the amount directly reduced past the turbine will decrease, as the fuel input decreases, to match the process demand. When the ramp up then occurs, there will be a sudden deficit of steam to the consumers that the accumulator has to match, as can be seen in the second lowest plot. The pressure in the accumulator then decrease. As the pressure in the accumulator decreases the fuel inputs to the boilers are increased in line with the accumulator control structure mentioned in subsection 4.10.2. This can be seen in plot 3 and 4.

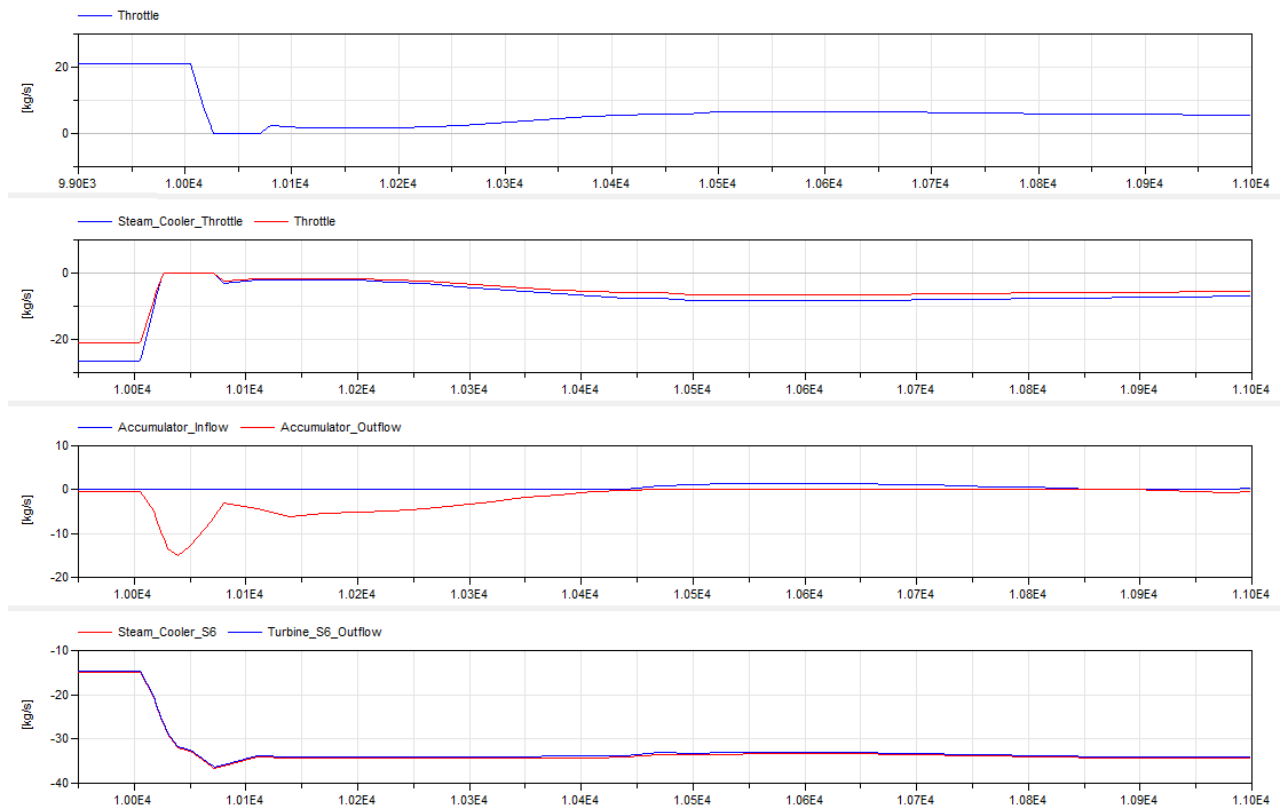


Figure 7.7: *The dynamic response of direct redistribution of steam past the turbine.*

In figure 7.7 about 20 kg/s of superheated steam is initially, directly reduced "Throttled" past the turbine. At a specific time the power is increased and the steam is redistributed into the turbine, as can be seen in the first plot. In the second plot it can be seen that the amount of steam going through the valve "Throttled" is 20 kg/s, whereas the amount of steam going past the steam saturator "Steam Cooler Throttle" is about 26 kg/s. In comparison with the forth plot, the steam saturator outflow after the turbine is basically the same as the inflow meaning that almost no water is injected. The extra injected water is covered by the steam accumulator during a sudden power increase as can be seen in the third plot.



### 7.3 Ramp up

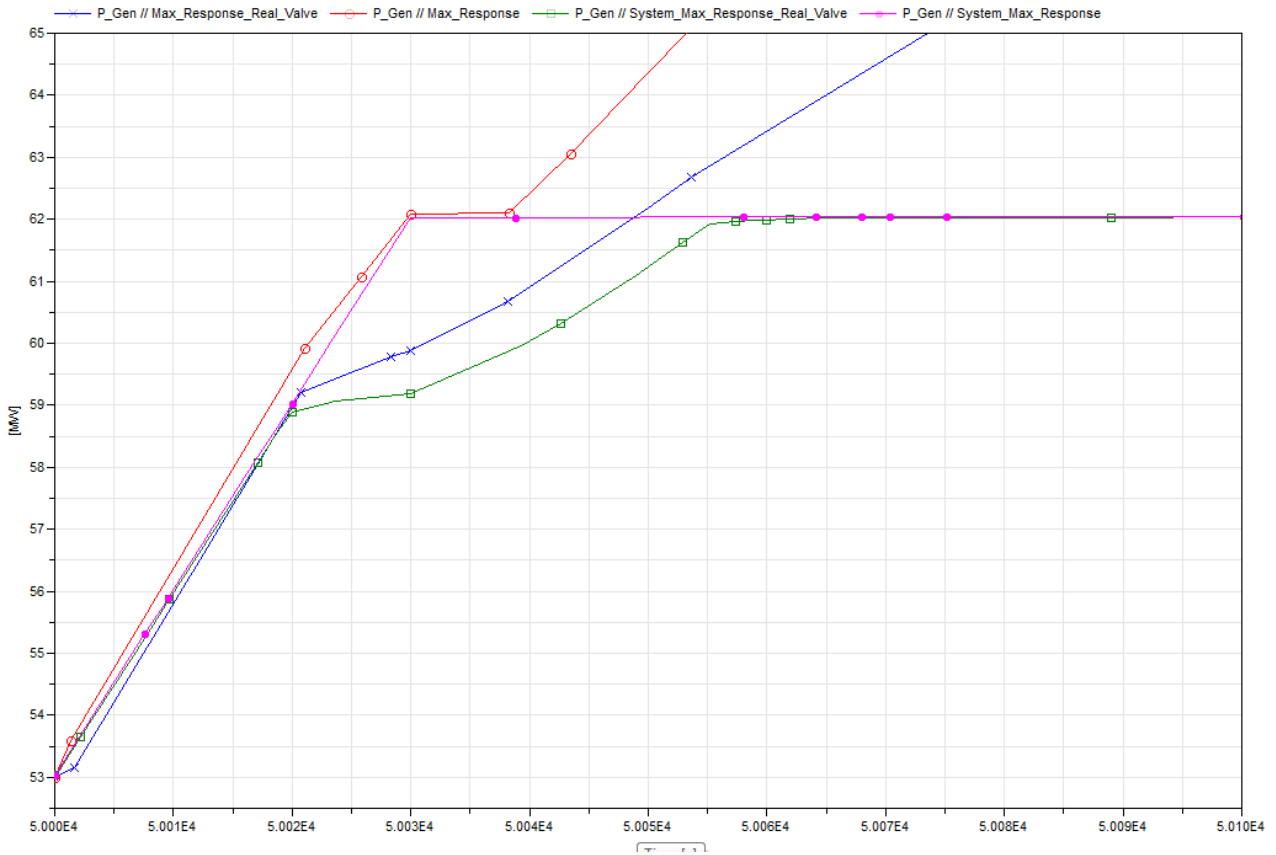


Figure 7.8: The power response of maximized oil fuel inputs and utilization of steam in drums and pipes with both P11 and P12 in operation.

In figure 7.8 the P12 and P11 boilers have had their oil fuel inputs maximized and steam accumulated in the tanks and pipes utilized fully (5% net-pressure reduction) to increase the power as much as possible under the course of 30 seconds. The red line "Pgen MaxResponse" gives the theoretical maximal response that is tested without any outside control. As can be seen in the plot the theoretical maximal response would be about 9 MW within 30 seconds. The purple line is the complete system response when a power increase of 9 MW over 30 seconds is set. The green and blue line represents the theoretically maximal and the real system's maximal response, respectively, with the real size valve.

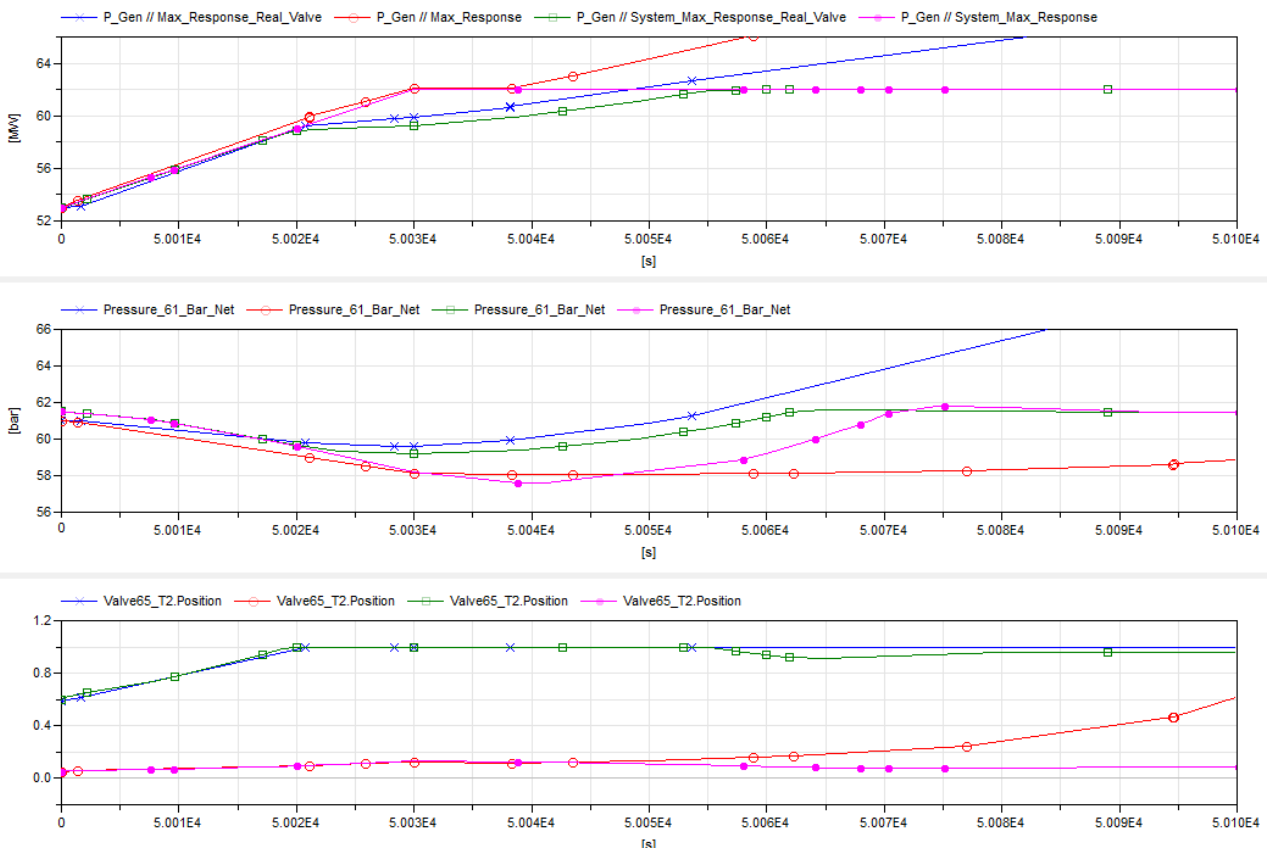


Figure 7.9: An extension of figure 7.8 showing the valve position and net pressure.

Figure 7.9 is an extended figure of 7.8 that also shows the net pressure and the opening of the valve in the different cases. The second plot represents the pressure in the net and the third the degree of opening of the valve.

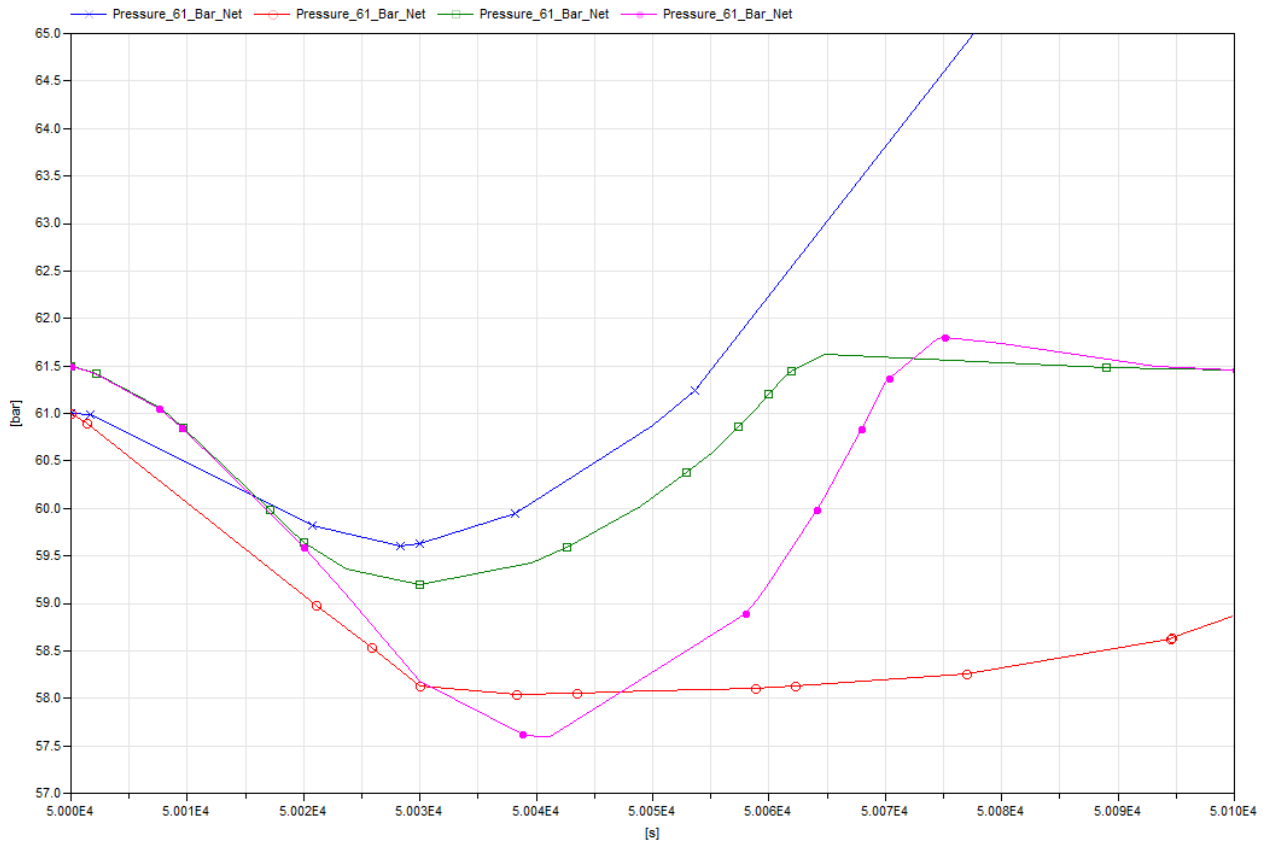


Figure 7.10: The pressure in the net as P11 and P12 are ramped up at different rates.

Figure 7.10 is an extension of plot 7.8 with the change of pressure in the net. Lines in both figures with the same color correspond to the same case and simulation.

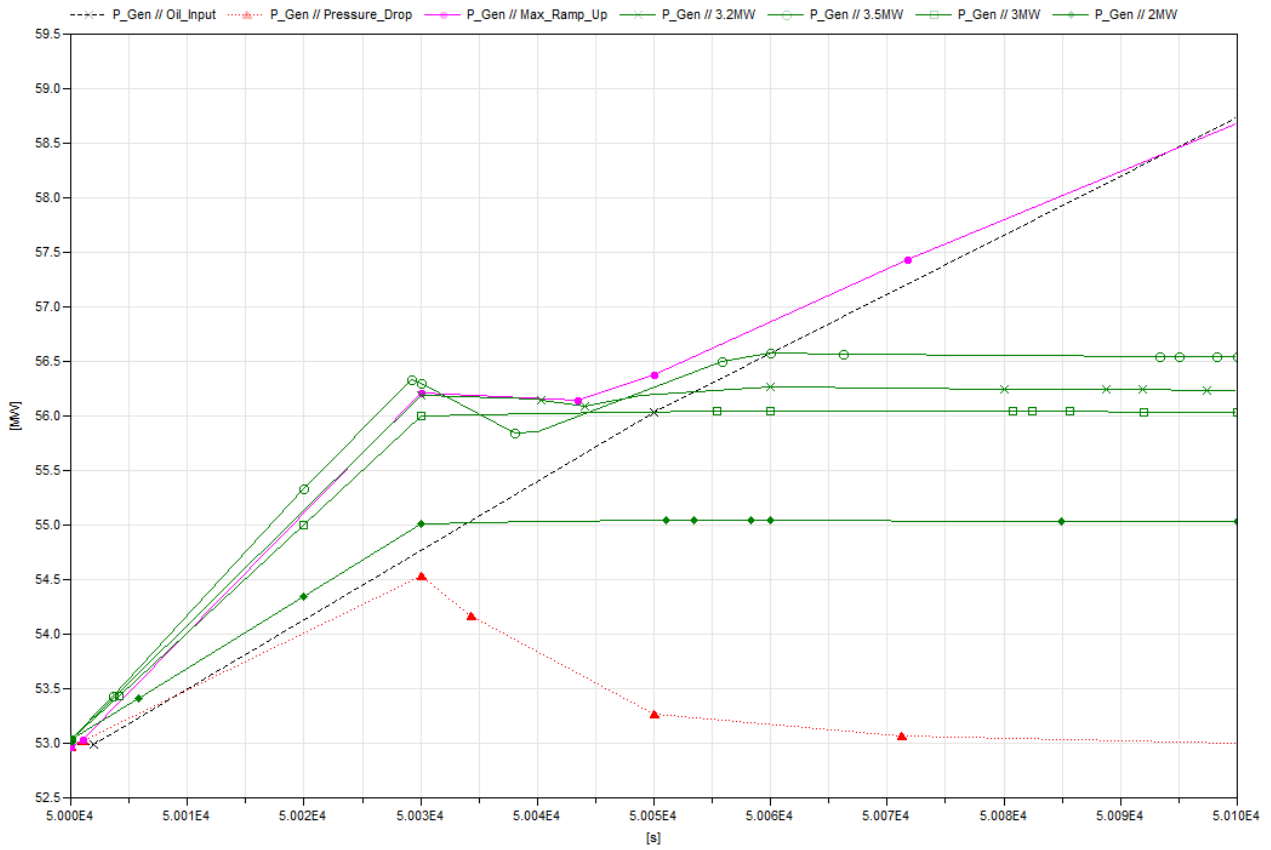


Figure 7.11: The power response of different cases with oil fuel inputs and utilization of steam in drums and pipes with P12 boiler alone.

In figure 7.11 the ramp up system with only P12 installed has been used to ramp up the power. The red line indicates the power increase over time as the pressure is set to decrease linearly by 3 bar over the course of 30 seconds with the use of a ramp function on the set point value to the PI-controller of that net. The pink line indicates that power increase in addition to the power increase from a full oil fuel ramp up in the P12. Different ramp ups has then been tested to verify the control system, corresponding to the green lines.

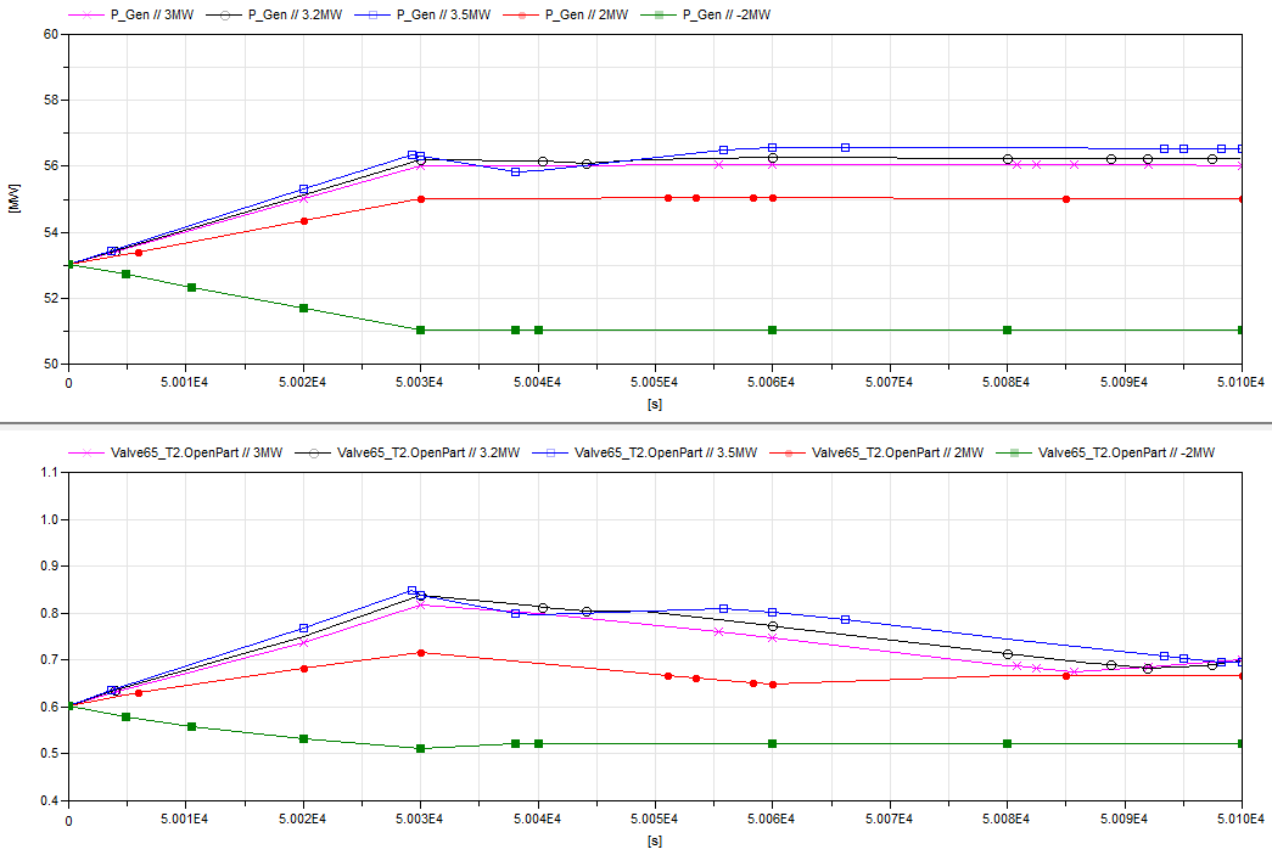


Figure 7.12: The valve opening when the oil input is increased in some different cases.

Figure 7.12 is an extension to figure 7.11 and it can be seen that the valve is never fully opened in any case.

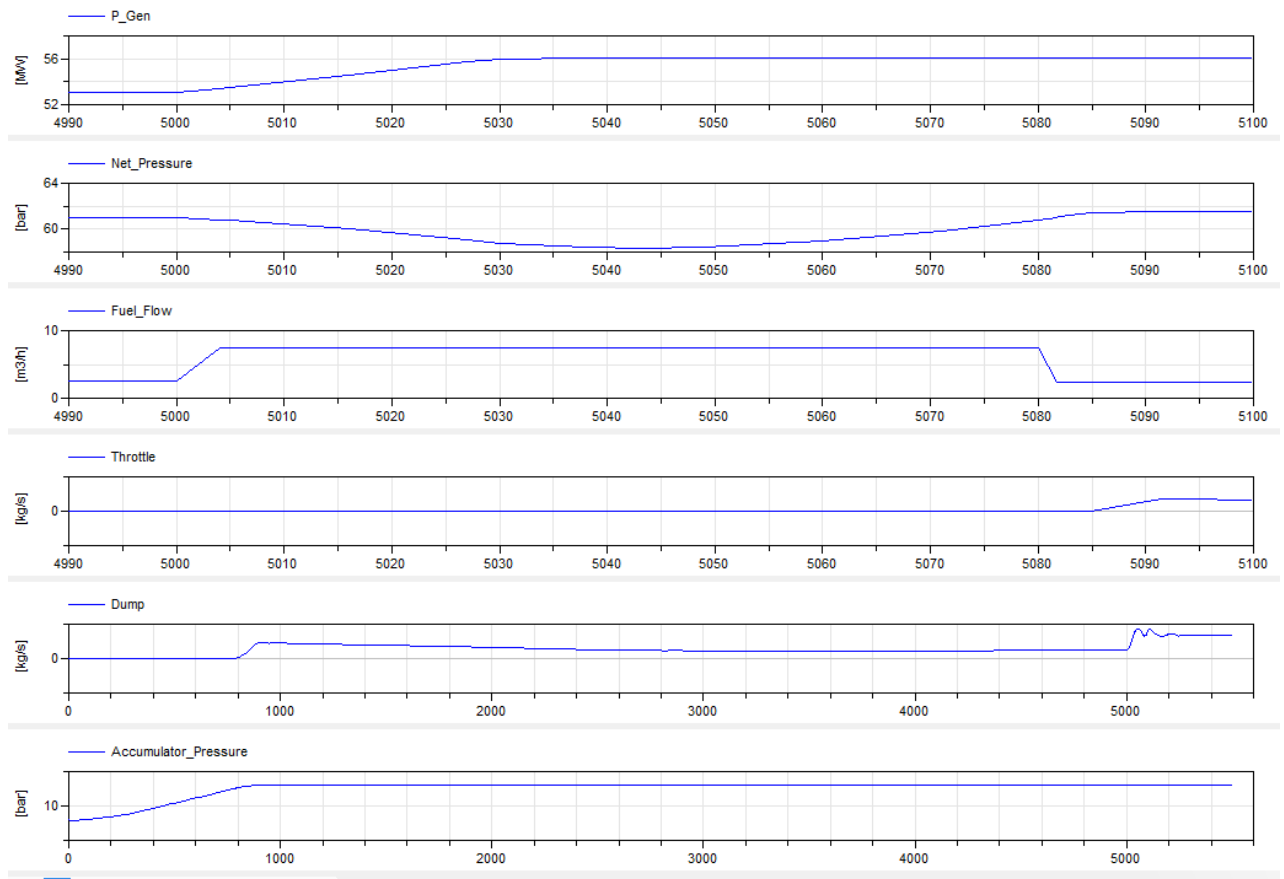


Figure 7.13: *The most important process characteristics during a power ramp up of 3 MW.*

In figure 7.13 the most important process characteristics can be seen, when the power is increased by 3 MW with the use of the P12 alone. Here the modelled control design "Ramp up" is in use.