Optimal Operation and Control of Heat to Power Cycles: a New Perspective from a Systematic Plantwide Control Approach

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Abstract

In this work, we use a plantwide control framework to systematically identify the control objectives, operational and environmental constraints, and degrees of freedom for a heat to power cycle with a drum, one pressure level and with power as the only valuable product. After controlling the active constraints and the unstable inventories, we are left with the fuel (MV1) as the only degree of freedom with a significant steady-state effect. However, the steam valve (MV2) can be used as a dynamic degree of freedom, to improve the response in transient operation while its steady-state effect is negligible. The result is an unified and systematic perspective on the optimal control operation problems for heat-to-power cycles.

Keywords: steam cycle, plantwide control, optimal operation, decentralized control

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Nomenclature

Controller

5	$1/R$	Droop proportional gain
	τ_C	Closed-loop time constant
	τ_I	Integral time
	τ_T	Tracking time constant
	CV	Controlled variable
10	K_c	Proportional gain
	LC	Level (holdup) controller
	mid	Logic block that selects the middle value of its inputs
	MV	Manipulated variable
	N/A	Not applicable
15	PC	Pressure controller
	PID	Proportional-integral-derivative controller
	TC	Temperature controller
	u	Controller output
	VPC	Valve position controller
20	WC	Power controller
	y	Controlled variable

Design Parameters

	C_v	Valve coefficient
	D_G	Electrical damping coefficient (machine base)

K_v Control valve coefficient

M_G Angular momentum of the generator (machine base)

UA   Heat exchanger specification

V   Volume

Physical Constants

μ Molar mass

c_p Specific heat capacity

R   Ideal gas constant

Subscripts

A   Attemperator

B   Bypass

C   Condenser

D   Drum

E   Economizer

G   Generator

P   Pump

S   Superheater

T   Turbine

V   Steam valve

Superscript

max   Maximum

min   Minimum

0	Inlet condition
g	Flue gas
mid	Middle
p	Process
ref	Reference
s	Steam (usually dropped)
sp	Setpoint
vap	Vaporization
w	Water (usually dropped)

Variables

ω	Grid frequency
H	Specific enthalpy
h	Enthalpy
L	Load
M	Holdup (mass)
m	Mass flow rate
n	Turbine speed
p	Pressure
T	Temperature
W	Mechanical Power
w	Algebraic states
x	Differential states
z	Valve opening

1. Introduction

Current industrial control solutions for thermal power plants[1] have evolved over the years based on industrial practices to a stage where it becomes less trivial to understand what are the operational objectives, constraints or degrees of freedom available for optimal operation. Moreover, their transfer to new cases or newcomers in the field may not be straightforward.

Often plant operators take established practices for granted, mainly because it has always been done in the same way. On the other hand, optimal operation changes with current operating conditions, i.e. feed composition, product specification, prices or equipment are subject to change during the operating life of a plant. However, it is difficult to identify the new optimal operation if the control policy is not systematically specified from the beginning.

This effect is felt stronger especially for steam cycles providing utilities (e.g. steam and power) for downstream units in chemical plants. In these cases, optimal operation of the steam cycles is often overlooked. However, considering the large amount of utilities used in chemical processing, there is much to gain from operating steam cycles at their optimum. For example, one of the question we address is what are the implications of controlling the superheated steam pressure. Often power plants are operated at constant pressure to provide faster changes in produced power. However, operation with floating pressure (i.e. the steam pressure follows the fuel rate) could potentially have higher efficiency at low load, for fossil fuel steam cycles (Silvestri et al., 1972) and especially for combined cycles (i.e. integration of a gas turbine with a steam turbine) (Polsky, 1982), or co-generation plants (i.e. plant providing both heat and power) (Jonshagen & Genrup, 2010).

This work expands the plantwide control analysis on a simple steam cycle presented in (Zotică et al., 2019) with a more detailed study. Therefore, the contribution of this work is twofold, and consists of a steady-state, and a dynamic

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[1]Thermal power plants, steam cycles and heat-to-power cycles are used interchangeably.

analysis. The former considers the optimal operation and control problem for a heat-to-power cycle and provide a clear and systematic procedure for identifying the operational objectives, specification or constraints and degrees of freedom from a steady-state point of view. The latter examines the dynamic performance of different control structures that can be used to implement optimal operation.

We accomplish these objectives in the framework of plantwide control, which handles control structure decisions for the entire plant. The goal is to find a control strategy, preferably a simple one, that acts on a short time scale to stabilize the plant (regulatory control), and on a longer time scale reaches optimal economic operation (supervisory control). The advantage of using a systematic plantwide control procedure is that it might reveal new control policies that might have been overlooked for existing processes (Downs & Skogestad, 2011).

Plantwide control has been extensively applied to chemical plants, and less attention has been given to heat-to-power cycles in the open literature. For example, the work by (Niva et al., 2017) presents a plantwide control analysis only for the combustion side of oxy-fired circulating fluidized bed boilers. The work by (Prasad et al., 2000) briefly discusses the use of a plantwide control approach to identify the main control objectives, operational constraints, degrees of freedom and controlled variables with the purpose of designing a model predictive control (MPC) strategy for a given thermal plant. The work by (Støre Govatsmark, 2003) applies a plantwide control design procedure to a combined cycle power plant. However, to the best of the authors knowledge a thorough analysis from a plantwide control of a steam cycle is missing, and it is therefore formalized in this work.

The structure of this paper is at follows, in Section 2, we describe the plantwide control procedure, which we then apply to a simple steam cycle in Section 3. In Section 4 we present different control strategies and operation modes for which we analyze the dynamic performance in Section 5. In Section 6 we discuss other aspects related to steam cycle operation and performance such as energy efficiency with floating pressure, influence of the response time of the drum level control, steam turbine control, and operation with given fuel rate.
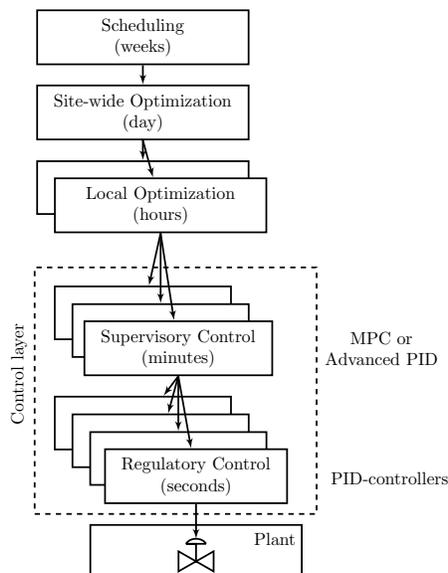
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Figure 1: Typical control hierarchy in a process plant.

2. Plantwide control

The typical control hierarchy in a process plant is decentralized and is decomposed on a time scale basis into several simpler layers: scheduling (weeks), site-wide optimization (days), local optimization (hours), supervisory control (minutes) and regulatory control (seconds), as shown in Fig. 1. Note that some processes can be slower. The top layers are responsible for production planning on a long time scale, while the lower control layer implements the setpoints given by the upper layer for optimal economic operation and stabilizes the plant. Each layer receives process measurements from the layers below, solves an optimization problem by using as degrees of freedom the setpoints to the lower layers (Skogestad, 2004).

To systematically design each layer, we use the plantwide control procedure proposed by (Skogestad, 2004). The procedure consists of a top-down analysis concerning optimal steady-state operation, and a bottom-up analysis targeting the lower control layer structure. The steady-state top-down analysis involves the following steps:

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Step 1 Define the optimal economic operation problem: the objective cost function J and the set of operational constraints.

Step 2 Identify the steady-state degrees of freedom (DOF) (i.e. setpoints for the lower layers). Determine the optimal operation for expected disturbances using a steady-state model.

Step 3 Implement optimal operation. Select the primary controlled variables (CV') as the active constraints from *Step 2*, and the self-optimizing variables (for unconstrained degrees of freedom) (i.e. variables that give acceptable loss when kept at constant setpoint).

Step 4 Choose the location of the throughput manipulator (TPM), i.e. decide where to set the production rate. This is both a dynamic issue (with implications on the inventory control structure design), and an economic issue (minimize back-off from active constraints).

The bottom-up design focuses on the control layer, which is divided into the supervisory and the regulatory control layer.

The regulatory control layer typically takes care of control on the fastest time scale. Controlled variables in the regulatory layer (CV') include variables that contribute to "stabilization" of the process, for example levels and pressures. In addition, they usually include a subset of the economic controlled variables (CV), typically active constraints, that should be tightly controlled for economic reasons. The regulatory layer is usually not subject to reconfiguration, so one should be careful about what happens if one has MV saturation in this layer (Reyes-Lúa & Skogestad, 2019). Considering the large number of control loops in a typical plant, simple PID-controllers are used for the regulatory layer.

The objectives of the supervisory (advanced) control layer are:

1. Achieve the economic objectives given by the upper optimization layers by controlling the primary CVs at setpoint using as degrees of freedom the setpoints to the regulatory layer or any unused manipulated variables.

2. Monitor the regulatory stabilizing layer to avoid saturation of MVs.

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3. Identify active constrains and self-optimizing variables changes based on the current operation region, and switch the control structure.

The steps of the bottom-up design are:

Step 5 Design the structure of regulatory control layer. The main issues are: first, to select what to control on a fast time scale, both for stabilizing control, and to achieve tight control of important active constraints, and second, to chose appropriate MVs and pairings.

Step 6 Design the structure of supervisory control layer. Decide between centralized control (i.e. Model Predictive Control) or decentralized control (i.e. advanced control structures with simple logic block to handle changes in active constraints (Reyes-Lúa et al., 2018).)

Step 7 Design the real-time optimization layer. Its objectives are to identify the active constraints and compute the optimal setpoints for the lower supervisory layer. For many plants, this layer is missing as it requires a full model.

3. Plantwide control for a simple heat to power cycle

3.1. Process Description

We consider the steam side of a heat-to-power cycle as shown in the simplified process flowsheet in Fig. 2. Fuel is burned with air in stoichiometric ratio in a combustion chamber resulting in high temperature flue gases. Thermal energy carried by the flue gas superheats the working fluid (water) in a boiler. Then, it is converted to mechanical energy in a turbine, followed by conversion to electrical energy (W) in a generator connected to the grid. In this paper, we consider only the steam side of the process, that is, the combustion side is excluded.

A detailed representation of the boiler-turbine system is shown in Fig. 3. The circulating working fluid (water) is heated from liquid (blue) to high-pressure superheated steam (red) by receiving heat resulted from burning fuel
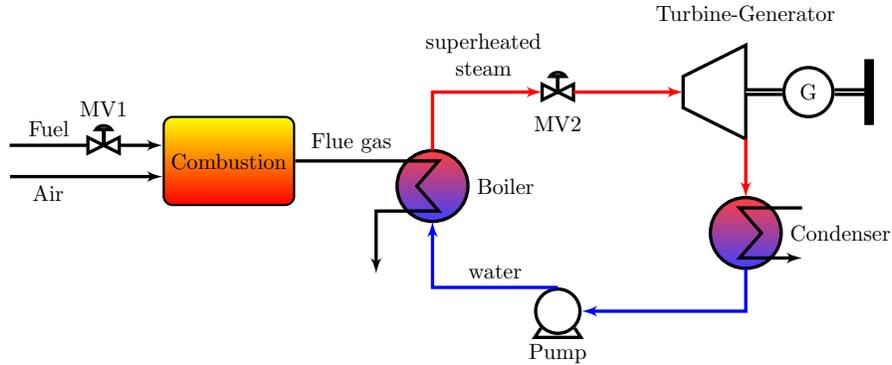
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Figure 2: Simplified heat-to-power cycle flowsheet. The air feed is set in ratio to the fuel, but the combustion side of the process is not included in this work.

(MV1) (black) in a series of three heat exchangers dedicated to well defined regimes, i.e. economizer (heating to saturated liquid), drum (evaporation) and superheater (superheating). The superheated steam is desuperheated by spraying cold feed water in the attemperator, therefore this is a bypass stream of the three heat exchangers. The superheated steam is expanded in a condensing type turbine, which drives a generator supplying electricity to the electric grid. Cooling water (MV4) is used as utility in the condenser. The low pressure water is then boosted by a variable speed pump (MV5) and it is fed to the boiler (i.e. economizer). The cycle process also includes a bypass of the turbine (MV3), and a direct bypass of the economizer cold side (MV6).

We choose this drum configuration over a once-through boiler (with a single heat exchanger instead of three) because it is most common both in operating power plants, and in chemical plants with on-site steam generation. The once-through boiler is in theory more efficient because it does not have the requirement of saturation (and thereby a fixed feedwater for a given steam pressure) at a given location inside the heat exchanger sequence. For a once-through boiler, the feedwater (MV5) may be used to control the steam temperature. We choose a single steam pressure level because we want to have a simple base for our analysis on which we can expand. For the same reason, we do not include
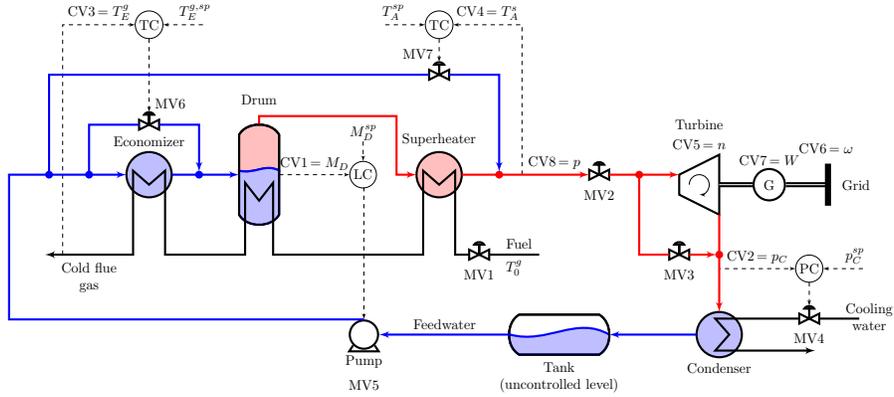
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Figure 3: Flowsheet of a steam cycle with a drum boiler, one pressure level, and condensing turbine. The system has 7 manipulated variables (MVs). There are shown 8 potential controlled variables (CVs). After closing 4 regulatory loops for temperatures, pressure and level and noting that the bypass MV3 should be kept closed if possible, MV1 (fuel) and MV2 (steam valve) are the two remaining degrees of freedom (See also Table 1) and Section 3.2.4). Liquid water is in blue, vapor in red.

steam extraction or back-pressure turbines.

3.2. Top-down analysis

We proceed to formalize the control problem for steam cycles by applying the top-down analysis to the described process. Therefore we systematically iden-
tify the control objectives, operational and environmental constraints, degrees of freedom, main disturbances and the location of the throughput manipulator.

3.2.1. Step 1. Operational objective

The plant has two operational objectives. On a slow time scale (steady-state) it should achieve the economic optimum, while it contributes to the grid stability on a fast time scale. Due to the time scale separation, these objectives are decoupled. However, the grid stability requirement may impose a back-off from the maximum power production. Depending on local conditions, the main operational objectives are:

1. Produce the energy as

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(a) power to the electric grid at the required voltage and frequency (usu-
ally large power plants with condensing turbines, i.e. $> 100\,\mathrm{MW}$);

(b) steam at the required flowrate and pressure level (usually for back-
pressure turbines in large chemical plants);

(c) power and steam (combined heat and power cycles);

2. Process a given amount of by-product (e.g. waste gases or biomass residues).

The same economic cost function, i.e. minimize the negative profit, can be
defined for all operational objectives, given by Eq.1.

$$J = -(p_W W + p_S S - p_F F - p_U U) \qquad [\$/\mathrm{s}] \qquad (1)$$

Here, W $[\mathrm{J\,s^{-1}}]$ is the produced power, S $[\mathrm{kg\,s^{-1}}]$ is the produced steam ($= 0$
in this paper), F $[\mathrm{J\,s^{-1}}]$ is the fuel (energy source), U $[\mathrm{kg\,s^{-1}}]$ is the utility
consumption, and p $[\$\mathrm{kg^{-1}}]$ or $[\$\mathrm{J^{-1}}]$ is the price of each. There may be addi-
tional terms, for example several feed energy sources or several steam products.
We analyze an operating plant and therefore, capital costs, personal, and main-
tenance costs are not included. The cost J should be minimized subject to
satisfying a set of constraints, related to products specifications, safe operation
and regulations related to the environment. Typical constraints for the opera-
tional objectives listed above for a steam cycle include (Prasad et al., 2000):

C1 Keep the electrical power (W) at a given value. This is for plants required
to participate in grid frequency regulation, i.e.$W \geq 100\,\mathrm{MW}$).

C2 Produce steam at the required demand (for cycles providing steam as
utility for chemical plants, and not included in the described process).

C3 Stabilize the process (i.e. keep the unstable drum level within limits).

C4 Keep the temperature of the superheated steam at a given value to maxi-
mize turbine work, but within boundaries to prevent large thermal gradi-
ents (i.e. $T_A^s = 529°\mathrm{C}$).

C5 Keep the superheated steam pressure below a maximum value to avoid

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high thermal and mechanical stress and to extend the operating life (i.e. $p \leq p^{\mathrm{max}} = 220\,\mathrm{bar}$).

C6 Keep the steam pressure above a minimum value to avoid boiler trip (i.e. $p \geq p^{\mathrm{min}}$).

C7 Keep the temperature (T_E^g) of the flue gas outlet below environmental limits, and above dew point to prevent corrosion °C ($T_E^g \geq 150\,\mathrm{°C}$). Note that only plants with a higher concentration of pollutants (NO_x or SO_2 have constraints on the maximum temperature, due to operation limits on the filters used to reduce emissions.

C8 Keep MV4 fully open (i.e. MV4=MV4$^{\mathrm{max}}$) to bring the condenser pressure at lower limit to maximize the pressure ratio in the turbine (i.e. $p_C = 0.1\,\mathrm{bar}$).

C9 Keep the turbine speed at the setpoint ($n = 50\,\mathrm{Hz}$). If connected to the grid, control is only needed at short time scale to avoid wear, because on a long time scale, the turbine speed is given by the grid frequency.

Note that industrial turbines are normally operated at constant speed, which can be the grid frequency or a different frequency (usually higher), depending on their design. In the latter, a gear box is used, but the turbine speed is still kept constant and it is not a degree of freedom available for operation. Variable speed turbines may be used for experimental low load organic Rankine cycles. However, variable speed turbines are out of the scope of this work, and the interested reader is refereed to the work by (Quoilin et al., 2011).

In addition to constraint C7, there are other operational constraints on the combustion side, including requirements for waste incineration, O_2, CO_2 and NO_x percentage in the flue gas or furnace pressure. However, a detailed analysis of the combustion side is outside of the scope of this paper, and we assume that these operational objectives are met on the combustion side of the process. The interested reader is referred to the work by (Niva et al., 2017) for an analysis on the combustion side for an oxy-fired circulating fluidized bed boiler.

3.2.2. Step 2 (a). Identify the steady-state degrees of freedom (DOF) (i.e. set-
290   *points for the lower layers)*

Table 1 shows the degrees of freedom together with comments on their implication to control. The MVs are also shown in Fig.3. Note that we have not decided yet on the pairing, and number of the MV and CV are not corresponding in the next sections (i.e. MV1 is not necessarily used to control CV1).

Table 1: Manipulated variables

Manipulated variable	Comments, analysis
MV1: Fuel	• At steady-state the power produced can only be changed by manipulating (MV1).
MV2: Steam valve	• The turbine valve should optimally be fully open to minimize throttling losses (Shinskey, 1978). • It has mainly a dynamic effect, as the steady-state effect on produced power is insignificant. • It can improve the dynamic response for power. • To contribute to grid frequency stability it may be required at nominal conditions to partly close the steam valve opening (e.g. 90 %). This will provide a back-off for transient operation.
MV3: Turbine bypass	• Normally closed, needed when the energy in the feed is larger than power demand. • Used to avoid too high pressure, i.e. if $p \geq$ 220 bar, MV3 opens to reduce the pressure.
MV4: Cooling water	• Open at MV4=MV4$^{\mathrm{max}}$ to minimize condenser pressure (p_C).
MV5: Feedwater pump	• Use only to control the drum level. • Cannot be used to control the steam pressure, as it has no steady-state effect (see Table 3).
MV6: Economizer bypass	• Use only if $T^g < T^{g,\mathrm{min}}$.
MV7: Attemperator	• Use only if $T^s > T^{s,\mathrm{max}}$.

295   *Steady-state effect of fuel (MV1) and steam valve (MV2).* Fig. 4 shows the open loop response for the superheated steam pressure (CV8 = p), and power produced (CV7 = W) to 1% increase in fuel MV2 in blue, and to fully opening the steam valve (MV2 = 1) in green. Let us explain the open loop response from physical considerations. Consider the linear valve $m = zK_V\Delta P$ where,

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m is the mass flow rate, z is the valve opening, K_V is the valve coefficient and ΔP is the pressure drop across the valve. Increasing z causes a fast increase of m, which results in a decrease of the pressure inventory before the valve. The latter results in a smaller ΔP, which results in a decrease of m after its initial increase. To increase m at steady-state, ΔP has to increase, and this can only be achieved by increasing the energy supplied to the system (MV1).

(a) Power   (b) Pressure
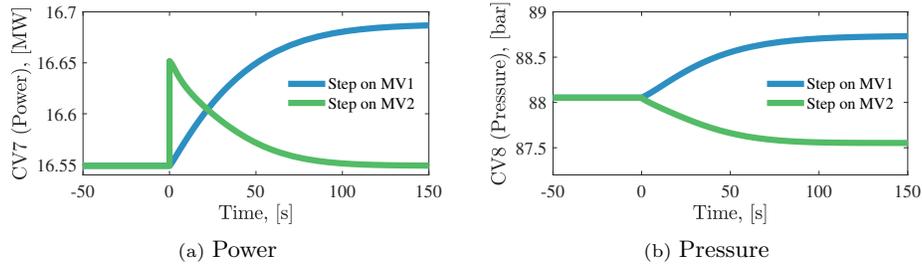
Figure 4: Open loop responses for pressure (p) at the valve inlet and power (W) to a step increase of 1% step increase in fuel (MV1) (green), and 0.1 in steam valve opening (MV2) (blue).

3.2.3. Step 2 (b). Identify the most important disturbances

The main disturbances for this process are given in Table 2.

Table 2: Main disturbances for steam cycle

Disturbance variable	Comments
DV1: Combustion temperature	Typically for waste heat
DV2: Fuel specific heat	Typically for waste heat
DV3: Grid frequency (Load)	Consumers increasing their demand (load) or producers decreasing their production
DV4: Required power setpoint	Typically for power plants required to participate in secondary or tertiary grid frequency regulation
DV5: Cooling water temperature	

3.2.4. Step 2 (c). Determine the optimal operation (including active constraints) for the expected disturbances using a steady-state model

Active constraints (AC) are variables that should be kept at their limiting value for optimality. To determine which constraints will be active, we can

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optimize the process at steady-state for the important disturbances. However, engineering insight is often enough to determine which constraints are active, and this is the approach we apply in this work. At the nominal operation we want to minimize bypass streams, that is the turbine bypass MV3, the economizer bypass MV6 and the attemperator MV7 should be closed to use the boiler efficiently. However, when a CV constraint becomes active, we use the MV to control the respective CV. This implies a CV-MV switch, and it can be handled by single loop PID-controllers without additional logic given that antiwindup is implemented (Reyes-Lúa & Skogestad, 2019).

The active constraints are:

(AC1) MV3=0 (MV constraint);

(AC2) MV4=MV4$^\mathrm{max}$ (MV constraint) or CV2$= p_C = p_C^\mathrm{min}$ (CV constraint) to maximize pressure ratio across the turbine and maximize work (W);

(AC3) MV6=0 (MV constraint) or CV3$= T_E = T_E^{g,\mathrm{min}}$ (CV constraint) to maximize boiler heat transfer area usage;

(AC4) MV7=0 (MV constraint) or CV4$= T_A^s = T_A^{s,\mathrm{max}}$ (CV constraint) to minimize desuperheating and maximize superheated steam temperature;

(AC5) $n = \omega$, (i.e. the turbine speed is equal to the grid frequency).

We use the term *or* for AC2, AC3 and AC4 because maximizing cooling (MV4=MV4$^\mathrm{max}$) results in $p_C = p_C^\mathrm{min}$, closing the economizer bypass stream (MV6=0) gives minimum temperature $T_E = T_E^{g,\mathrm{min}}$, and closing the attemperation stream (MV7=0) gives maximum $T_A^s = T_A^{s,\mathrm{max}}$. When $p_C < p_C^\mathrm{min}$, we give-up MV4=MV4$^\mathrm{max}$ and use MV4 to increase p_C. When $T_E < T_E^{s,\mathrm{min}}$ we give-up MV6=0 and open MV6 to increase T_E^g. When $T_A^g > T_A^{s,\mathrm{max}}$ we give-up MV7=0, and open MV7 to decrease T_A^g. As mentioned, this CV-MV switch is handled by PID-controllers without additional logic block.

3.2.5. Step 3. Economic controlled variable (CV) selection

The objective is to select controlled variables such that we keep optimal (or near optimal) operation when disturbances occur. The first controlled variables candidates are the active constraints from Section 3.2.4, as well as variables that need to be controlled to stabilize the process. Table 3 shows the possible controlled variables including the active constraints (a subset of the operational constraints from *Step 1* in Section 3.2.1).

Table 3: Candidates controlled variables

Controlled variable	Comments, analysis
CV1: Drum level (M_D)	• Levels are unstable inventories and they need to be controlled • No steady-state effect
CV2: Condenser pressure (p_C)	• see Section 3.2.4
CV3: Cold flue gas temperature (T_E^g)	• see Section 3.2.4
CV4: Superheated steam temperature (T_A^s)	• see Section 3.2.4
CV5: Turbine speed	• Active constraint for all operation regions.
CV6: Grid freqeuncy	• Imposed by grid stability
CV7: Power produced	• Only for plants required to participate in grid frequency regulation
CV8: Steam pressure	• Given by the fuel (MV1) according to the boiler energy balance • Should not be at fixed setpoint to utilize the fuel and boiler efficiently

3.2.6. Step 4. Location of throughput manipulator

The location of the throughput manipulator (TPM) is important from a dynamic point of view as it determines the structure of the inventory (pressure) control system and also affects the dynamic performance for cases when the TPM is used for control. In general, the TPM can be located at the feed, inside the process or at the product. For a power plant, the product is the power output, but with a fixed turbine speed, there is no way of manipulating it directly, Thus, for a power plant, there are only two possibilities:

• TPM at the feed, that is, the fuel (MV1) is the TPM

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- TPM inside the plant, that is, the steam valve (MV2) is the TPM.

In many cases, the fuel rate is given (typically for base load boilers) or the fuel rate is limiting the power output. In such cases, we clearly want to have the TPM located at the feed (MV1) in order to maximize power production. This case is considered briefly in the discussion section, but otherwise the main focus of this paper is when the power demand is given. To track variations in the power demand, it would be best from a dynamic point of view to locate the TPM at the steam valve (MV2) at the inlet to the power-producing turbine. Nevertheless, most control structures use the fuel (MV1) as the TPM also in this case, and there are several main reasons for this. First, it may happen that one would like to operate with a fixed fuel under some conditions. Second, as seen from Fig. 4, the steady-state effect of the steam valve (MV2) on the power (W) is very small when we have a constant fuel rate (MV1) and in addition we may want to operate with a fully open steam valve to minimize throttling losses. Third, the pressure drop over MV2 is more a dynamic performance matter. The higher the pressure drop, the higher the energy and mass stored in the boiler, and therefore the system has better capability to change the load at the required rate (e.g. %/min).

3.3. Bottom-up design

We continue with the bottom-up design for the described process.

3.3.1. Step 5. Structure of the regulatory layer

Liquid levels generally need to be controlled to maintain stability (see Section 2). The power cycle in Fig. 3 contains two liquid levels, but since this is a closed system only one of them should be controlled, usually the smallest holdup. Thus, we decide to control the boiler drum level (CV1 = M_D) and leave the feedwater tank level uncontrolled. The steady-state value of M_D does not matter, except that it contributes to energy storage, which has dynamic implications. Next, the steam pressure (CV8 = p) is often controlled because it may be drifting, and control of it may contribute to more stable and predictable operation. However,

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as we will see, control of steam pressure requires closing the steam valve (MV2) which gives losses and is not optimal from an economic point of view. We will therefore not include control of CV8 in the regulatory layer, but will leave for the supervisory control layer (step 6 in Section 4). Condenser pressure (CV2 = p_C) is usually also controlled, both because this contributes to stability and because it is optimal to keep it above its lower constraint to avoid too much liquid at the outlet of turbine. Two other constraints that are controlled in the regulatory layer are superheated steam temperature (CV4 = T_A) and cold flue gas temperature (CV3 = T_E). CV4 must be below a maximum for material reasons in the turbine and CV3 should be above a minimum, for example, to avoid corrosion caused by condensation. In the regulatory layer, we usually use single-loop PID control, so for each CV we need to identify an appropriate input (MV). We can make a decision based on mathematical tools such as the relative gain array (RGA). Alternatively, as in this work, we can use guidelines such as the pair close rule (i.e. small effective time delay from the MV to CV), or, input saturation rule (i.e. pair an important CV (which cannot be given-up) with an MV that is unlikely to saturate (Reyes-Lúa et al., 2018)).

We have 7 manipulated variables, but for economic reasons the turbine by-pass (MV3) should always be closed. The steam valve (MV2) and fuel (MV1) will be used for control of power production and pressure in the supervisory layer. Thus, to control CV1, CV2, CV3 and CV4 we have as manipulated variables MV4, MV5, MV6 and MV7. We follow the pair-close rule, and suggest the following pairings for the regulatory layer:

- Use the cooling water (MV4) to control the condenser pressure (CV2);

- Use the feedwater pump (MV5) to control the drum level (CV1) (only DOF left to control the level)

- Use the economizer bypass (MV6) to control the flue gas temperature (CV3)

- Use the attemperator (MV7) to control the superheated steam tempera-

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ture (CV4) (only DOF available)

- Assume turbine speed is equal to the grid frequency

Note that MV4, MV6 and MV7 are likely to saturate at maximum cooling,
zero bypass and zero bypass, respectively. Fortunately, this is not a problem,
because when we reach one of these constraints, it is optimal to give up control
of the corresponding CV. This happens because the corresponding CV will move
away from its constraints of minimum pressure (CV2), minimum flue gas tem-
perature (CV3) and maximum steam temperature (CV4), respectively. Thus,
no further attention from the supervisory control layer is required when these
saturations happen.

4. Step 6. Control structures for supervisory control

From an optimal operation point of view, we want on a slow time scale to
maximize boiler efficiency (i.e. keep bypass streams closed and let the pressure
float) and minimize throttling losses (i.e. keep all valves close to maximum). On
a short time scale we may need participate in grid frequency control. We can
meet both objective due to their time scale separation, and this requires using
the steam valve (MV2) dynamically, and drive to its nominal opening (e.g. 90
% (Weissbach et al., 2006)) at steady-state.

We assume that all other loops are closed according to the pairing from sec-
tion 3.2.5, and therefore we analyse only the two remaining degrees of freedom:
MV1 (fuel) and MV2 (steam turbine valve). The remaining CVs from Table
3 are the power produced (CV7 = W) and the superheated steam pressure
(CV8 = p). The main issues that we consider concern:

1. pairing, that is what to do with the remaining degrees of freedom, MV1
   and MV2?

2. should the pressure be controlled?

In the following, we show a simplified flowsheet of the steam side, with the
two remaining degrees of freedom: MV1 (fuel) and MV2 (steam valve). The

boiler illustrated symbolizes the economizer and its bypass, drum, superheater and attemperator.

We analyse the case where we want to keep the power produced at its set-point, and we start by presenting the common control structures in industrial steam cycles.

4.1. Standard industrial control structures for control of power and pressure

The standard industrial control structures are boiler driven, turbine driven, floating pressure and its variation, sliding pressure (Klefenz, 1986; Welfonder, 1999; Fachbereich Anwendungsfelder der Autumation, 2003). The objective of this analysis is to understand their steady-state and dynamics characteristics.

4.1.1. Floating pressure operation

In floating pressure operation mode, Fig.5, the superheated steam pressure (CV8) is not controlled, and it is given by the fuel (MV1), according to the energy balance. The power produced can be controlled by manipulating the fuel (MV1), the only DOF with a significant steady-state effect. Floating pressure operation is optimal from an energy point of view because it allows for the steam valve (MV2) to be fully open. When we say that steam valve is opened, it may well be partly open because of the back-off required to participate in droop control (see the discussion in Section 6.3). However, because of the boiler inertia, this operation mode has a slow time constant for controlling the power produced. When the pressure becomes an active constraint (i.e. $p = p^{\min}$ or $p = p^{\max}$), we give-up controlling the power using MV1 (fuel), and use it to control the pressure instead. This is called CV-CV switching, and we can use a *MID* block (i.e. logic to select the middle output of all three controllers). Note that it is more efficient to use MV1 (fuel) directly to control the pressure once it reaches its maximum limit than using MV3 to bypass the steam turbine. Also note that all control structures imply a *MID* selector to keep the pressure within bounds, but this is not shown to simplify the illustrations.
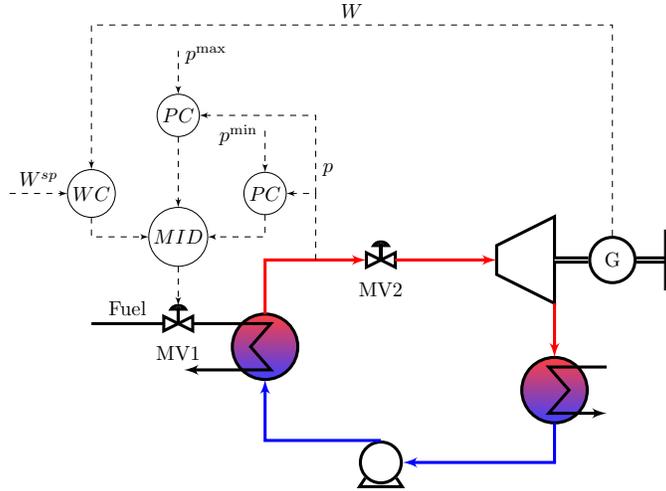
21

Figure 5: Floating pressure operation mode with a MID selector to keep the pressure within bounds ($p^{\min} \leq p \leq p^{\max}$).

4.1.2. Boiler driven operation

In boiler driven operation mode, the power produced is kept at setpoint by manipulating the fuel MV1 (the throughput manipulator in this case), while the superheated steam pressure is kept at constant setpoint using the steam valve MV2, as shown in Fig. 6. For this reason, boiler driven can be considered as an extension of floating pressure. In this case, MV2 can only be used to improve the dynamic response of the cycle, as it has a negligible steady-state effect (see Fig. 4).

4.1.3. Turbine driven operation

Turbine driven is the reverse pairing of boiler driven, i.e. the power produced is controlled using the steam valve MV2 (the throughput manipulator in this case), and the steam pressure is controller using the fuel MV1, as shown in Fig. 7. Its advantage is a faster time response for control of power ($CV7 = W$).

Both turbine and boiler driven have the advantage of utilizing the system's energy storage because of pressure build-up in the drum and superheater. However, compared to floating pressure, there is some loss of energy efficiency.
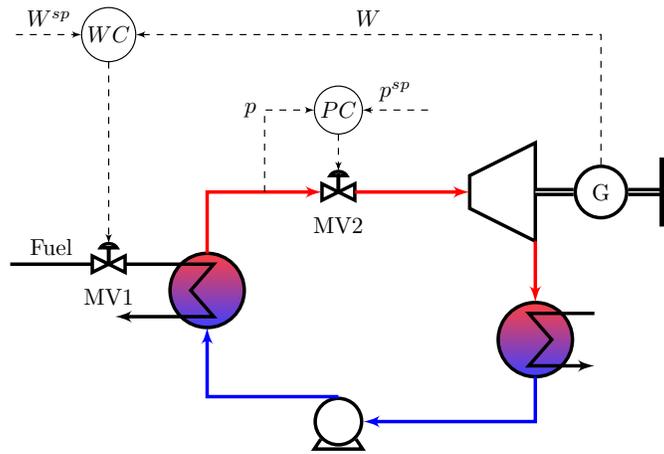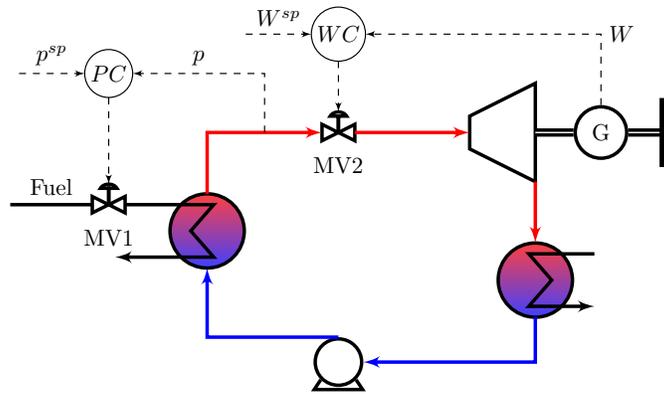
22

Figure 6: Boiler driven operation mode

Figure 7: Turbine driven operation mode

4.1.4. Sliding pressure operation

In practice, power plants operators prefer to control the pressure. This operation mode is a modification of floating pressure, as shown in Fig. 8 (Klefenz, 1986). The sliding pressure curve is pre-defined as function of the produced power (as a simple curve), and the steam mass flow is used to as an indirect measure of the power produced in many control loops. Note that disturbances in boiler and combustion may result in changes in steam mass flow (m), and therefore measuring the steam mass flow rate may give a false indication of the changes in produced power. The pressure setpoint is only changed at steady-state, but not dynamically, during power setpoint changes.
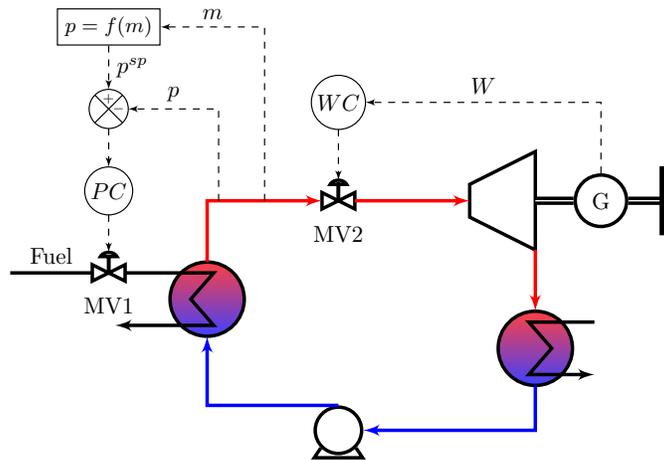

Figure 8: Sliding pressure operation (not as optimal at steady state as floating pressure) (Klefenz, 1986).

Comparison of different pressure operation modes. Fig. 9 illustrates the three pressure operation modes:

- constant (blue line), which is the operation mode for turbine driven and boiler driven. This strategy give fast load changes response.

- sliding (green line), where the pressure is kept constant at high load ($W \geq W^L$) for fast load change response, and it follows the power produced at lower loads ($W \leq W^L$) to increase the boiler efficiency.

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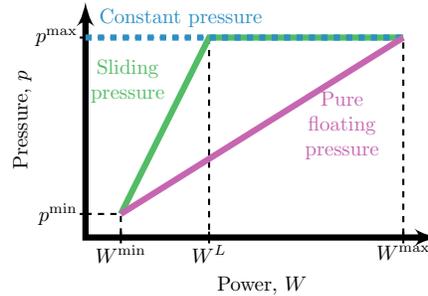- pure floating (mauve line), where the pressure follows the power produced and the throttling losses are minimized.

Figure 9: The three pressure operation modes discussed in this work

4.2. Improved control structures for floating pressure operation

We want to look into dynamic improvements of floating pressure operation. This operation mode is optimal from a steady-state point of view because it is optimal to keep the steam valve (MV2) fully open to minimize throttling loses. However, the dynamic response for controlling produced power (CV7 = W) is rather slow because the throughput manipulator is located at the feed side (MV1). Two alternatives for this are:

1. valve position controller (VPC), Fig. 10

2. parallel control, Fig. 11, using two controllers: a PI-controller for MV1 and P-controller for MV2.

4.2.1. Valve position control

In VPC there is one fast acting MV1 that controls the CV, and one slow MV2 that acts to bring MV1 to its nominal value (Shinskey, 1988). In our case, the fast MV is MV2 (steam valve), and the slow MV is MV1 (fuel), as shown in Fig. 10. Valve position control acting on a valve-turbine system is also described in (Farmer & Lipták, 2006).
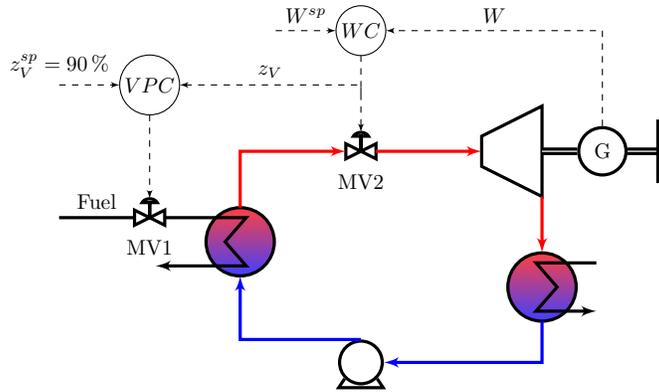
25

Figure 10: Valve position controller (VPC) to improves floating pressure operation

4.2.2. Parallel control

⁵²⁰ Similarly to VPC, this control structure can be used when two MV act on the same CV, but with different time constants (Balchen & Mummé, 1988). Only one of the two controllers can have integral action, otherwise there maybe no unique steady-state solution for the MVs (Åström & Hägglund, 2006). MV1 (fuel) is the only degree of freedom with a significant steady-state effect on the power, and therefore we use a PI-controller for MV1, and P-controller for MV2 (steam valve), as shown in Fig.11. Once the error is zero, the P-controller takes MV2 to its nominal values, which is set as the controller bias.

5. Simulation study: optimal operation of a simple heat-to-power cycle

⁵³⁰ We consider a typical steam cycle for simulating the control structures presented in Section 4:

- Floating pressure (Fig. 5)

- Boiler driven (Fig. 6)

- Turbine driven (Fig. 7)

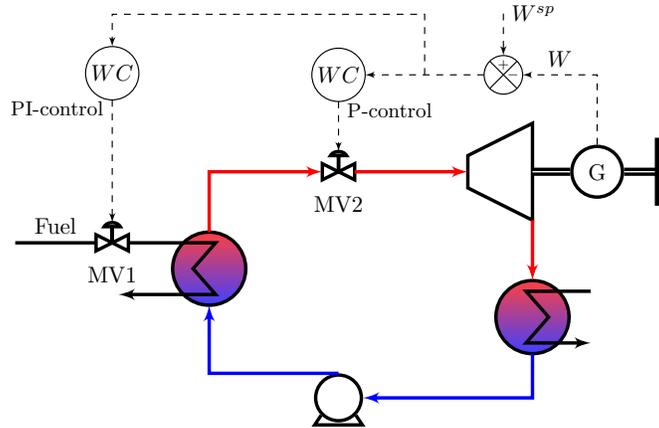⁵³⁵ - Valve position control (Fig. 10)

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Figure 11: Parallel control with one PI-controller and one P-controller to improve floating pressure operation

- Parallel control (Fig. 11)

5.1. Model

A heat-to-power cycle can be decomposed into three subsystems (SS), which can be modelled sequentially (Maffezzoni et al., 1983):

- $SS1$: water cycle
- $SS2$: combustion
- $SS3$: generator and connection to the electric grid.

Steam cycles models with different complexity are presented in the open literature, and a good overview of modelling methods and tools is given by (Alobaid et al., 2017).

For control purposes, simpler models are often used, and the work by (Ordys, A. W.; Pike & Katebi R. M.; Grimble, 1994) outlines simple models for each component of a heat-to-power cycle, which can then be used in a modular simulation approach. A detailed dynamic model that has been extensively used for both modelling and controller synthesis is the drum boiler is presented in the work by (Åström & Bell, 2000). Object oriented approaches have become an attractive alternative for modelling due to their reusability and versatility.

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Modelling and regulatory control design of a subcritical steam cycle using an object oriented language and library is described in the work by (Chen et al., 2017).

With respect to steam turbines performance maps, static laws are commonly used because there is no accumulation in the turbine. The most common is Stodola's law of cones (Cooke, 1985), or constant mass flow coefficient (considering chocking conditions) (Cordes, 1963). Both of these laws related the current operating conditions (i.e. *off-design* conditions) to the design point. In addition to first principle derived relations, empirical linear relations between the power produced and the steam mass flows, called *Willans lines*, are described and used in the work by (Sun & Smith, 2015).

For our propose, the model has to be simple and robust, yet it also needs to capture the main dynamics of the process. We develop a first principle model for a typical steam cycle to test our analysis. As mentioned in Section 3, we consider only the water side subsystem (i.e. SS1). The interface with SS1 is modelled via the hot flue gas inlet temperature, and the interface with SS3 is modelled via the generator frequency.

The model consists of both algebraic mass- and energy balance representing fast time scale processes, as well as dynamic equations representing the longer time scales. Therefore, the model is a system of differential and algebraic equations (DAE).The differential states (x) are the temperatures on the hot side of the heat exchangers (e.g. T_E, T_D and T_S), the superheated steam temperature after the attemperator (T_A), the holdups in the drum (M_D) and superheater (M_S) and the frequency (ω). The algebraic states (w) are the flue gas temperature on the cold side of the heat exchangers (e.g. T_E^g, T_D^g and T_S^g), turbine inlet pressure (p_T), and the produced power (W). The DAE model has a total of 12 states (7 differential and 5 algebraic). The detailed model equations are given in Appendix B.

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Table 4: Nominal operating conditions

Variable	Unit	Value	Unit
Holdup	Economizer	100	kg
	Drum	3000	
	Superheater	100	
	Attemperator	10	
Water temperature	Inlet	45	°C
	Economizer	303	
Steam temperature	Drum	303	°C
	Superheater	595	
	Attemperator	529	
Fuel (combustion temperature)	Economizer	150	°C
	Drum	425	
	Superheater	777	
	Inlet	1000	
Flowrate	Pump	10.6309	$\mathrm{kg\,s^{-1}}$
	Economizer bypass	0	
	Attemperator	0.6309	
	Turbine bypass	0	
	Gas	31.4018	
Power		16.55	MW
Frequency		50	Hz

5.2. Nominal operating conditions

We are interested in optimal operation of existing heat to power cycles, therefore, the equipment design is given, and we must decide how to use it optimally. We consider rezonable values for the nominal operating conditions for a simple steam cycle with one pressure level (this may be typical for an older operating plant). Similar values are found in Skogestad (2008) and Åström & Bell (1987). Table 4 shows the nominal operating conditions. The design parameters are given in Table A.9, and are computed by solving the model at steady-state for the nominal conditions (Table 4).

5.3. Controller

Eq. 2 shows the time domain expression for the PI-controllers used. Note that we consider the saturation limits for the applied input u^p (i.e. a valve cannot be more than fully open of close, or mass flows cannot be negative.), and

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therefore antiwindup is implemented. We use the back-calculation antiwindup method, where the controller output tracks the input applied to the process (u^p) with a time constant (τ_T) equal to the integral time (τ_I)(Åström & Hägglund, 2006).

$$u(t) = u^0 + K_C e(t) + \frac{K_C}{\tau_I} \int_0^t e(t)dt + \frac{1}{\tau_T} \int_0^t e^u(t)dt \tag{2a}$$

$$e = y^{sp} - y \tag{2b}$$

$$e^u = u^p - u \tag{2c}$$

$$u^p = \min(u^{\max}, \max(u, u^{\min})) \tag{2d}$$

5.3.1. Controller tuning

We find the controllers tuning parameters (proportional gain K_C and integral time τ_I) by identifying a first-order plus time-delay (FOPTD) model ($\frac{k}{\tau s+1}e^{-\theta s}$) or integrating model $\left(g(s) = \frac{k'}{s} \right)$ from a step response in the input u, followed by applying the SIMC tuning rules (Skogestad, 2003) with a chosen closed loop time constant τ_C.

For a first-order model, we use Eq. 3.

$$K_C = \frac{1}{k} \frac{\tau}{\tau_C + \theta} \tag{3a}$$

$$\tau_I = \min(\tau_0, 4(\tau_C + \theta)) \tag{3b}$$

where, k is the steady-state gain from u to y, τ is the open loop time constant, τ_C is the closed loop time constant and θ is the time delay.

For integrating process (i.e. for plants with large time constant such as levels), we use Eq. 4.

$$K_C = \frac{1}{k'} \frac{1}{\tau_C + \theta} \tag{4a}$$

$$\tau_I = 4(\tau_C + \theta) \tag{4b}$$

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where, k' is the slope, τ is the open loop time constant, τ_C is the closed loop
time constant and θ is the time delay.

5.4. Step 5. Regulatory controller design

5.4.1. Controller tuning

We begin with tuning the controllers for the regulatory layer (i.e. level
controller and active constraints).

An secondary decision in decentralized control, is the order of tuning the
PI controllers. This is an important decision in highly coupled processes, and
we base our decision based on effective time delays in the process (Skogestad,
2003). In our case, we use a sequential tuning method, that is we first tune
the level controller, then close the loop, tune the next controller and repeat the
procedure. Table 5 gives the tuning parameters for the drum level control (M_D),
superheated steam controller (T_A), and flue gas outlet temperature controller
(T_E^g). Note that we do not need to tune the condenser pressure controller as
we consider it constant, i.e. perfect control. The value for the closed loop time
constant τ_C is taken quite large to account for any unmodelled capacities and
holdups, and make the model time scale more realistic.

Table 5: Level and temperature controllers tuning

Type	Loop	τ_C [s]	K_C	τ_I [s]
LC	MV5-CV1	10	0.1	40
TC	MV6-CV3	20	0.05	10
TC	MV7-CV4	15	-0.0008	1

5.5. Step 6. Supervisory controller design

We proceed with the supervisory control design and we tune the controllers
using the fuel MV1 and the steam valve MV2 for the structures presented in
Section 4. In designing the supervisory control layer, we keep the same tuning
for the regulatory layer (Table 5), and follow the same tuning procedure. Is
important to note that we use the initial response in tuning all controllers for
MV2 (steam valve), as we are interested in using it on a fast time scale (see Fig.

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4 and Section 3.2.2). For boiler driven, we close first the pressure loop. Then, the open loop response from fuel MV1 to power CV7 has one left-hand-plane

₆₃₅ zero and one left-hand-plane pole, and we use a pure I-controller tuned based on the initial response.

Table 6: Standard industrial controllers tuning

	Floating pressure		Boiler driven		Turbine driven	
MV-CV	MV1-CV7	MV2=0.9	MV1-CV7	MV2-CV8	MV1-CV8	MV2-CV7
τ_C s	30	N/A	30	5	15	5
K_C	0.0028	N/A	0	-1.48	1.1574	0.0004
τ_I	40	N/A	0.1	20	50	1

Table 7: Parallel controllers tuning

	VPC		PI and P control	
MV-CV	MV1-MV2	MV2-CV7	MV1(PI)-CV7	MV2(P)-CV7
τ_C s	50	5	30	5
K_C	-2.84	0.0004	0.0041	0.0004
τ_I	115	1	55	N/A

5.6. Step 7. Control structure performance

We test the control structures for setpoint changes and disturbance rejection.

5.6.1. Setpoint changes

₆₄₀ Fig. 12a shows the response for the power, and Fig. 12b shows the pressure response, while Fig. 12c and Fig. 12d show the input usage for fuel MV1 and steam valve MV2 respectively to a 10% step decrease followed by 10 % step increase in the power setpoint for all five control structures described in Section 4.

5.6.2. Disturbance rejection

₆₄₅ Fig. 13a shows the response for the power, and Fig. 12b shows the pressure response, while Fig. 13a and Fig. 13d show the input usage for fuel MV1 and steam valve MV2 respectively to a disturbance of 50 °C step increase in
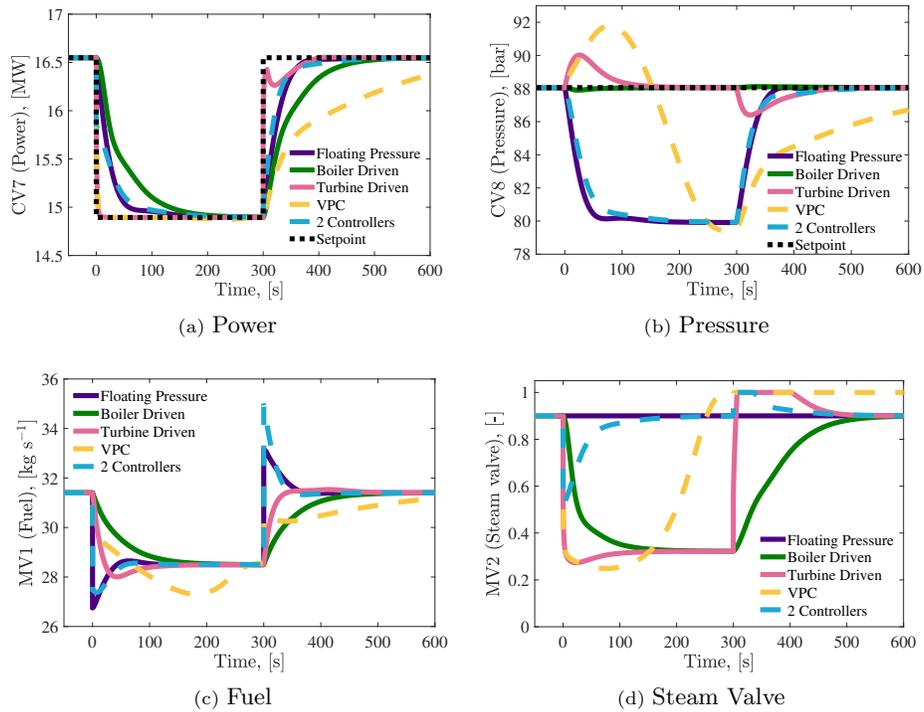
(a) Power

(b) Pressure

(c) Fuel

(d) Steam Valve

Figure 12: Closed loop responses to 10 % step decrease at time $t = 0\,$s followed by 10 % step increase at time $t = 300\,$s in the power setpoint. Only turbine driven and boiler driven have pressure control.
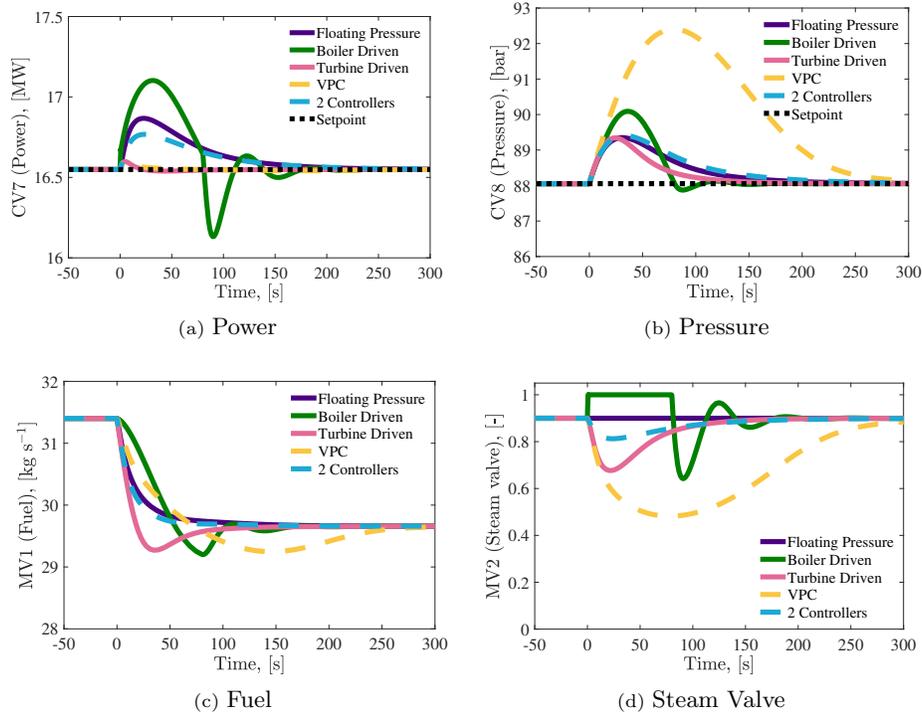
Figure 13: Closed loop responses to a step change of $50\,^{\circ}\mathrm{C}$ in T_0^g at time $t = 0\,\mathrm{s}$. Only turbine driven and boiler driven have pressure control.

the combustion temperature for all five control structures from Section 4. This high change in temperature could be for example given by changes in the fuel composition or heat quality.

5.6.3. Summary of comparison of the five control structures

Comparing the three common industrial standards (floating pressure, boiler driven, turbine driven) in Fig. 12, boiler driven structure reacts slower for change in the produced power, but has the advantage that the fuel rate (MV1) changes smoothly, and it does not overshoot as for all other control structures. Turbine driven gives the fastest response to a step change in power (CV7), yet, in comparison to boiler driven, the pressure (CV8) drifts significantly from the setpoint in transient operation. These effects can be explained considering the smaller effective time delay from controlling CV7 or CV8 using the steam valve

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MV2, contrast to using the fuel MV1. The VPC control structure has as similar response to turbine driven for a power setpoint decrease, while is is the slowest to a setpoint increase because the steam valve MV2 saturates. In addition, by design, the VPC is tuned slow, and tuning it faster would result in an aggressive controller with a high input usage for fuel MV1. In terms of performance the two controllers (parallel control) seems very good and has the benefit of floating pressure at steady state.) Considering throttling losses for the steam valve MV2, both boiler and turbine driven results in higher losses because MV2 needs to closes more to keep the setpoint for power (CV7), compared to the other control structures that do not have pressure control. To answer the question if the pressure should be controlled (Section 4), we can say that controlling the pressure gives a faster response when steam valve is used to control the power, while letting the pressure float minimizes the throttling losses.

The response for a disturbance in the combustion temperature (T_0^g) shows that the boiler driven control structure may not be suited for plants with large variations in this disturbances. An increase in T_0^g increases the enthalpy of the hot flue gases, which results in more heat transferred in the boiler, and an increase in the steam pressure (CV8). To decrease the pressure to its setpoint, the steam valve MV2 has to open (Fig. 13d), which results in a higher overshoot for the power produced (Fig. 13a) compared to the other control structures. Moreover, in this particular example, the steam valve (MV2) saturates, and we loose control of the pressure during transient operation. Note that the pressure response for boiler driven follows the floating pressure initially (Fig. 13b), but then it decreases faster because the steam valve MV2 is fully open for boiler driven, while for floating pressure is kept at 90 % opening.

6. Discussion

6.1. Throttling losses

Having the steam valve partly open results in throttling losses. We define throttling as a reduction in pressure without removal of energy in form of heat

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₆₉₀ or work, i.e. isenthalpic process. Throttling is irreversible and it translates into increase of entropy and thereby exergy losses and decrease of available work (Shinskey, 1978).

(a) Enthalpy-Entropy diagram

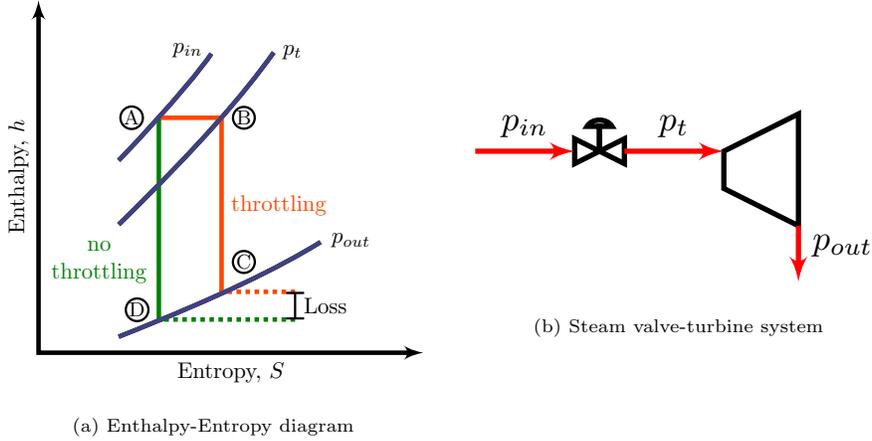(b) Steam valve-turbine system

Figure 14: Enthalpy-Entropy diagram for an expansion process with and without throttling, left, and, a steam valve-turbine system indicating the corresponding pressures, right.

Figure 14 shows the enthalpy-entropy diagram for an expansion process with and without throttling, where the purple lines represent the lines of constant

₆₉₅ pressure. The green path from A to D represents the isentropic expansion from inlet pressure p_{in} to outlet pressure p_{out} without throttling. The orange path from A to B represents throttling from inlet pressure p_{in} to pressure p_t. The orange path from B to C represents the isentropic expansion from the pressure after throttling p_t to outlet pressure p_{out}.

₇₀₀ Considering constant outlet pressure p_{out}, with throttling, steam is expanded at a higher enthalpy, thus resulting in increase of entropy and loss of available work. The loss in available work is graphically represented by the difference in enthalpy between D and C.

Mathematically, the loss of available work is quantified by Eq. 5.

$$W_{\text{Loss}} = \text{v}\Delta p \tag{5}$$

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where, W_{Loss} = loss in available work, v is the volumetric flow (assumed constant), and $\Delta p = p_{in} - p_t$ is the pressure drop across the steam valve.

6.2. Floating pressure efficiency

The throttling losses mentioned above are relevant if we could replace the valve by an adjustable small turbine. However, this is not the case here. Instead, we consider keeping the steam valve fully open and let the pressure float, leading to a lower steam drum pressure at low loads. This does not in itself give an increased efficiency in terms of power produced because it does not require more energy to increase the pressure. However, by lowering the pressure and thus the temperature in the drum on the steam side, we get improved temperature driving forces. Thus, with floating pressure operation we are able to extract more energy from the fuel because we get a lower the flue gas exit temperature. In some cases, we are not allowed to lower the flue gas exit temperature because of corrosion issues, and then there will be no efficiency benefit of floating pressure operation. In Table 8, we analyse the new steady-state operation conditions for a decrease in power by 10 % from nominal (i.e. at 90 % load), both for floating and constant pressure. We assume that the exit flue gas temperature (CV3=T_E^g) is not controlled, that is, the economizer bypass MV6 is kept closed. We assume that the turbine efficiency is equal in all cases.

Table 8: Comparison of the steady-state values for floating and constant pressure operation modes at 90 % load without controlling the cold flue gas temperature.

Pressure	Power (MW)	Fuel (MV1) $(\mathrm{kg\,s^{-1}})$	Pressure (CV8) (bar)	T_E^g (CV3) (°C)	T_S^s (°C)	Attemperator (MV7) $(\mathrm{kg\,s^{-1}})$
Floating	14.9	27.79	79.9	128.4	611.4	0.7045
Constant	14.9	27.86	88.05	130.7	616.5	0.74

At 90 % load, the loss in used fuel for constant pressure is only 0.2%, and the additional pump work needed to boost the pressure by 8 bar accounts to 0.05% of the produced power, which adds to only 0.25%. If the temperature of the cold flue gas is controlled at its minimum limit (i.e. CV3= $T_E^g \geq T_E^{g,\min}$), then the loss in efficiency is reduced to 0.05% (i.e. account for the pump), both

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floating and constant pressure operation modes At 65 % load, the energy efficiency loss for constant pressure operation increases to 1 % (without flue gas
730 temperature control). Therefore, the energy efficiency increases at low loads in floating pressure operation, though the increase is not significant. These numbers depend naturally on the process design, especially how the heat exchange area is distributed between the economizer, drum and superheater.

6.3. Steam turbine control

735 For a stand-alone turbine, or when a gear box is used to connect the turbine and the generator, the turbine rotational speed may be used as a degree of freedom, but we are here considering a turbine connected to the grid without a gearbox. More precisely, the turbine is connected to an electric generator through a shaft and the electric generator is connected to the grid. In principle,
740 no control of the turbine is needed, because inertia and self-regulation will imply that all these frequencies (turbine speed ω_T, generator speed and grid frequency ω) are the same at steady state. However, in practice, speed (frequency) control is needed for two reasons:

1. Local level (speed control of turbine). To protect the turbine/ generator
745 system from damage caused by fast changes in the turbine speed, we must keep the turbine frequency close to the grid frequency on a fast time scale. This is done by installing a steam valve upstream the turbine (MV2) which controls CV5 $= \omega_t - \omega$.

2. Grid level (droop control of grid frequency). The grid frequency ω should
750 be kept close to its desired setpoint ω^{sp} (e.g., at 50 Hz in Europe and 60 Hz in the US). The value of ω is directly proportional to the amount of kinetic energy (inertia) stored in all the rotating equipment in the grid. Any imbalance between power production and power demand will therefore change ω. There is a certain self-regulation in the power demand, but this
755 is not enough. Thus, to maintain a desired grid frequency ω^{sp} in spite of variations in the power demand, some of the main power producers must

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participate in controlling ω. That is, we need to control CV6 = $\omega - \omega^{sp}$. The manipulated variables for this is the power production for each unit i (W_i), which at steady state requires manipulating the fuel rates (MV1$_i$). This control task is divided into primary (droop), secondary and tertiary grid frequency control.

The local level turbine speed control (CV5) is always present (Kurth & Welfonder, 2006), (Fachbereich Anwendungsfelder der Autumation, 2003). As mentioned, the inherent self-regulation will keep CV5=0 at steady-state. Thus, integral action is not needed for control of CV5, so in practice a proportional controller (droop) is used. We will not discuss the control of CV5 in this paper, because it is generally considered a part of the equipment protection, and is not available for control engineers. Furthermore, because the self-regulation of CV5 is fast anyway, the design of this controller will not affect the rest of the control system.

Next consider grid frequency control. Not all power producers participate in grid frequency control, but the ones that do usually get a higher power price. Let the power production (actually, the setpoint for power production) from each producer be written as $W_i^{sp} = W_{i,0}^{sp} + \Delta W_i^{sp}$ where ΔW_i^{sp} comes from the primary frequency control (proportional droop) and W_{i0}^{sp} from the secondary frequency control. Fig. 15 shows the primary and secondary control loops for plant i in an isolated area with N power plants participating in grid regulation. Note that the inner turbine control loop are not explicitly shown, but this is inside the *Power plant i* block.

Let us first consider the primary droop control which takes place on a fast time scale.

Ideally, we want to avoid centralized coordination of the participating power producers at the fast time scale. The solution is then that each producer has local control of the grid frequency, CV6. However, these local controllers cannot have integral action, because otherwise there is no unique steady state, and one may even get into cases where the controllers fight each other, possibly resulting

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in one power plant closing down and another reaching full capacity (Cohn, 1984), (Åström & Hägglund, 2006).

To solve this issue, we use proportional control of CV6= $\omega - \omega^{sp}$. This gives a unique steady state, where the power change from each producer i is uniquely given by the change in grid frequency, $\Delta W_i^{sp} = -1/R_i(\omega^{sp} - \omega)$. Here $1/R_i$ is the proportional controller gain, typically between 3 and 10 %/%, where R_i is the steady-state process gain from power to frequency. The MV available for achieving the desired change in power production (ΔW_i) is as mentioned the fuel (MV1), but to speed up the dynamic response one frequently makes use of the steam valve (MV2). The required response time is usually specified in the contract for each producer. Note that the steady-state effect of MV2 on the power production is negligible (Fig. 4).

Next consider the secondary frequency control which involves a centralized controller with integral action. Integral action is needed because the proportional action in droop control results in a steady-state offset in frequency. This controller changes the bias $W_{i,0}^{sp}$ in the power setpoint for each producer (adjusted with a gain α_i) on a slow time scale. Finally, for larger changes in power demand on a longer time scale, it may be necessary to start up or close down power production (tertiary frequency control).

When a plant participates in droop control, the fuel (MV1) has to be lower that its maximum, which gives a loss in power production. Furthermore, for fast response to changes in power demands, the steam valve (MV2) has to be partly closed (e.g. 90 % opening) at nominal operation, which gives a loss in efficiency. These issues explain why producers who participate in droop control get a higher electricity price.

6.4. Operation with given fuel rate (MV1)

In this case, MV1 must be used to control the TPM. Hence, from a steady-state point of view, we have no degrees of freedom left to control the power produced, and the steam cycle becomes a "swing power producer" (Fig. 16). In this case, the power plant clearly cannot participate in grid frequency control.
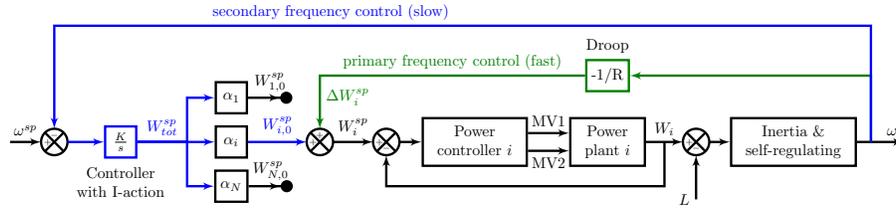
Figure 15: Primary (green) and secondary (blue) frequency control for power plant i in an area with N power plants participating in grid frequency control (adapted from (Wood et al., 2014).)

More importantly, there must be some other means (not shown) to make the power output balance the fuel rate by controlling the steam pressure. For example, there could be another steam flow added to the turbine inlet or excess steam could be withdrawn and used for other purposes. In addition, to make sure that the pressure is kept within bounds, it is suggested (Fig. 16) to give up controlling the fuel rate when a pressure constraint is reached (i.e. $p = p^{\max}$ or $p = p^{\min}$).
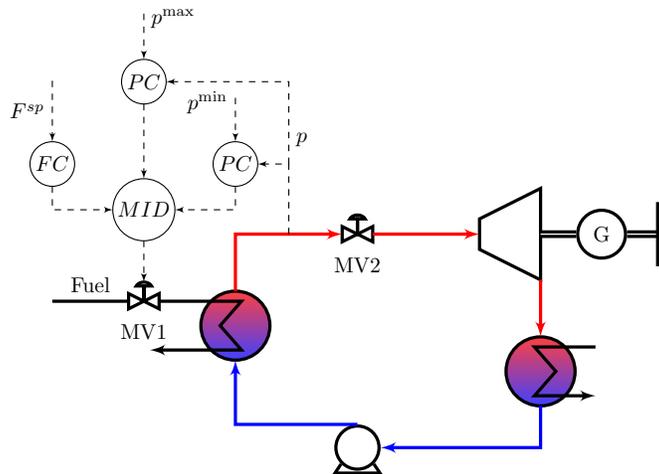
Figure 16: Control structure with given fuel rate (MV1) and fully open steam valve (MV2) to maximize power production. The mid-selector gives up controlling the fuel rate when a pressure constraint becomes active. Under normal conditions, pressure is controlled by some other manipulated variable (not shown), for example, by producing steam.

6.5. Influence of level control time constant

Similarly to floating pressure, the drum level can be let to float between it's minimum and maximum limits, to utilize the stored energy in the hot water. This can be achieved with a slow level control, in which case, the drum level would be allowed to decrease to a lower level and the steam flow would be longer sustained when it is required to produce more power, while feedwater is slowly pumped into the system. Note that with tight (fast) level control, cold feedwater is pumped rapidly in the system which decreases the drum temperature faster in transient operation. However, for the studied simulation case, a slow level control showed limited improvement of the dynamic performance. Fig. 17 shows the power response for setpoint changes (10% decrease at time $t = 0$s and 10% increase at time $t = 300$s) for floating pressure (Fig. 17a), boiler driven (Fig. 17b), VPC (Fig. 17c) and two controllers (Fig. 17d). Turbine driven has an insignificant change. Note that the larger the closed loop time constant (τ_C), the slower the drum level control is.

7. Conclusions and final remarks

In this work, we used the systematic framework of plantwide control to analyse the control and optimal operation of a simple steam cycle with one pressure level, drum and condensing turbine.

After controlling the unstable inventory (drum level CV1), and the active constraints: condenser pressure CV2, superheated steam temperature CV4, cold flue has temperature CV3, we have two degrees of freedom left: the fuel MV1 and the steam valve MV2. MV2 only has a dynamic effect on the power produced, as shown in the response to setpoint changes in Fig. 12 and in the disturbance rejection response in Fig. 13.

At low loads, letting the pressure float is slightly more efficient. By keeping the pressure constant the dynamic performance is improved, especially for the turbine driven operation (Figs. 12 and 13).
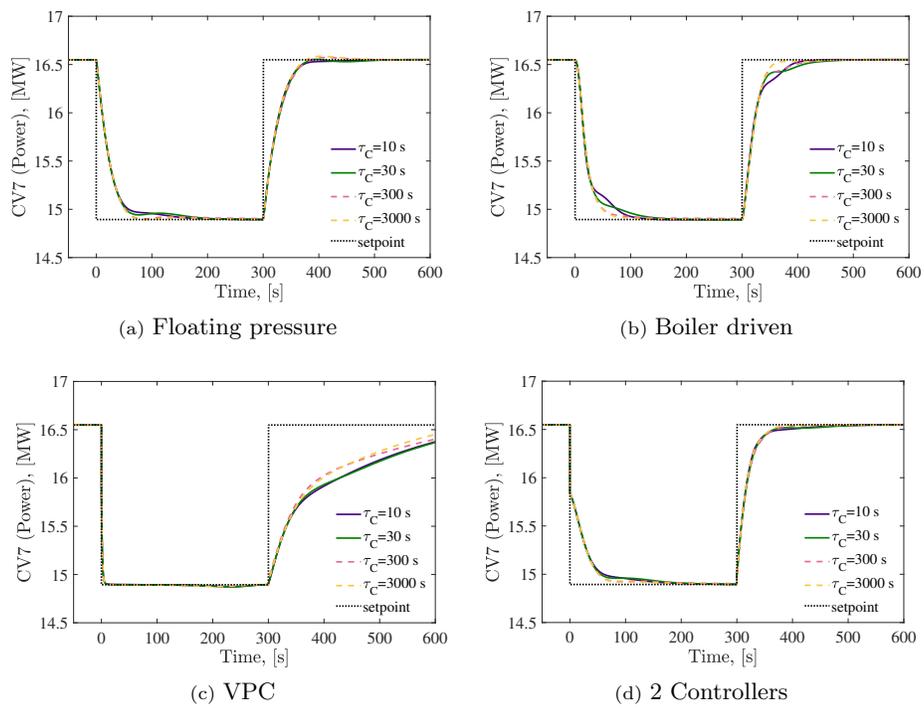
Figure 17: Influence of the tuning of the drum level controller on the power response to setpoint changes.

Of interest for future work is a more comprehensive analysis of the control implications of variables heat sources with varying inlet temperature. The extend the existing to which the storage capacity of the process can be utilized as a short-time buffer between supply and demand, should also be further investigated.

Acknowledgment

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Appendix A. Model design parameters

Table A.9: Design parameters

Parameter	Value	Unit
UA_E	95.12	$\mathrm{kW\,{}^{\circ}C^{-1}}$
UA_D	46.4	$\mathrm{kW\,{}^{\circ}C^{-1}}$
UA_S	19.94	$\mathrm{kW\,{}^{\circ}C^{-1}}$
$C_{v,D}$	10	$\mathrm{kg\,{}^{\circ}C^{-1}}$
K_v	2.32	$\mathrm{kg\,bar^{-1}}$
ϕ_D	3.625	$\mathrm{kg\,s^{-1}\sqrt{K}bar^{-1}}$
M_0	7.56	$\mathrm{pu\ MW\ pu\ rad^{-1}\,s^{-1}}$
D_0	2	$\mathrm{pu\ MW\ pu\ rad^{-1}\,s^{-1}}$

Appendix B. Detailed Model

Appendix B.1. Thermodynamics

Assumptions

(A1) Constant specific heat for each fluid (water, steam and flue gas);

(A2) The reference temperature is $T^{ref} = 0$ °C;

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(A3) The boiling reference temperature is $T^{refB} = 576$ °C (drum nominal
temperature);

(A4) Ideal gas behaviour for steam;

(A5) Saturated steam pressure follows Antoine equation (Eq. B.2b).

Considering a constant c_p, the specific enthalpy for the gas, water and steam
has a linear dependency on the temperature, as shown in Eq. B.1.

$$\Delta H^g = c_p^g(T_j^g - T^{ref}) \quad \forall j \in (i, S, D, E) \tag{B.1a}$$

$$\Delta H^w = c_p^w(T_j - T^{ref}) \quad \forall j \in (P, E) \tag{B.1b}$$

$$\Delta H^s = c_p^w(T^b - T^{ref}) + c_p^s(T_j - T^b) + \Delta H^v(T^b) \tag{B.1c}$$

$$\forall j \in (D, S, A, T)$$

Table B.10 shows the specific heat for each component.

Table B.10: Specific heat

Component	c_p	Unit
water	4.18	$\mathrm{kJ\,kg^{-1}\,°C^{-1}}$
steam	3	$\mathrm{kJ\,kg^{-1}\,°C^{-1}}$
hot flue gas	1.25	$\mathrm{kJ\,kg^{-1}\,°C^{-1}}$

The saturation pressure in the drum is computed using Antoine relation (Eq.
B.2b) as a function of the temperature.

$$p_D = 10^\alpha \tag{B.2a}$$

$$\alpha = A - \frac{B}{T_D + C} \tag{B.2b}$$

where T is in K and p_D is in bar, and the constants are $A = 5.11564\,B = 1687.537\,C = 42.98$.(Reid et al., 1987)

Appendix B.2. Economizer and bypass

Assumptions

49

(A6) Constant inlet temperature (due to tight condenser pressure control,see below);

985 (A7) Constant water holdup ($\rightarrow$ neglect the mass balance);

(A8) Static mixing for the bypass and economizer outlet streams (i.e. fast heat and mass dynamics and negligible holdup)($\rightarrow$ static mass and energy balances);

We write a dynamic energy balance in temperature form for the steam side, and an algebraic energy balance for the gas side.
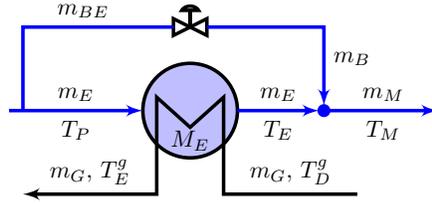

Figure B.18: Economizer

990

$$\frac{dT_E}{dt} = \frac{1}{M_E}\left(m_E(T_P - T_E) + \frac{Q_E}{c_p^w}\right) \tag{B.3a}$$

$$0 = m^g c_p^g (T_D^g - T_E^g) - Q_E \tag{B.3b}$$

$$Q_E = UA_E\left(\frac{T_D^g + T_E^g}{2} - \frac{T_D + T_E}{2}\right) \tag{B.3c}$$

$$T_M = \frac{m_E T_E + m_{BE} T_P}{m_M} \tag{B.3d}$$

Appendix B.3. Mass flowrates

The flowrate for the pump, economizer bypass and attemperator are directly given by (PI)-controllers (we assume fast inner cascade controllers on the valve position), according to the general Eq. B.4. Antiwindup with a tracking time

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constant equal to the integral time ($\tau_T = \tau_I$) is used.

$$m_i = m_i^0 + K_{C,i}e_i + \frac{K_{C,i}}{\tau_I}\int_0^t e_i(t)dt + \frac{1}{\tau_{T,i}}\int_0^t e_{m,i}(t)dt \qquad \text{(B.4a)}$$

$$e_i = y_i^{sp} - y_i \qquad \text{(B.4b)}$$

$$e_{m,i} = m_{i,p} - m_i \qquad \text{(B.4c)}$$

$$m_{i,p} = \max(m_i, 0) \qquad \text{(B.4d)}$$

$i \in (P, AE, BE)$ and $y \in (M_D, T^g, T_S)$

The remaining flowrates are computed from steady-state mass balances, according to Eq. B.5.

$$m_E = m_P - m_{AE} - m_{BE} \qquad \text{(B.5a)}$$

$$m_D = C_{V,D}(p_D - p_S) \qquad \text{(B.5b)}$$

$$m_S = m_V - m_{AE} \qquad \text{(B.5c)}$$

$$m_V = z_V K_C(p_S - p_T) \qquad \text{(B.5d)}$$

Appendix B.4. Drum

Assumptions

(A9) Perfect mixing;

(A10) Equal temperature in liquid and vapour phases;

(A11) Negligible vapour holdup (compared to the liquid holdup);

(A12) Saturated steam;

(A13) Outlet flow is given by a linear valve (fully open) equation as a function of the pressure drop;

(A14) Fixed vaporization in the drum, i.e. the drum inlet is saturated liquid water, and the outlet is saturated vapour. This means that the vaporization is know *a-priori*. Note that fixing the vaporization point may not be optimal

for operation, as the heat transfer area is not optimally utilized. However, a variable phase transition point raises additional modelling challenges, which we want to avoid;

For the drum, we write a dynamic mass (Eq.B.6a) and energy balance on temperature form on the steam side (Eq.B.6b), and algebraic energy balance on the gas side(Eq.B.6c).
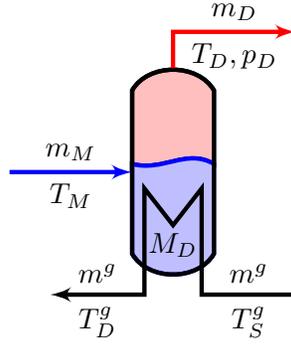

Figure B.19: Drum

$$\frac{dM_D}{dt} = m_M - m_D \tag{B.6a}$$

$$\frac{dT_D}{dt} = \frac{1}{M_D c_p^w}(m_M(H_M - c_p^s T_D) + \dots \tag{B.6b}$$

$$- m_D(H_D - c_p^s T_D) + Q_D)$$

$$0 = m^g c_p^g(T_D^g - T_S^g) - Q_D \tag{B.6c}$$

$$Q_D = U A_D \left(\frac{T_S^g + T_D^g}{2} - \frac{T_M + T_D}{2}\right) \tag{B.6d}$$

Appendix B.5. Superheater and attemperator

1010   *Assumptions*

(A15) The steam holdup accounts for the entire steam holdup in the cycle ($\rightarrow$ need to consider a dynamic mass balance);

(A16) Static mixing in the attemperator (i.e. fast heat and mass dynamics and negligible holdup) ($\rightarrow$ static mass and energy balance);

We write a dynamic mass (Eq.B.7a) and energy balance on temperature form on the steam side (Eq.B.7b), and algebraic energy balance on the gas side(Eq.B.7c).
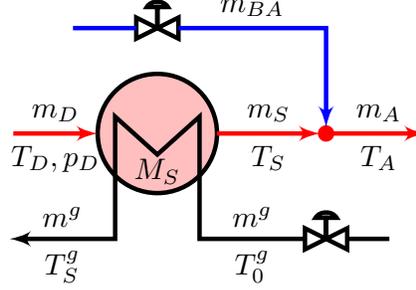

Figure B.20: Superheater and attemperator

$$\frac{dM_S}{dt} = m_D - m_S \tag{B.7a}$$

$$\frac{dT_S}{dt} = \frac{1}{M_S c_p^s}(m_D(H_D - c_p^s T_S) + \dots \tag{B.7b}$$

$$- m_S(H_S - c_p^s T_S) + Q_S)$$

$$0 = m^g c_p^g (T_0^g - T_S^g) - Q_S \tag{B.7c}$$

$$Q_S = U A_S \left(\frac{T_0^g + T_S^g}{2} - \frac{T_S + T_D}{2}\right) \tag{B.7d}$$

$$0 = m_s H_s + m_A H_P - m_A H_A \tag{B.7e}$$

1015 *Appendix B.6. Steam valve, turbine and generator*

Assumptions. Steam turbine valve

(A17) Linear valve equation and pressure drop;

(A18) Isenthalpic;

(A19) Neglijable holdup;

1020 *Assumptions. Turbine*

(A20) Turbine map: constant mass flow coefficient ($\phi = \frac{m\sqrt{T}}{p}$);

(A21) Isentropic expanssion with 100% efficiency;

(A22) Speed is given by generator frequency;

(A23) Neglijable holdup;

Assumptions. Generator

(A24) Another power plant is responsible for keeping the frequency at the nominal value, therefore we can only use a P-controller for frequency control;

Appendix B.7. Condenser

Assumptions

(A25) Tight pressure control, i.e. constant condenser pressure ($\rightarrow$ is not modelled and the cycle is open);
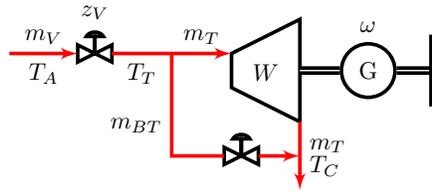

Figure B.21: Steam turbine

$$m_T = m_V - m_{BT} \tag{B.8a}$$

$$m_{BT} = 0 \tag{B.8b}$$

$$0 = \frac{m_T \sqrt{T_A}}{p_T} - \phi_d \tag{B.8c}$$

$$T_C = T_T \left(\frac{p_C}{p_T} \right)^{(R \backslash c_p^s)} \tag{B.8d}$$

$$0 = W + m_T c_p^s (T_T - T_C) \tag{B.8e}$$

$$\frac{d\omega}{dt} = \frac{1}{M_g} \left(P - L - D_g(\omega - \omega_0) \right) \tag{B.8f}$$

Appendix B.8. General for for heat exchangers

Assumptions

(A26) Constant and negligible holdup for the hot side;

(A27) Constant UA (heat transfer coefficient U $(W/(m^2 K)$ times heat surface area A (m^2);

(A28) Temperature difference (ΔT) is the difference between the algebraic mean on each side.

(A29) Neglected wall capacity