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 identified 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 hot flue gas (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 a heat-to-power cycle.

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

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Nomenclature

Controller

\( \tau_I \) Integral time

CV Controlled variable

\( K_c \) Proportional gain

LC Level (holdup) controller

mid Logic block that select the middle value of all its inputs

MV Manipulated variable

N/A Not applicable

PC Pressure controller

PID proportional-integral-derivative controller

TC Temperature controller

VPC Valve position controller

WC Power controller

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

ref  Reference

s  Steam (usually dropped)
Variables

\( \omega \)  Grid frequency
H  Specific enthalpy
h  Enthalpy
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 have evolved over the years based on industrial practices to a stage where it becomes less trivial to understand what are the optimal operational objectives, constraints

\(^1\)Thermal power plants, steam cycles and heat-to-power cycles are used interchangeably.
or degrees of freedom. Moreover, their transfer to new cases or newcomers in the field may not be straightforward.

Often plant operators take old 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. Usually in these cases, optimal operation of the steam cycles if 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.

Therefore, the objective of this work is twofold, and consists of a steady-state, and a dynamic analysis. The first is to examine the optimal operation and control problem for a heat-to-power cycle and provide a clear and systematic procedure for identifying the operational objective, specification or constraints and degrees of freedom from a steady-state point of view.

We accomplish these objectives in the framework of plantwide control, which handles control structures decisions for the entire plant. The goal is to find a control strategy, preferably a simple on, 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).

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 [1] presents plantwide control analysis for the combustion side of oxy-fired circulating fluidized bed boilers. The work by [2] use 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 advantage of using a systematic plantwide control procedure is that it
might reveal new control policies that might have been overlooked for existing processes [3].

There are several books presenting the control loops in a typical heat-to-power cycle. To understand the current industrial policies and their dynamic characteristics, we build on the control strategies described in the work by [4], [5] and [6].

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 optimisation (days), local optimisation (hours), supervisory control (minutes) and regulatory control (seconds), as shown in Fig. 1. The top layers are responsible for production planing 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, and gives the
setpoints for the lower layers [7].

To systematically design each layer, we use the plantwide control procedure proposed by [7]. 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:

**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).

**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 backoff 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 of one has MV saturation in this layer. Considering the large number of control loops in a typical plant, simple PID-controllers are used for the regulatory layer.

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2 variables that give acceptable loss when kept at constant setpoint.
The objectives of the supervisory (advanced) control layer are:

1. Achieve the economic objectives given by the upper optimisation 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.

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

The step 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. implement Model Predictive Control) or decentralized control (i.e. implement advanced control structures with simple logic block to handle changes in active constraints [8].)

**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. In this process, hot flue gas may be produced by burning waste or fuel. Thermal energy from the hot flue gases 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 Fig. 2, the dotted line represents the boundary of the analysed system, i.e. the steam side.

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 exchanging heat with hot flue gas (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 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 requir-
Figure 3: Flowsheet of a steam cycle with a drum boiler, one pressure level, and condensing turbine. After closing the basic regulatory loops for temperatures, pressure and level, MV1 and MV2 are the two remaining degrees of freedom (See also Table 1) and Section 3.2.4). Liquid water is in blue, vapor in red.

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3.2. Top-down analysis

We proceed by applying the top-down analysis to the described process.

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 objective are:

1. Produce the energy as
   (a) power to the electric grid at the required voltage and frequency (usually large power plants with condensing turbines, i.e. $> 100 \text{MW}$);
   (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:

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

Here, $W \ [\text{J s}^{-1}]$ is the produced power, $S \ [\text{kg s}^{-1}]$ is the produced steam (= 0 in this paper), $F \ [\text{J s}^{-1}]$ is the hot flue gases (energy source), $U \ [\text{kg s}^{-1}]$ is the utility consumption, and $p \ [\$/\text{kg}^{-1}]$ or $[\$/\text{J}^{-1}]$ is the price of each. There may be additional terms, for example several feed energy sources or several steam products. We analyze an operating plant and therefore, capital costs, personal, and maintenance 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 operational objectives listed above for a steam cycle include $[2]$:

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

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 maximize turbine work, but within boundaries to prevent large thermal gradients (i.e. $T_s^A = 529 \degree C$).

C5 Keep the superheated steam pressure below a maximum value to avoid high thermal and mechanical stress and to extend the operating life (i.e. $p \leq p_{max} = 220$ bar).

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

C7 Keep the temperature ($T_{gE}$) of the flue gas outlet below environmental limits, and above dew point to prevent corrosion $\degree C$ ($T_{gE} \geq 150 \degree C$). 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 Maximize cooling ($MV4 = MV4_{max}$) to bring the condenser pressure at lower limit to maximize the pressure ratio in the turbine (i.e. $p_C = 0.1$ bar).

C9 Keep the turbine speed at the setpoint ($n = 50$ 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.

In addition to 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 [1] for an analysis on the combustion side for an oxy-fired circulating fluidized bed boiler.

3.2.2. Step 2. Degrees of freedom

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 decide yet on the pairing, and number of the MV and CV are not corresponding
(i.e. MV1 is not necessary used to control CV1.) in the next sections.

### Table 1: Manipulated variables

<table>
<thead>
<tr>
<th>Manipulated variable</th>
<th>Comments, analysis</th>
</tr>
</thead>
<tbody>
<tr>
<td>MV1: Hot flue gas</td>
<td>• At steady-state the power produced can only be changed by manipulating (MV1).</td>
</tr>
<tr>
<td></td>
<td>• The turbine valve should ideally be fully open to minimize throttling losses.</td>
</tr>
<tr>
<td></td>
<td>• It has mainly a dynamic effect, as the steady-state effect on produced power is insignificant.</td>
</tr>
<tr>
<td></td>
<td>• It can improve the dynamic response for power.</td>
</tr>
<tr>
<td></td>
<td>• 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.</td>
</tr>
<tr>
<td>MV2: Steam valve</td>
<td>• Normally closed, needed when the energy in the feed is larger than power demand.</td>
</tr>
<tr>
<td></td>
<td>• Used to avoid too high pressure, i.e. if ( p \geq 220 \text{ bar} ), MV3 opens to reduce the pressure.</td>
</tr>
<tr>
<td>MV3: Turbine bypass</td>
<td>Open at MV4=( MV4^{\text{max}} ) to minimize condenser pressure (( p_C )).</td>
</tr>
<tr>
<td>MV4: Cooling water</td>
<td>• Use only to control the drum level.</td>
</tr>
<tr>
<td></td>
<td>• Cannot be used to control the steam pressure, as it has no steady-state effect (see Table 3).</td>
</tr>
<tr>
<td>MV5: Feedwater pump</td>
<td>• Use only if ( T_g &lt; T_g^{\text{min}} ).</td>
</tr>
<tr>
<td>MV6: Economizer bypass</td>
<td>• Use only if ( T_s &gt; T_s^{\text{max}} ).</td>
</tr>
<tr>
<td>MV7: Attemperator</td>
<td>• Use only if ( T_g^{\text{max}} ).</td>
</tr>
</tbody>
</table>

**Steam turbine steady-state effect.** The open loop response for the superheated steam pressure (\( p \)), and power produced (\( W \)) from opening the steam valve (\( z_V \)) by 10% is shown in Fig. 4. Note that the pressure (\( p \)) and power (\( W \)) are normalized with respect to their initial conditions. Let’s explain the open loop response from physical considerations. Consider the linear valve \( \dot{m} = z K_V \Delta P \) where, \( \dot{m} \) is the mass flow rate, \( z \) is the valve opening, \( K_V \) is the valve coefficient and \( \Delta P \) is the pressure drop across the valve. Increasing \( z \) causes a fast increase of \( \dot{m} \), which results in a decrease of the pressure inventory before the valve. The latter results in a smaller \( \Delta P \), which results in a decrease of \( \dot{m} \) after its initial increase. To increase \( \dot{m} \) at steady-state, \( \Delta P \) has to increase, and this can only be achieved by increasing MV1.
Economizer bypass. Bypass streams are used as additional degrees of freedom for operation when both the cold and hot flows are fixed \[10\]. In this process, the cold side stream (i.e. feedwater flow) is given by the drum level controller, while the hot side stream is given by the load demand (at steady-state, see details in Table 1). Typical steam cycles operate with a reverse bypass (i.e. the cold side stream is recycled at the heat exchanger inlet) \[11\], while direct bypass (i.e in direction of the flow) is used in chemical plants. Reverse bypasses is not recommended because it require a pump, thus making it less energy efficient than direct bypass. In steam cycles, the role of the recycle is to increase the temperature of the feedwater and prevent gas condensation on the pipes and reduce corrosion. Note that condensation happens on the cold pipe wall, which has a temperature closer to the water temperature due to higher heat transfer coefficient on the water side \[11\].

3.2.3. Step. Disturbances

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

<table>
<thead>
<tr>
<th>Disturbance variable</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>DV1: Hot flue gas temperature</td>
<td>Typically for waste heat</td>
</tr>
<tr>
<td>DV2: Hot flue gas specific heat</td>
<td>Typically for waste heat</td>
</tr>
<tr>
<td>DV3: Grid frequency (Load)</td>
<td>Consumers increasing their demand (load) or producers decreasing their production</td>
</tr>
<tr>
<td>DV4: Required power setpoint</td>
<td>Typically for power plants required to participate in secondary or tertiary grid frequency regulation</td>
</tr>
<tr>
<td>DV5: Cooling water temperature</td>
<td></td>
</tr>
</tbody>
</table>

3.2.4. Step 2. Active Constraints

Active constraints (AC) are variables that should be kept at their limiting value for optimality. To determine which constraints will be active, we can 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 MV3, MV6 and 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 imply a CV-MV switch, and it can handled by single loop PID-controllers without additional logic.

The active constraints are:

(AC1) MV3=0 (MV constraint);

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

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

(AC4) MV7=0 (MV constraint) or \(T_A = T_A^{\text{max}}\) (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\textsuperscript{max}) results in \( p_C = p_C^{\text{min}} \), closing economizer bypass stream (MV6=0) gives minimum temperature \( T_E = T_E^{\text{min}} \), and closing the attemperation stream (MV7 = 0) gives maximum \( T_A = T_A^{s,\text{max}} \). When \( p_C < p_C^{\text{min}} \), we give-up MV4=MV4\textsuperscript{max} and use MV4 to increase \( p_C \). When \( T_A > T_A^{s,\text{max}} \) we give-up MV6=0 and use MV7 to decrease \( T_E \). When \( T_E > T_E^{s,\text{max}} \) we give-up MV7=0, and use MV7 to increase \( T_E \). 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).

<table>
<thead>
<tr>
<th>Controlled variable</th>
<th>Comments, analysis</th>
</tr>
</thead>
<tbody>
<tr>
<td>CV1: Drum level (( M_D ))</td>
<td>• Levels are unstable inventories and it needs to be controlled</td>
</tr>
<tr>
<td>CV2: Condenser pressure (( p_C ))</td>
<td>• No steady-state effect</td>
</tr>
<tr>
<td>CV3: Cold flue gas temperature (( T_g ))</td>
<td>• see Section 3.2.4</td>
</tr>
<tr>
<td>CV4: Superheated steam temperature (( T_s ))</td>
<td>• see Section 3.2.4</td>
</tr>
<tr>
<td>CV5: Turbine speed</td>
<td>• Active constraint for all operation regions.</td>
</tr>
<tr>
<td>CV6: Grid frequency</td>
<td>• Imposed by grid stability</td>
</tr>
<tr>
<td>CV7: Power produced</td>
<td>• Only for plants required to participate in grid frequency regulation</td>
</tr>
<tr>
<td>CV8: Steam pressure</td>
<td>• Given by MV1 according to the boiler energy balance</td>
</tr>
</tbody>
</table>

• 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) depends on the main control objective of the power plant (see Section 3.2.1), and we can identify two main cases (i.e. economic modes) [7]:

Case I The throughput manipulator is the produced power, i.e. the plant participates in grid frequency regulation, or generates steam for a chemical process). Optimal operation implies maximizing the fuel and boiler efficiency.

Case II The throughput manipulator is the hot flue gas energy source, MV1), and optimal operation implies maximizing production (i.e. power or steam). This case typically applies to low load boilers, usually with cheap fuels, e.g. waste gasses, biomass or solar.

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. 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=MD) and leave the feedwater tank level uncontrolled. The steady-state value of MD does not matter, except that it contributes to energy storage, which has dynamic implications. Next, the steam pressure (CV8) is often controlled because it may be drifting and control of it may contribute to more stable and predictable operation. However, 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). Condenser pressure (CV2) 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) and cold flue gas temperature (CV3). 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, 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 [8]).

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

- Use MV4 to control the condenser pressure (CV2);
- Use MV5 to control the drum level (CV1) (only DOF left to control the level)
- Use MV6 to control the flue gas temperature (CV3)
- Use MV7 to control the superheated steam temperature (CV4) (only DOF available)
- Assume turbine speed is equal to the grid frequency

Note that MV4, MV6 and MV7 are likely to saturate at max 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 temperature (CV3) and maximum steam temperature (CV4), respectively. Thus, no further attention from the supervisory control layer is required when these saturations happen.

3.3.2. Steam turbine control

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, no control of the turbine is needed, because inertia and self-regulation will imply that all these frequencies (turbine speed \( \omega_T \), generator speed and grid frequency \( \omega \)) 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 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 \( CV_5 = \omega_t - \omega \).

2. Grid level (droop control of grid frequency). The grid frequency \( \omega \) should be kept close to its desired setpoint \( \omega^{sp} \) (e.g., at 50 Hz in Europe and 60 Hz in the US). The value of \( \omega \) 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 \( \omega \). There is a certain self-regulation in the power demand, but this is not enough. Thus, to maintain a desired grid frequency \( \omega^{sp} \) in spite of variations in the power demand, some of the main power producers must participate in controlling \( \omega \). That is, we need to control \( CV_6 = \omega - \omega^{sp} \). 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 [12], [6]. 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

\[
W_{\text{sp}}^i = W_{\text{sp}}^{i,0} + \Delta W_{\text{sp}}^i
\]

where \( \Delta W_{\text{sp}}^i \) comes from the primary frequency control (proportional droop) and \( W_{\text{sp}}^{i,0} \) from the secondary frequency control. Fig. 5 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 in one power plant closing down and another reaching full capacity [13], [14].

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
Figure 5: Primary (green) and secondary (blue) frequency control for power plant $i$ in an area with $N$ power plants participating in grid frequency control.

Given by the change in grid frequency, $\Delta W_{sp}^i = -1/R_i(\omega_{sp} - \omega)$. Here $1/R_i$ is the proportional controller gain, typically between 3 and 10 %/%, where $R_i$ is the process gain from power to frequency. The MV available for achieving the desired change in power production ($\Delta W_i$) is as mentioned the fuel rate (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 $\alpha_i$) 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 rate (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.
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 to participate in grid frequency control. We can meet both objectives due to their time scale separation, and this requires using MV2 dynamically, and drive to its nominal opening (e.g. 90% \(^{15}\)) at steady-state.

We assume that all other loops are closed according to the pairing from section 3.2.5 and therefore we analyze only the two remaining degrees of freedom: MV1 (hot flue gas) and MV2 (steam turbine). 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 and MV2. The boiler illustrated symbolizes the economizer and its bypass, drum, superheater and attemperator.

We analyze first case I, and we start by presenting the common control structures in industrial steam cycles.

4.1. Case I - 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 \(^{17}\). 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, the superheated steam pressure is not controlled, and it is given by MV1, according to the energy balance. The power produced (throughput manipulator) can be controlled by manipulating 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. Due to 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. \( p = p_{\text{min}} \) or \( p = p_{\text{max}} \)), we give-up controlling the power using MV1, and use it to control the pressure instead. This is called CV-CV switching, and we can use a \textit{MID} block (i.e. logic to selects the middle output of all three controllers). Note that it is more efficient to use MV1 directly to control the pressure once it reaches its maximum limit is more efficient than using MV3. Also note that all control structures imply a \textit{MID} selector to keep the pressure within bounds, but this is not shown to simplify the illustrations.

![Figure 6: Floating pressure operation mode with a MID selector to keep the pressure within bounds](image)

4.1.2. Boiler driven operation

In boiler driven operation mode, the power produced is kept at setpoint by manipulating MV1 (hence the name boiler driven), while the superheated
steam pressure is kept at constant setpoint using MV2, as shown in Fig. 7. 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).

**Figure 7: Boiler driven operation mode**

### 4.1.3. Turbine driven operation

Turbine driven is the reverse pairing of boiler driven, i.e. the power produced is controlled using MV2 and the steam pressure is controller using MV1, as shown in Fig. 8. Its advantage is a faster time response for control of power $W$.

**Figure 8: Turbine driven operation mode**

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, the fuel and boiler are utilized less efficiently due to throttling losses.
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. 9. The setpoint is model-based given by the measured steam mass flowrate. This is not as optimal as floating pressure as we rely on accurate flow measurement \((m)\), and a good model.

![Figure 9: Sliding pressure operation (not as optimal as floating pressure)](image)

4.2. Improved floating pressure operation using the steam valve \(MV2\)

We want to look into dynamic improvements of floating pressure operation. This operation mode is optimal from a steady-state point of view, but as mentioned, the dynamic response for using \(MV1\) to control the produced power is too slow. However, we may use \(MV2\) to improve the dynamic response for produced power. Two alternatives for this are:

1. valve position controller (VPC), Fig. 10
2. parallel control, Fig. 11 using a PI-controller for \(MV1\) and P-controller for \(MV2\).

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. In our case, the fast MV is \(MV2\) (steam valve), and the slow MV is \(MV1\) (hot flue gas), as shown in Fig. 10.
4.3. Parallel control

With two different controllers, we can have only one controller with integral action. MV1 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, as shown in Fig.11.

4.4. Case II - the heat input is the throughput manipulator

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". The pressure must still be kept within bounds, and the control solution is to give-up the TPM when a pressure constraint is reached (i.e. $p = p^{\text{max}}$ or $p = p^{\text{min}}$) as shown in Fig. 12. In this case, there are other active plants that support grid frequency control. If power control is required, we need to introduce additional operational degrees of freedom to act as a buffer between demand and supply. This could
be, for example, feed storage tanks, or steam accumulators. Note that for some processes and to some extend, we could use the existing storage capacities of the process, i.e. the drum, superheater, or steel.

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

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

5.1. Model

A heat-to-power cycle can be decomposed into three subsystems (SS), which can be modelled sequentially [17]:

- **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 [18].

For control purposes, simpler models are often used, and the work by [19] outlines simple models for each components 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 [20]. Objected oriented approaches have become an attractive alternative for modelling due to their reusability and versatility. Modelling and regulatory control design of a subcritical steam cycle using an objected oriented language and library is described in the work by [21].

With respect to steam turbines performance maps, static laws are commonly used. The most common is Stodola’s law of cones [22], or constant mass flow coefficient (considering chocking conditions) [24]. Both of these laws related the current operating conditions (called 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 [24].

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.

5.1.1. Model

The model consists of both algebraic mass- and energy balance representing small 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 ($\omega$). The algebraic states ($w$) are the flue gas temperature on the cold side of the heat exchangers (e.g. $T_{gE}^g$, $T_{gD}^g$ and $T_{gS}^g$), turbine inlet pressure ($p_T$), and the produced power ($W$). The DAE model has a total of 12 states (seven differential and five algebraic). The detailed model equations are shown in Appendix B.
Table 4: Nominal operating conditions

<table>
<thead>
<tr>
<th>Variable</th>
<th>Unit</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Holdup</td>
<td>Economizer</td>
<td>100</td>
<td>kg</td>
</tr>
<tr>
<td>Drum</td>
<td></td>
<td>3000</td>
<td></td>
</tr>
<tr>
<td>Superheater</td>
<td></td>
<td>100</td>
<td></td>
</tr>
<tr>
<td>Attemperator</td>
<td></td>
<td>10</td>
<td></td>
</tr>
<tr>
<td>Water temperature</td>
<td>Inlet</td>
<td>318</td>
<td>K</td>
</tr>
<tr>
<td>Economizer</td>
<td></td>
<td>576</td>
<td></td>
</tr>
<tr>
<td>Drum</td>
<td></td>
<td>576</td>
<td></td>
</tr>
<tr>
<td>Superheater</td>
<td></td>
<td>868</td>
<td></td>
</tr>
<tr>
<td>Attemperator</td>
<td></td>
<td>802</td>
<td></td>
</tr>
<tr>
<td>Gas temperature</td>
<td>Economizer</td>
<td>423</td>
<td>K</td>
</tr>
<tr>
<td>Drum</td>
<td></td>
<td>698</td>
<td></td>
</tr>
<tr>
<td>Superheater</td>
<td></td>
<td>105</td>
<td></td>
</tr>
<tr>
<td>Inlet</td>
<td></td>
<td>1273</td>
<td></td>
</tr>
<tr>
<td>Flowrate</td>
<td>Pump</td>
<td>10.6309</td>
<td>kg s$^{-1}$</td>
</tr>
<tr>
<td>Economizer bypass</td>
<td></td>
<td>0</td>
<td></td>
</tr>
<tr>
<td>Attemperator</td>
<td></td>
<td>0.6309</td>
<td></td>
</tr>
<tr>
<td>Turbine bypass</td>
<td></td>
<td>0</td>
<td></td>
</tr>
<tr>
<td>Gas</td>
<td></td>
<td>31.4018</td>
<td></td>
</tr>
<tr>
<td>Power</td>
<td></td>
<td>16.55</td>
<td>MW</td>
</tr>
<tr>
<td>Frequency</td>
<td></td>
<td>50</td>
<td>Hz</td>
</tr>
</tbody>
</table>

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 reasonable values for the nominal operating conditions for a simple heat to power cycle with one pressure level (this may be typical for an older operating plant). Similar values are found in [25] and [26]. Table 4 shows the nominal operating conditions. The design parameters are given in Table A.8 and are computed by solving the model at steady-state for the nominal conditions (Table 4).

5.3. Step 5. Regulatory controller design

5.3.1. 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 [27]. In our case, we first tune the level controller, then close the loop, tune the next controller and repeat the procedure. We find the controllers tuning parameters (proportional gain $K_C$ and integral time $\tau_I$) by identifying a FOPTD model

$$\frac{k\exp(-\theta s)}{\tau s + 1}$$

from a step response in the input, followed by applying the SIMC tuning rules [27] with a chosen closed loop time constant $\tau_C$. Table 5 gives the tuning parameters for the drum level control ($M_D$), superheated steam controller ($T_{\text{SA}}$), and flue gas outlet controller ($T_{\text{EG}}$). 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 $\tau_C$ is taken quite large to account for any unmodelled capacities and holdups, and make the model time scale more realistic.

<table>
<thead>
<tr>
<th>Type</th>
<th>Loop</th>
<th>$\tau_C$ [s]</th>
<th>$K_p$</th>
<th>$\tau_I$ [s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>LC</td>
<td>MV5 – CV1</td>
<td>10</td>
<td>0.1</td>
<td>40</td>
</tr>
<tr>
<td>TC</td>
<td>MV6 – CV3</td>
<td>20</td>
<td>0.05</td>
<td>10</td>
</tr>
<tr>
<td>TC</td>
<td>MV7 – CV4</td>
<td>15</td>
<td>-0.0008</td>
<td>1</td>
</tr>
</tbody>
</table>

5.4. Step 6. Supervisory controller design

We proceed with the supervisory control design and we tune the controllers using MV1 and MV2 for the structures presented in Section 3. In design the supervisory controllers, 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, as we are interested in using MVs on a fast time scale (see Fig. 4 and Section 3.2.2).

5.5. Step 7. Control structure testing

5.6. Setpoint changes

Figure 13 shows the response for the power, while Fig. 15 and Fig. 16 show the input usage for MV1 and MV2 respectively to a 1% step decrease in the
Table 6: Standard industrial controllers tuning

<table>
<thead>
<tr>
<th></th>
<th>Floating pressure</th>
<th>Boiler driven</th>
<th>Turbine driven</th>
</tr>
</thead>
<tbody>
<tr>
<td>MV-CV MV1-P MV2=0.9</td>
<td>45</td>
<td>30</td>
<td>15</td>
</tr>
<tr>
<td>$\tau_C$</td>
<td>N/A</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>$K_C$</td>
<td>0.0019</td>
<td>0.0028</td>
<td>1.1574</td>
</tr>
<tr>
<td>$\tau_I$</td>
<td>40</td>
<td>20</td>
<td>50</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>0.0004</td>
</tr>
</tbody>
</table>

Table 7: Parallel controllers tuning

<table>
<thead>
<tr>
<th></th>
<th>VPC</th>
<th>PI and P control</th>
</tr>
</thead>
<tbody>
<tr>
<td>MV-CV MV1-MV2 MV2-P</td>
<td>50</td>
<td>25</td>
</tr>
<tr>
<td>$\tau_C$</td>
<td>-2.84</td>
<td>0.0004</td>
</tr>
<tr>
<td>$K_C$</td>
<td>0.0004</td>
<td>0.0049</td>
</tr>
<tr>
<td>$\tau_I$</td>
<td>115</td>
<td>55</td>
</tr>
<tr>
<td></td>
<td></td>
<td>N/A</td>
</tr>
</tbody>
</table>

power setpoint for all five control structures from Section 4.

5.7. Disturbance rejection

Figure 17 shows the response for the power, while Fig. 17 and Fig. 20 show the input usage for MV1 and MV2 respectively to a disturbance of 10 K step increase in the hot flue gas inlet temperature for all five control structures from Section 4.

6. Discussion

Comparing the common industrial standards (Fig. 13, 15, 16), it is clear that using MV2 significantly improves the dynamic response for the power produced to a step change in power setpoint. The floating pressure control structures react slower compared to boiler driven or turbine driven. 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 MV2, contrast to using MV1. The VPC control structure is the slowest to reach the setpoint. By design, the VPC is tuned slow, and tuning it faster would result in an aggressive controller with a high input usage for MV1.
Figure 13: Power response to 1% step decrease in the power setpoint

Figure 14: Pressure response to 1% step decrease in the power setpoint. Only turbine driven and boiler driven have pressure control.
Figure 15: MV1 usage to a 1% step decrease in the power setpoint

Figure 16: MV2 usage to a 1% step decrease in the power setpoint
Figure 17: Power response to a step change of 10 K in $T_0^g$

Figure 18: Pressure response to a step change of 10 K in $T_0^g$. Only turbine driven and boiler driven have pressure control.
Figure 19: MV1 usage to a step change of 10 K in $T_0^g$

Figure 20: MV2 usage to a step change of 10 K in $T_0^g$
Considering throttling losses for MV2, both boiler and turbine driven results in higher losses because MV2 needs to close more to keep the setpoint for 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 for power, while letting the pressure float minimizes the throttling losses.

The response for a disturbance in the hot flue gas temperature ($T_g^0$) shows that the boiler driven control structure may not be suited for plants with large variations in this disturbance. An increase in $T_g^0$ increases the enthalpy of the hot flue gases, which results in more heat transferred in the boiler, and an increase in the steam pressure (CV7). To decrease the pressure to its setpoint, MV2 has to open (Fig. 20), which results in a higher overshoot for the power produced (Fig. 17) compared to the other control structures.

In addition to the simulations presented, we investigated the possibility of using the drum storage by having a level controller, that is three times slower (i.e. $\tau_C = 30$ sec alternative to $\tau_C = 10$ sec). However, this showed negligible improvement in the dynamic response for power for both setpoint changes and disturbances.

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 two degrees of freedom left: MV1 and MV2. MV2 only has a dynamic effect on the power produced, as shown in the response to setpoint changes in Fig. 13 and in the disturbance rejection response in Fig. 17.

Of interest for future work is a more comprehensive analysis of the control implications of variables heat sources with varying inlet temperature. The ex-
tend 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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References


Appendix A. Model design parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$UA_E$</td>
<td>95.12</td>
<td>kW K$^{-1}$</td>
</tr>
<tr>
<td>$UA_D$</td>
<td>46.4</td>
<td>kW K$^{-1}$</td>
</tr>
<tr>
<td>$UA_S$</td>
<td>19.94</td>
<td>kW K$^{-1}$</td>
</tr>
<tr>
<td>$C_{v,D}$</td>
<td>10</td>
<td>kg bar$^{-1}$</td>
</tr>
<tr>
<td>$K_v$</td>
<td>2.32</td>
<td>kg bar$^{-1}$</td>
</tr>
<tr>
<td>$\phi_D$</td>
<td>3.625</td>
<td>kg s$^{-1}$  $\sqrt{Kbar}^{-1}$</td>
</tr>
<tr>
<td>$M_0$</td>
<td>7.56</td>
<td>pu MW pu rad$^{-1}$ s$^{-1}$</td>
</tr>
<tr>
<td>$D_0$</td>
<td>2</td>
<td>pu MW pu rad$^{-1}$ s$^{-1}$</td>
</tr>
</tbody>
</table>

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 $T_{ref} = 0$ K;
The boiling reference temperature is $T_{refB} = 576$ K (drum nominal temperature);

Ideal gas behaviour for steam;

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.

$$
\Delta H_g = c_g^p (T_j - T_{ref}) \quad \forall j \in (i, S, D, E) \\
\Delta H_w = c_w^p (T_j - T_{ref}) \quad \forall j \in (P, E) \\
\Delta H_s = c_w^p (T_b - T_{ref}) + c_s^p (T_j - T_b) + \Delta H^v(T_b) \quad \forall j \in (D, S, A, T) 
$$

Table B.9 shows the specific heat for each component.

<table>
<thead>
<tr>
<th>Component</th>
<th>$c_p$</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>water</td>
<td>4.18</td>
<td>kJ kg$^{-1}$ K$^{-1}$</td>
</tr>
<tr>
<td>steam</td>
<td>3</td>
<td>kJ kg$^{-1}$ K$^{-1}$</td>
</tr>
<tr>
<td>hot flue gas</td>
<td>1.25</td>
<td>kJ kg$^{-1}$ K$^{-1}$</td>
</tr>
</tbody>
</table>

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

$$
p_D = 10^\alpha \\
\alpha = A - \frac{B}{T_D + C} 
$$

where $T$ is in K and $p_D$ is in bar, and the constants are $A = 5.11564$ $B = 1687.537$ $C = 42.98$.

Appendix B.2. Economizer and bypass

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

(A7) Constant water holdup (→ neglect the mass balance);

(A8) Static mixing for the bypass and economizer outlet streams (i.e. fast heat and mass dynamics and negligible holdup) (→ 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.21: Economizer

\[
\begin{align*}
\frac{dT_E}{dt} &= m_E(T_P - T_E) + \frac{Q_E}{M_E c_p \cdot w} \quad \text{(B.3a)} \\
0 &= m^g c_p (T^g_D - T^g_E) - Q_E \quad \text{(B.3b)} \\
Q_E &= U A_E \left( \frac{T^g_D + T^g_E}{2} - \frac{T_D + T_E}{2} \right) \quad \text{(B.3c)} \\
T_M &= \frac{m_E T_E + m_{BE} T_P}{m_M} \quad \text{(B.3d)}
\end{align*}
\]

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

\[
\begin{align*}
m_i &= m^0_i + K_{C,i} e + K_{I,i} \int_0^t e(t) dt \quad \text{(B.4a)} \\
e_i &= y_{i,p} - y_i \quad \text{(B.4b)}
\end{align*}
\]
\( i \in (P, AE, BE) \) and \( y \in (M_D, T^y, T_S) \)

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

\[
\begin{align*}
    m_E &= m_P - m_{AE} - m_{BE} \\
    m_D &= C_{V,D}(p_D - p_S) \\
    m_S &= m_V - m_{AE} \\
    m_V &= z_V K_C (P_S - P_T)
\end{align*}
\] (B.5a) (B.5b) (B.5c) (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).
\[
\frac{dM_D}{dt} = m_M - m_D \quad \text{(B.6a)}
\]

\[
\frac{dT_D}{dt} = \frac{1}{M_D c_w^g}(m_M(H_M - c_s^p T_D) + \ldots \quad \text{(B.6b)}
\]

\[- m_D(H_D - c_s^p T_D) + Q_D \]

\[0 = m^g c_p^g(T_D^g - T_S^g) - Q_D \quad \text{(B.6c)}
\]

\[Q_D = UA_D \left(\frac{T_S^g + T_D^g}{2} - \frac{T_M + T_D}{2}\right) \quad \text{(B.6d)}
\]

**Appendix B.5. Superheater and attemperator**

**Assumptions**

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

(A16) Static mixing in the attemperator (i.e. fast heat and mass dynamics and negligible holdup) (→ 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).
Appendix B.6. Steam valve, turbine and generator

Assumptions. Steam turbine valve

(A17) Linear valve equation and pressure drop;

(A18) Isenthalpic;

(A19) Negligible holdup;

Assumptions. Turbine

(A20) Turbine map: constant mass flow coefficient ($\phi = \frac{mvT}{P}$);

(A21) Isentropic expansion with 100% efficiency;

(A22) Speed is given by generator frequency;

(A23) Negligible holdup;

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

**Assumptions. Condenser**

Tight pressure control, i.e. constant condenser pressure (→ is not modelled and the cycle is open);

\[ m_T = m_V - m_{BT} \]  
\[ m_{BT} = 0 \]

\[ 0 = \frac{m_T \sqrt{T_A}}{p_T} - \phi_d \]  
\[ T_C = T_T \left( \frac{p_C}{p_T} \right)^{R/c_p} \]

\[ 0 = W + m_T c_p (T_T - T_C) \]

\[ \frac{d\omega}{dt} = \frac{1}{M_g} (P - L - D_g (\omega - \omega_0)) \]

**General for heat exchangers**

Constant and negligible holdup for the hot side;

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

Temperature difference (\( \Delta T \)) is the difference between the algebraic mean on each side.

Neglected wall capacity