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

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Abstract

The present control solutions for power plants[1] have been developed to a level where it is difficult to make significant performance improvements using more advanced and systematic control policies, such as model predictive control, unless one has a clear definition of the overall control problem. Hence, the objective of this work is to make a first step in the direction of systematically defining the optimal operation and control problem for power plants. We use a systematic plantwide control framework to analyze a simple heat-to-power steam cycle with a single pressure level and a drum boiler. We evaluate two typical economic modes: I) given plant load, and II) given heat input. We determine that after fulfilling controlling the active constraints there are two unconstrained degrees of freedom left that can be used to achieve optimal operation.

Keywords: Plantwide control, power plant operation, power plant control, steam cycle

1. Introduction

The objective of this work is to provide a new perspective on optimal operation and control of heat-to-power cycles by viewing it from the more general setting of plantwide control. To the best of the authors knowledge, this procedure has so far mainly been applied to chemical plants, and it is yet to be applied to the steam side of heat-to-power cycles. Power plant control systems have been developed by industrial practices over many years, and it is not obvious what are the actual specifications and degrees of freedom. Moreover, power plants contribute to the electric grid stability, which may have priority over optimal operation of the plant itself.

In this paper we analyze this systematically, in the framework of plantwide control, and the goal is to find a control policy, preferably a simple one, which stabilizes the plant on a short time scale (regulatory control), and achieves near-optimal operation on a longer time scale (supervisory control).

The rest of the paper is organized as follows: in Section 2 we present the plantwide procedure; followed by applying it to the water side of a power plant in Section 3; presenting the resulted control structures and operation modes in Section 4 and making the final remarks in Section 5.

[1]In this work, the terms heat-to-power cycles and power plants are used interchangeably.

2. Plantwide control procedure

The typical control hierarchy in a process plant is illustrated in Figure 1. It is decomposed based on time scale separation into several simpler layers: scheduling (weeks), site-wide optimisation (days), local optimisation (hours), supervisory control (minutes) and regulatory control (seconds). The layers communicate by sending the process measurements upwards (i.e from the lower to the upper layers), while the setpoints for each layer are given by solving an optimisation problem in the upper layer (Skogestad, 2004). The procedure consists of a top-down analysis concerning optimal steady-state operation, and a bottom-up analysis targeting the lower regulatory control layer structure. In this work, we focus on the steady-state top-down analysis which involves the following steps (Skogestad, 2004):

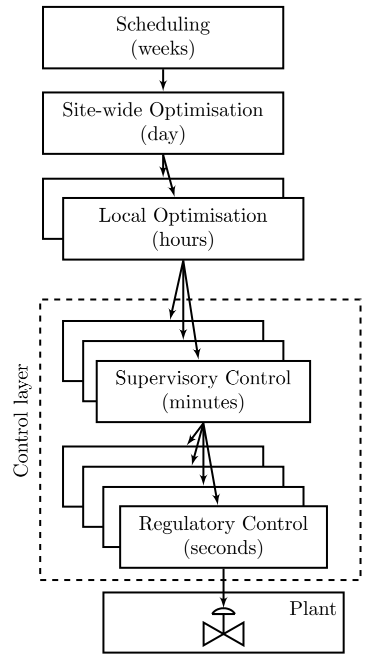


Figure 1: Typical control hierarchy in a process plant.

Step 1 Define the optimal economic operation (cost function J and constraints).

Step 2 Identify the steady-state control degrees of freedom (DOF) (i.e. manipulated variables that are the setpoints for the lower layers). Use a steady-state model to find the optimal operation for the expected disturbances.

Step 3 Based on the previous step, select the control variables (CVs). Always control the active constraints, and decide to which process variables to allocate the remaining DOF.

Step 4 Select the location of the throughput manipulator (TPM), i.e. choose where to set the production rate. This is a dynamic issue, but with economic implications.

3. Plantwide control applied to a simple heat to power cycle

3.1. Process Description

We consider the steam side of a heat-to-power cycle with a drum boiler and a single pressure level, as shown in the process flowsheet in Figure 2. In the boiler, there are three physically independent heat exchangers (e.g. economizer (ECO), evaporator (EVAP) and superheater (SH)) dedicated to well defined regimes such as heating of liquid water to or close to boiling point, evaporation and superheating. The superheated steam is expanded in a condensing type turbine, which drives a generator connected to the electric grid. We choose the drum configuration over a once-through boiler (with a single heat exchanger) because it is most common in operating power plants. Further, the drum has the advantage of energy storage which can provide additional operational flexibility.

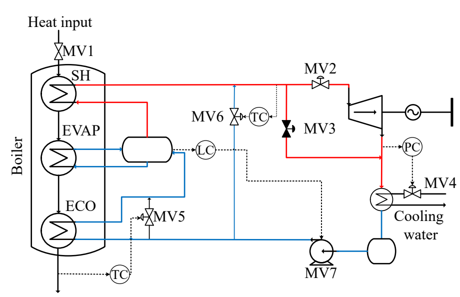


Figure 2: Flowsheet of a power plant with a drum boiler and one pressure level. MV1 and MV2 are the two remaining degrees of freedom considered in this paper. Liquid water is in blue and vapor in red.

3.2. Top-down analysis

We proceed by applying the top-down analysis to the described process.
Step 1. The plant has two operational objectives. On a slow time scale (steady-state) it should achieve the economic optimum, while on a fast time scale it contributes to the grid stability. 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. We define the objective cost function to be minimize the negative profit, given by Eq.1.

$$J = -(p_W W + p_Q Q - p_F F - p_U U) \qquad [\$/s] \tag{1}$$

Here, W [J/s] is the produced power, Q [kg/s] is the produced steam (if any), F [J/s] is the heat input source, U [kg/s] are the utilities, and p [\$/kg] or [\$/J] is the price of each. As we consider an operating plant, capital costs, personal, and maintenance costs are not included. The cost function should be minimized subject to satisfying a set of constraints, related to products specifications, safe operation and environment. Typical constraints for a heat-to-power cycle include:

- C1 Superheated steam pressure and temperature (e.g. $T \leq 550$ °C, $P \leq 200$ bar to minimize thermal and mechanical stresses and extend the operating life).
- C2 Condenser pressure should be low to maximize the pressure ratio in the turbine, but also above a minimum threshold to minimize condensation on the last turbine blade and reduce erosion (e.g. $P \geq 0.01$ bar).
- C3 Exhaust gas temperature should be low to minimize heat losses, but above the dew point to prevent corrosion (e.g. $T \leq 100$ °C).
- C4 Drum level (stabilizaion and safety).
- C5 Requirement to participate in grid frequency regulation (some plants).
- C6 Fixed turbine speed (equal to the grid frequency, e.g. $n = 50$ Hz in Europe). For this reason, the turbine speed is not a degree of freedom for operation.

In addition, there are operation constraints on the combustion side (e.g. O_2, CO_2 or NO_x percentage in the flue gas or furnace pressure for combustion power-plants), but these are not relevant for the purpose of this study of the water side.

Step 2. Table 1 shows the degrees of freedom, main disturbances and control variables including the active constraints (a subset of the operational constraints from *Step1*). Here, the active constraints are determined based on engineering insight. The MVs are also shown in Figure 2.

Table 1: Manipulated, disturbances and controlled variables

Manipulated variable	Disturbance variable	Controlled variable
MV1: Heat (fuel) input	DV1: Quality of heat input	CV1: Power
MV2: Steam turbine valve	DV2: Grid frequency (Load)	CV2: (Live) Steam pressure
MV3: Turbine bypass*	DV3: Cooling water temperature	CV3: Drum level
MV4: Cooling water		CV4: Steam temperature*
MV5: Economizer bypass		CV5: Condenser pressure*
MV6: Feedwater bypass		CV6: Exhaust gas temperature*
MV7: Feedwater pump		

* Active constraint (at steady-state) controlled in Figure 2.

Step 3. We want to select control variables such that desired optimal operation is maintained even when disturbances occur. Firstly, all the active constrains need to be controlled, and one degree of freedom is used to control each of them. One possible pairing for the identified constraints is shown in Figure 2, and is: control of steam temperature with MV6, gas temperature with MV5, condenser pressure with MV4, and the bypass MV3 should be closed. Further, the drum level can be controlled with MV7. The turbine speed is usually controlled by changing the steam mass flow using MV2. In this case, the setpoint given to the speed controller becomes a new degree of freedom instead of the valve position. In practice, this is often called *droop control*. In practice all turbines have droop control (Kurth and Welfonder, 2006). Note that the pairing of MVs and CVs (here, the active constraints) can be different, and that only the number of the remaining degrees of freedom is important for this step. We now have two degrees of freedom to use for optimal operation: the heat input MV1 and the and the speed controller setpoint MV2.

3.3. Droop Control

Droop control is commonly used to proportionally allocate the load between electrical generators running in parallel, to avoid a scenario where the load is taken by only one of them (Lipták and Bálint, 2003). This is a proportional controller, with a gain of typical 3 %/% to 10 %/%. In droop control, there is a fast MV (i.e. MV2, turbine valve), and a MV with a slow effect (i.e. MV1, heat). For example, when the turbine speed decreases, the droop controller acts by opening MV2, which increasing the steam mass flow. This can be sustained on a long term only if the heat increases as well. Figure 3 shows an example of 4 %/% droop, i.e. a change of power of 100 %/% gives $50 \cdot 0.04 = 2\,\mathrm{Hz}$ change in turbine frequency setpoint (Anderson and Fouad, 2003).

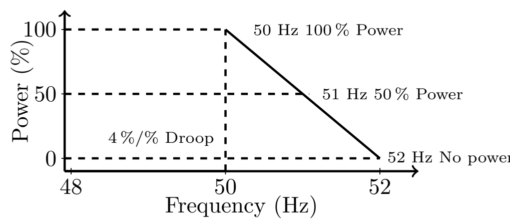


Figure 3: Droop characteristics of 4 %/%.

Step 4. The location of the throughput manipulator depends on the type of power plant, and we can identify two cases (i.e. economic modes) (Skogestad, 2004):

Case I The load is the throughput manipulator and optimal operation means maximizing the fuel usage (efficiency) (i.e. the plant is required to participate in the grid frequency regulation, or to supply utilities to a downstream process).

Case II The heat input is the throughput manipulator and optimal operation implies maximizing production (load). This case applies when the heat input is cheap, or “free” (e.g. solar or heat recovery from a gas turbine).

4. Control structures and operation modes

The standard industrial control structures are boiler driven, turbine driven, floating pressure and its variation, sliding pressure (Welfonder, 1999). However, if one disregards the common industrial practices for power plant control, alternative control structures can be proposed. For example, using valve position control (VPC), or both a P and a PI controller with a MID-selector. In the following, we only show the pairing options for the two remaining degrees of freedom (MV1 and MV2), and we assume that the inner turbine speed controller is active. We start with Case I.

4.1. Floating pressure operation

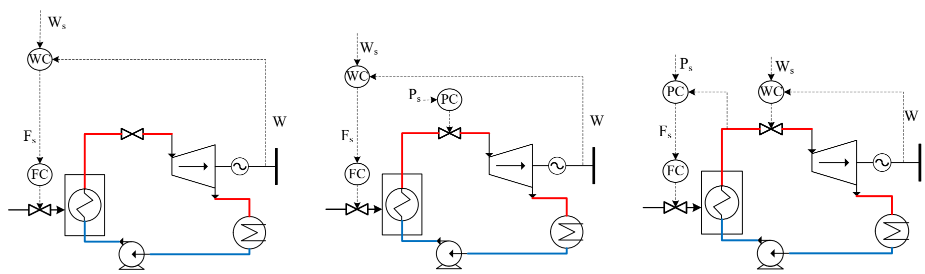
One degree of freedom is used to keep the throughput manipulator to its specification. The simplest options is to use MV1 to control the power output, and have MV2 fully open to minimize throttling losses. This structure is called floating pressure because the live steam pressure is left uncontrolled, as shown in Figure 4a. However, when a constraint on the maximum allowed pressure is reached, and the set of active constraints has changed, we have to give-up another constraint that is less important (Reyes-Lúa et al., 2018). In this case, we have to use the steam turbine bypass (MV3) as a degree of freedom. The disadvantage is a slow time response for load changes.

4.2. Boiler driven operation

To improve the response time of the system, one should control another intermediate variable. The only option is to keep the live steam pressure at a given set-point. The result is the boiler driven control structure, shown in Figure 4b. The output power is controlled with MV1, while the live steam pressure is controlled with MV2. This structure has the advantage that it dynamically responds faster to a change in load by utilizing the energy stored in the system, which comes at an expense of not utilizing the heat input at the maximum. Though it has a faster response compared to the floating pressure, the system still has a large time constant from MV1 to the power output, and the overall response is slow.

4.3. Turbine driven operation

The reverse pairing is turbine driven, as illustrated in Figure 4c. The live steam pressure is controlled with MV1, while the power output is controlled with MV2. It also has the advantage of utilizing the drum and the superheater energy storage, and the response time to load changes is faster. However, it also does not efficiently utilizes the heat input due to valve throttling losses.



(a) Floating Pressure (b) Boiler driven (c) Turbine driven

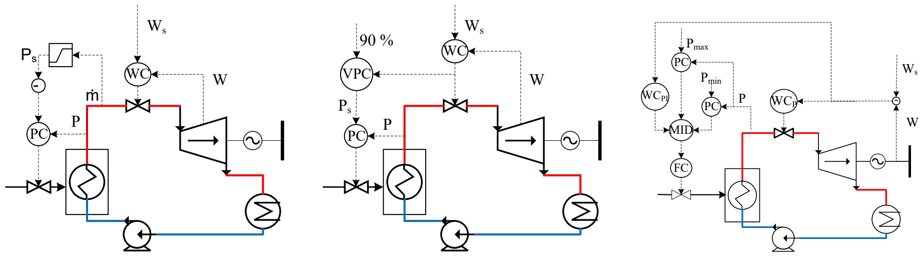
Figure 4: Control structures for three operation modes for a simple heat-to-power cycle.

4.4. Sliding pressure operation

Sliding pressure is when the pressure setpoint is adjusted online, usually for the turbine driven operation mode. In this case, the power plant can participate in the primary frequency control, at the expense of having to back-off from optimal operation and have MV2 partially open (e.g. 90 %) (Weissbach et al., 2006). The cycle efficiency decreases due to throttling losses in the steam valve, but it allows the power plant to have a energy backup that can be dynamically used by fully opening the valve, and participate in the grid frequency control. However, assuming an isenthalpic expansion through the valve, the steady-state effect will be small. In practice, it is common to have a pressure controller with a varying setpoint given by a master controller in a cascade structure that receives the load as input, as shown in Figure 5a (Klefenz, 1986). Here, the pressure setpoint is changed based on a model that has the steam mass flowrate as input and the pressure setpoint as output.

Valve position controller. The conventional sliding pressure control structure can be improved by adding a valve position controller to bring back MV2 to its nominal opening (at steady-state) after it has dynamically contributed to grid frequency regulation, as shown in Figure 5b.

MID selector. We use both MV1 and MV2 to control the power, and the pressure is floating, as illustrated in Figure 5c. Here we use one P-controller, and one PI-controller because we can have only one integral action in a two-inputs one-output system. When pressure becomes an active constraint (i.e. at the minimum or maximum), we use a *MID* selector to control it using MV1.



(a) As shown in (Klefenz, 1986) (b) With a valve position controller (c) With a *MID* selector

Figure 5: Control structures for sliding pressure operation mode

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

From a steady-state point of view, there are no degrees of freedom left to control the power. However, the turbine bypass can be used to reduce the power produced by decreasing the steam mass flow. However, the efficiency of the cycles will also decrease. Ideally, one should make use of energy storage solutions of superheated steam for the situations when more steam can be produced than is needed. This is an attractive idea for power plants using solar energy, especially since the maximum solar power is at midday, when the electricity market demand is lower.

5. Conclusions and final remarks

We have systematically identified the operational objectives, operational and environmental constraints for a heat to power cycle. The degrees of freedom left are MV1, the heat input, and MV2, the steam turbine valve. Now that we have systematically defined the optimal operation and control problem for power plants, it becomes easier to make improvements using more advanced and systematic control policies, such as model predictive control.

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