Control structures for heat delivery in compact bottoming cycles for heat and power production ⋆

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Abstract: Compact designs of combined cycle power plants based on gas turbines and steam bottoming cycle (CCGTs) are deemed as a promising technology for increasing energy efficiency and reducing greenhouse gas emissions of offshore oil and gas production facilities. The control of such systems can be challenging due to the need for operational flexibility regarding production of power and heat to satisfy the corresponding demands, and it differs from traditional onshore designs in dynamic characteristics and requirements. In this work, we propose and evaluate the performance of control structures for compact steam bottoming cycles with combined heat and power production, focusing on the solutions for satisfying heat demands and their effect on power production. The proposed control structures are based on the different prioritization of operational objectives and constraints, using simple control elements to switch between operating regions. The control structures were evaluated under different disturbances on the gas turbine loads and on the heat demand. It is shown that controlling the intermediate pressure in the steam turbine, which serves as source of steam for heat production, is necessary for achieving the heat demand objectives. We also show that sudden disturbances on the heat demand heavily impact the power production, and it is desirable that such disturbances happen on a ramp-like manner. Overall, we highlight how near-optimal operation with satisfaction of constraints can be achieved with the use of well-designed, simple control structures.

Keywords: process control, combined cycle power plants, control structure design, plantwide control, advanced regulatory control

1. INTRODUCTION

Offshore oil and gas production is an energy-intensive process in both power and heat. The largest contributor to greenhouse gas (GHG) emissions is gas turbines, which are utilized for electrical power production and as mechanical drives for export gas compressors. This occurs mainly due to the combustion of gas in turbines on platforms. Gas turbines contributed to 82.91% of the GHG emissions from the Norwegian Continental Shelf in 2023 (Norwegian Offshore Directorate, 2024). Some oil and gas installations, such as Floating Production, Storage and Offloading vessels (FPSOs), are not easily adaptable for emission reduction measures such as electrification. Therefore, improvements in the efficiency of gas turbines are desired to reduce GHG emissions. To that end, an option is to add a bottoming cycle to convert excess heat from the gas turbine exhaust to produce additional power and/or heat. Combined cycle power plants based on gas turbines and steam bottoming cycle (CCGTs) are standard, high efficient and flexible power generation technologies onshore. However, CCGTs are not widespread offshore due to space and weight limitations (Kloster, 1999; Montañés et al., 2023), which limits the use of traditional drum-based steam generators for which operation and control are well known (Ahluwalia and Domenichini, 1990).

Compact and low weight equipment is required offshore to attain a feasible design (Kloster, 1999; Voldsund et al., 2023). This leads to CCGT designs based on steam cycles with once-through steam generator (OTSG) waste heat recovery boilers with a single pressure level, and lower water inventory in hold-ups. In addition, the tube bundles must use small tube diameter in the OTSG heat exchangers, leading to smaller thermal inertia of the OTSG walls, which in turn affects the dynamic response time of the OTSG (Montañés et al., 2021). These systems have a different control philosophy from traditional drum-based systems, for which the hold-ups are actively controlled (Zotică et al., 2020). Reliability and availability are crucial in offshore oil and gas operations, and particularly the availability of low carbon technologies is of paramount importance to enable deployment and emission reductions. The CCGT must be set up for flexible operation, in order to provide variation management to the offshore energy system heat and power demands across multiple time

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scales. Due to these reasons, careful studies on control structure design are crucial. Recent work has addressed process control of the power only configuration for offshore combined cycles (Nord and Montañés, 2018; Zotică et al., 2022; Montañés et al., 2024). However, the operation of the heat and power configuration has received little attention in the past, which is the focus of this work.

Here, we focus on a cogeneration heat and power (CHP) CCGT power plant configuration in an FPSO system context. The goal is to design and compare different decentralized control structures for operation of the steam bottoming cycle, in order to evaluate how the choices on control structure design impact the operational performance of the system. We focus on the control loops that are needed for heat demand operation.

2. REFERENCE SYSTEM AND DYNAMIC MODEL

In this work, we focus on CCGT offshore that produces both electricity and heat as main products. We build on the process design of a FPSO CCGT from the case study presented by Montañés et al. (2023). A process flowsheet of the system in shown in Fig. 1. The combined cycle contains two Siemens SGT-750 gas turbines (GT) and one steam turbine (ST) to cover the power demand. The heat in the gas turbine exhaust is recovered and transferred into the bottoming cycle by the once-trough steam generators (OTSGs) connected to the GTs' exhaust. The superheated steam is then mixed and used to generate power in an ST. Steam extracted between the high-pressure (HP) and low-pressure (LP) sections of the ST is used to provide heat in the heat demand unit. This unit is here modelled as a steam condenser, where the goal is to heat the cold fluid to its desired temperature, representing the heat demand from the offshore energy system. After the LP ST, the steam is cooled in the condenser and recycled back to the OTSGs by two feedwater pumps. The nominal operating conditions as well as the method for process design optimization are presented in Montañés et al. (2023). We develop a new dynamic process model in Modelica language and Dymola simulation environment (Dassault Systèmes, 2024), building on models from the Thermal Power Library (Modelon AB, 2024). We parameterize the new process model for the process configuration and a selected design presented in Montañés et al. (2023). The underlying dynamic models and modeling framework are presented for a different configuration by Montañés et al. (2021); Montañés et al. (2024). In this work, we apply the same dynamic modeling approach for GTs, OTSGs, STs, valves, pumps, condensers and control blocks.

3. CONTROL STRUCTURE DESIGN

Here, we follow the self-optimizing control procedure of identifying the relevant operational constraints to design a control structure that operates near optimally given disturbances (Skogestad, 2000). According to the steady-state analysis in Montañés et al. (2024) for the steam cycle, the most relevant constraints for operation are related to the outlet temperature of the OTSGs and the steam header pressure, which means that these process variables should be controlled. Based on that, the basic control loops for the bottoming cycle are given in Fig. 1. Given

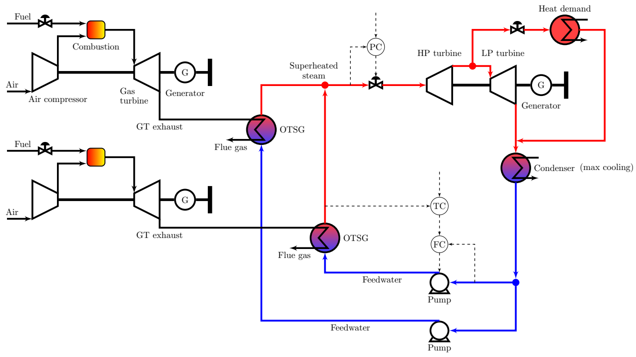


Fig. 1. Process flowsheet for the combined cycle, with feedback controllers based on Montañés et al. (2024), common to all cases investigated in this work. This corresponds to the constant pressure operation mode, while the sliding pressure operation mode does not present a pressure controller for the superheated steam.

that control of steam header pressure reduces the energy efficiency of the process, we also evaluate the operation of the process without pressure control at the steam header ("sliding pressure" operation mode).

Additional control loops are needed to deal with variable heat demand, while at the same time assuring safe operation of the bottoming cycle; refer to Fig. 2 for the control structures considered in this work. The relevant objectives for a bottoming cycle with heat extraction are:

C1. $T_{out}^{cold} = T_{out}^{cold,sp}$: deliver the required heat by heating the cold stream to its desired temperature.

C2. $p_I \geq p_I^{min}$: keep the intermediate pressure (IP) between the HP and LP STs above the minimum allowed limit to avoid tripping of the LP ST. This is also linked to C1 in some operating conditions, as the saturation temperature of the steam depends on the pressure, which limits the achievable temperature of the cold product (T_{out}^{cold}).

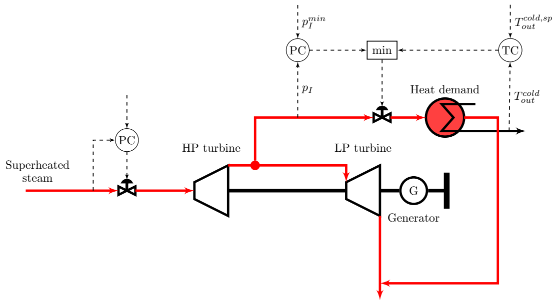
C3. $p_I \leq p_I^{max}$: keep the intermediate pressure below a maximum allowed limit to avoid high mechanical stress on the LP ST (unlikely to be active).

C4. Maximize the flow that goes to the STs whenever possible, to maximize electrical power generation.

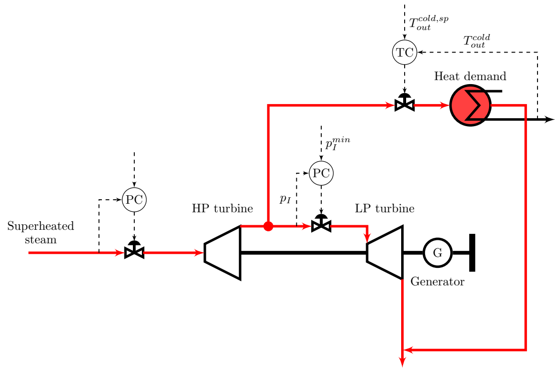
Objectives C1 and C2 may be conflicting in cases where not enough steam is generated in the cycle, as increasing the steam draw to achieve the heat demand will decrease the pressure at the draw point. Therefore, one should establish a priority for these constraints when proposing a control structure. Fig. 2 shows the different control structures that were proposed, for which we have different priorities for the operational constraints.

The control structures are proposed with the following rationale:

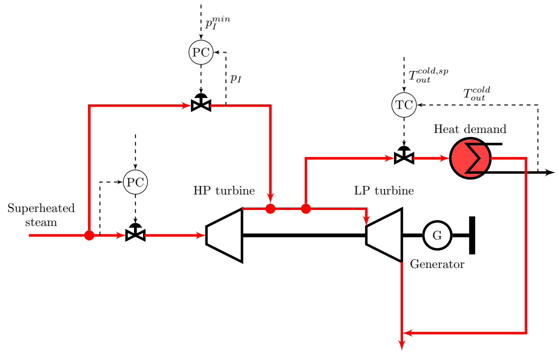
- If C2 is deemed more important than C1, we propose the control structure in Fig. 2a, where control of T_{out}^{cold} is given up in favor of control of p_I, which is achieved with the use of a selector block. A selector of type "min" is used due to the process gains and the constraint sign (Krishnamoorthy and Skogestad, 2020).



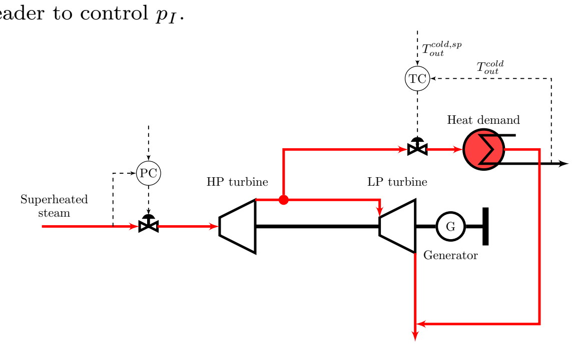
(a) **"Selector"**: use of same valve to alternately control T_{out}^{cold} and p_I.



(b) **"IP valve"**: use of dedicated intermediate pressure valve between the HP and LP turbines to control p_I.



(c) **"Bypass valve"**: use of bypass valve from the OTSG header to control p_I.



(d) **"No IP control"**: reference structure with no additional control loops for p_I.

Fig. 2. Control structures for heat demand operation.

- If it is not desirable to give up on C1, we can satisfy C2 by constraining the flow to the LP ST to control p_I (giving up on C4), as proposed in Fig. 2b.
- Another possibility of giving up on C4 over C2 is by introducing a bypass from the main superheated steam header for controlling p_I, see Fig. 2c.

These control structures are compared to a reference strategy of floating (uncontrolled) intermediate pressure, p_I, as shown in Fig. 2d.

3.1 Implementation methodology

Using the PID formulation in Skogestad (2003), all feedback controllers are implemented and tuned using the SIMC tuning rules with $\tau_C = 5\ s$, where τ_C is the tuning parameter that dictates the desired closed-loop time constant. This gives the controller parameters in Table 1 for the PI or pure I controllers, depending on the identified system dynamics.

Table 1. Controller tunings for each system configuration at constant pressure operating mode.

	K_c	τ_I $[s]$	K_I
Common controllers			
OTSG header pressure	-8	20	
T_{out}^{cold}	0.4167	7	
Controllers for p_I			
Selector			-0.94
IP valve	-1.692	2	
Bypass valve			0.2588

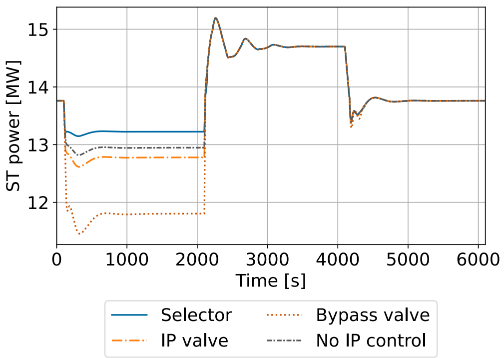
We use the methodology described in Zotică et al. (2022) for implementing a feedback temperature controller based on nonlinear input/output transformations, with the use of additional measurements related to the exhaust gas and to the OTSG header for nonlinear feedforward action.

3.2 Simulation scenarios

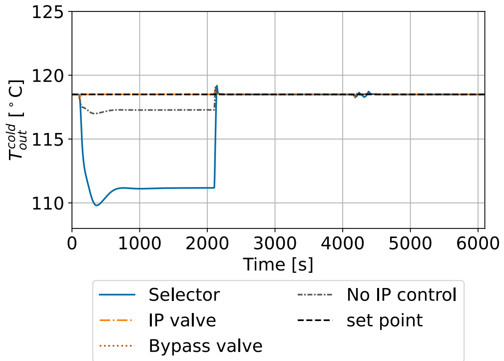
To evaluate the performance of the different control structures, the closed-loop system models were simulated for different operating conditions using step- and ramp-like disturbances in variables of interest to the system, namely the GT loads and the inlet conditions of the cold fluid to be heated (heat demand). For the dynamic evaluation of the control structures, the closed-loop systems were subject to changes of $\pm 10\%$ on the disturbance values. A steady-state analysis of the influence of the gas turbine loads on the bottoming cycle system performance was also performed by simulating the systems for the respective loads until a steady state was achieved.

4. RESULTS

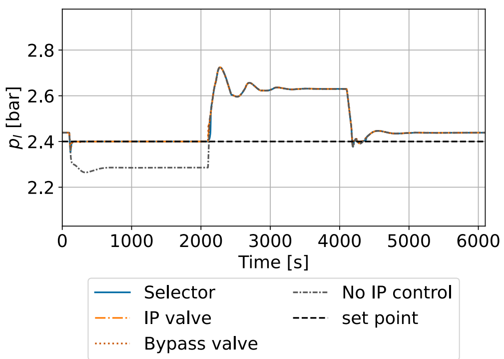
Fig. 3 shows the dynamic response of all closed-loop systems in constant header pressure mode when subject to step changes in the GT loads. As the systems are designed to operate above the limiting intermediate pressure at the nominal condition, the control structures behave the same at the nominal operating point and for higher GT loads. We can see that the use of the bypass valve leads to the largest power losses at the steam cycle for lower GT loads. For the selector control structure, control of the heat demand is given up in favor of maintaining the pressure above its threshold. We can also note that not controlling the intermediate pressure leads to a loss of control of the heat delivery, due to the corresponding saturation temperature of the steam not being high enough. The use of a dedicated valve for control of the intermediate pressure is shown to be a good alternative to ensure that the heat demand is satisfied, with a lower power output for the steam cycle.



(a) ST power response



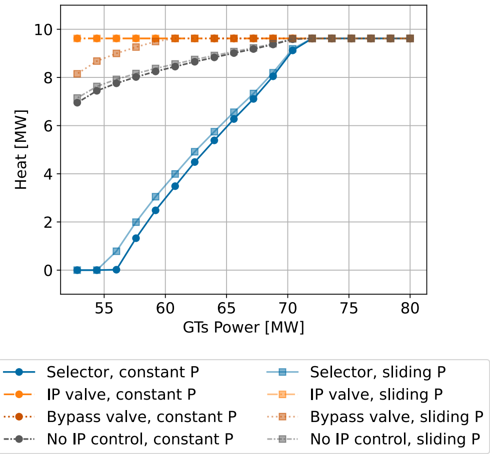
(b) Cold fluid outlet temperature response



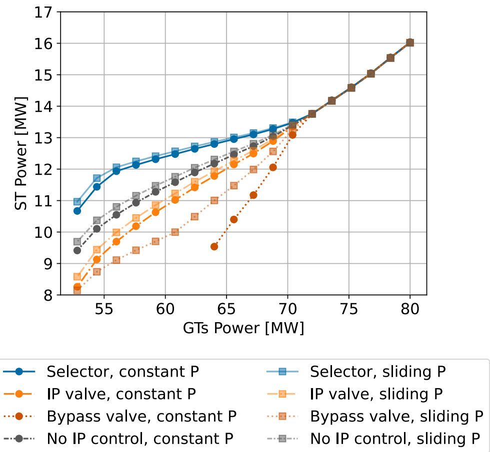
(c) Intermediate pressure response

Fig. 3. Step changes in load of GTs for system operating at constant pressure mode.

The steady-state behavior as function of the GTs load can be seen in Fig. 4. Comparing the two modes of operation in terms of OTSG header pressure (constant vs. sliding pressure), we see slightly higher power and heat output for the sliding pressure operation mode. We can also see that using the bypass valve for controlling the intermediate pressure (control structure "Bypass", see Fig. 2c) gives lower power and heat outputs when compared with the use of a dedicated valve between the turbines ("IP valve", see Fig. 2b), making it a worse option overall. Comparing the control structures labelled "IP valve", "No IP control", and "Selector" (Figs. 2b, 2d and 2a, respectively), we see that there is a trade-off between produced power and delivered heat demand, with "IP valve" being the control structure that prioritizes heat delivery and "Selector" being the one that prioritizes produced power at the STs.



(a) Heat flow response



(b) ST power response

Fig. 4. Steady-state curve for closed-loop operation as a function of the power provided by the gas turbines. Two modes of operation in terms of OTSG header pressure (constant vs. sliding pressure).

The effects of changes of the heat demand side into the system operation are seen in Figs. 5 to 7. We see a similar behavior to the previous simulations in that a larger loss of recovered energy is obtained when using the turbine bypass valve to control the intermediate pressure. While a step change in the inlet cold fluid temperature (Fig. 5) does not heavily impact the system dynamically, a step change in the cold fluid flowrate (Fig. 6) gives large overshoots on the produced power, linked to the initial effect on the intermediate pressure. Notice that these overshoots happen for all control structures, and they happen in a faster time scale than that of the controllers' tunings. A ramp change in the cold fluid flowrate of the same magnitude (Fig. 7) removes the undesired overshoots, as the changes in the process happen more gradually.

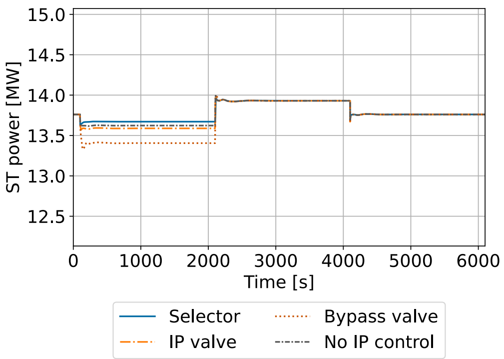


Fig. 5. ST power response for step changes in inlet temperature of cold fluid (heat sink) for system operating at constant pressure mode.

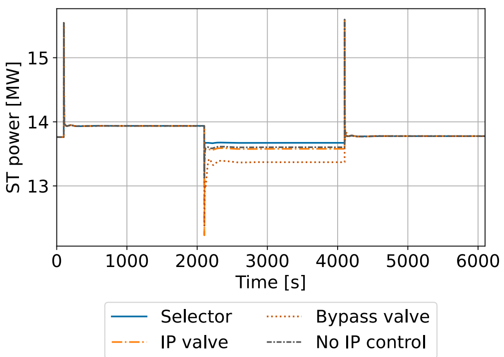


Fig. 6. ST power response for step changes in flowrate of cold fluid (heat sink) for system operating at constant pressure mode.

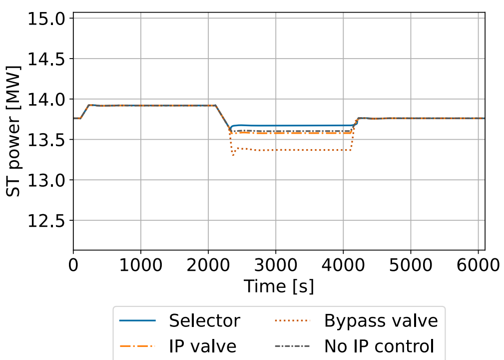


Fig. 7. ST power response for ramp changes in flowrate of cold fluid (heat sink) for system operating at constant pressure mode. Ramps are implemented at a rate of 5% of the nominal flowrate per minute.

5. DISCUSSION

5.1 Changes in control objectives and optimal operation

In the control structures presented in this work, we rely on changes of controlled variables during operation as the disturbances change. This is done due to constraints becoming active in some operating regions, which means that the optimal control policy is to control these constraints to their limiting values. This approach can be viewed as the solving of optimization problems through the implementation of feedback control, and is particularly straightforward if we know that the optimum lies at the system's constraints. Notice that even a control structure such as Fig. 2b can be interpreted as a switching system for optimal operation, where the operation with low GT loads has the dedicated valve being used to control the intermediate pressure (an active constraint), and operation with high GT loads has the valve being saturated at its maximum allowed opening, with the pressure control being given up, and the flow through the ST being maximized.

Even though most control problems are concerned with the dynamic performance of the system, control structure design is mostly related to the *steady-state* performance of the system. We illustrate in Fig. 4 how the choice of controlled variables affects the system's steady-state behavior, especially in terms of power and heat production. Proper tuning of the dynamic elements of the controllers, namely PID and ani-windup action tuning parameters, ensure that the designed control structure behaves well dynamically. In this case, this could be, for example, minimizing the overshoots in control of p_I (see Fig. 3c), which reduces the need for large back-offs. One may propose flexible control structures that allow for switching the steady-state priority of the power and heat production objectives, but that is beyond the scope of the present study.

5.2 Flexible operation of CHP CCGT systems

One of the main challenges for the operation of CCGT systems with production of heat and power is the variations in demand, which lead to large changes in flowrates through the different sections of the steam cycle. This may cause significant efficiency losses in equipment such as the STs (Thern et al., 2014). Not only this is an issue to equipment design, it is also an issue for the control layer design, which should be prepared for such changes and give appropriate responses. In particular, the design constraints also play a role during operation, as these should be ensured by the control layer. These constraints may refer to minimum and maximum flowrates, pressures, or temperatures, which can be measured online and controlled in a simple manner. Therefore, the study of control solutions for such systems is of particular importance, such that process flexibility can be achieved without great compromises in energy efficiency.

For this system, there is the need for maintaining the intermediate pressure above a minimum threshold for delivering the necessary heat. Intermediate pressure turbines are found on other non-compact CHP plants onshore, based on heat recovery units with drum-based systems. This is known as automatic extraction turbines, and they allow part of the steam to be withdrawn at an intermediate stage, while the remainder of the steam is exhausted to the condenser. These turbines require special governors and valves to maintain constant pressure of the extraction steam while the turbine load and extraction demand are varying (Bolland, 2010). However, to the best of the authors' knowledge, the evaluation of its performance in an offshore energy systems context and for compact designs has not been addressed before in the literature.

With the use of the control structure in Fig. 2b, called "IP valve", we can maintain the pressure level at the expense of reducing the opening of the feed valve to the LP ST. This may cause undesired operating conditions, both in terms of the pressure after the valve and the flow through the LP turbine (which are correlated). These constraints should also be taken into account by the control layer. This evaluation is dependent on the turbine technology being considered, which is beyond the scope of the present study.

5.3 Disturbances on heat demand

The heat demand for the CHP CCGT plant in this work is modelled as a cold fluid being heated to the desired temperature. The disturbances in this system can therefore be in the cold fluid's inlet temperature, desired outlet temperature, or feed flowrate. Out of these, the most relevant is the flowrate, which is related to the desired production. Its variability, however, is comparatively slow to the dynamics of the steam cycle for most systems. This means that a slow change for these disturbances is a reasonable assumption, which implies that the feedback controllers will have good performance for rejecting these disturbances. We illustrate this in the simulations of this work, where fast (step-like) changes in cold fluid flowrates resulted in large overshoots (Fig. 6), whereas slow (ramp-like) changes were tracked with good performance (Fig. 7).

6. CONCLUSION

This work has investigated the use of different control structures for the operation of a bottoming cycle for a CHP CCGT power plant. We highlight that the choice of a control structure is heavily linked to the operational objectives that are prioritized, be it the maximization of the produced power or the satisfaction of constraints. Moreover, achieving near-optimal operation of such systems with simple control structures has the potential to increase their applicability and efficiency, leading to an overall reduction in emissions and fuel usage.

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