A Model of the Control System of a Carbon Dioxide Gas Turbine in Supercritical Condition

Marcin Zawadzki, Jarosław Milewski[a] and Arkadiusz Szczęśniak
Institute of Heat Engineering, Faculty of Power and Aeronautical Engineering, Warsaw University of Technology, 21/25 Nowowiejska Street, 00-665 Warsaw, Poland

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Abstract: The primary goal of the paper was to develop a model for a supercritical carbon dioxide gas turbine. The model was built using the GateCycle program It is designed for potential implementation in emerging Concentrating Solar Plants, with a focus on simple yet efficient construction in the recuperation model. Analyses were conducted on three types of power control systems: bypass, inventory, and turbine inlet temperature-based, using the Lee-Kesler real gas model for calculations. Key mathematical formulas used by the program are cited, and results are thoroughly analyzed and presented in charts. In conclusion, a combination of bypass and inventory control systems is recommended.

1 INTRODUCTION

The global energy industry is currently moving toward renewable energy sources. The main obstacle standing in its way is the heavy dependence of energy production on non-anthropogenic factors. Various strategies could be applied, such as the integration of Renewable Energy Sources (RES) with electrolysers for synthetic fuel generation (Milewski *et al.*, 2024) or combining RES with traditional technologies. An interesting idea is the use of gas turbines operating in closed circuits. These systems show good controllability, which can help solve the outlined problem. The closed-cycle Gas Turbine (GT) could serve as a power conversion system for non-CO2 emission energy sources (Olumayegun, Wang and Kelsall, 2016) such as nuclear reactors, concentrated solar power, biomass, geothermal, and fuel cells (Kim, Kim and Favrat, 2012).

Contrary to appearances, this is not a new, innovative technology, as the first concepts for the construction of closed-cycle gas turbine systems appeared as early as the 1930s (Miller, 1984). The most likely reason for the stagnation in the development of this area was the rapid growth of open-system gas turbine systems. Open-circuit systems dominated the gas turbine market after the use of blade cooling technology, which enabled turbine inlet temperatures of 1,700°C, allowing them to achieve significantly higher efficiency than closed-circuit systems. Thanks to the development of high-temperature-resistant materials and the emergence of new potential applications, closed-cycle gas turbine systems are once again becoming the focus of research centers and are poised to enter the industrial market. Due to their nature, the opportunity for commercial implementation of closed-loop gas turbine systems is in Concentrating Solar Plants (CSP) in Solar Tower (ST) systems (Stein and Buck, 2017; Merchán *et al.*, 2021) . This technology is recognized as a renewable and carbon-neutral source of energy (Wu *et al.*, 2021) . For this reason, the main parameters of the system have been selected so that it can be implemented in newly built power plants of this type. The world's largest Solar Tower Concentrated Solar Plant has a net capacity of 377 MW (Boretti, Castelletto and Al-Zubaidy, 2019). State-of-the-art technologies using molten salts allow working medium temperatures of 550°C and energy storage for several hours.

The optimization of the system with recompression was performed by Sarkar et al.

(Sarkar and Bhattacharyya, 2009) . The authors indicated that the maximum efficiency improvement using reheating is determined to be 3.5% under

[a] https://orcid.org/0000-0003-1215-1802

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optimum conditions. An in-depth analysis of the topic was reported by Dostal et al. (Dostal, Driscoll and Hejzar, 2004) , who concluded that the recompression cycle offers the best efficiency and retains its simplicity. Intercooling, however, is less appealing for this cycle type because it only provides a slight boost in efficiency. Reheating holds more promise, though it is suitable solely for indirect cycles.

Ma et al. (Ma *et al.*, 2024) developed a model of a 5 MW S-CO2 recompression cycle oriented toward operational control. The model includes validated dynamic models of the following components: printed circuit heat exchangers, compressors, and gas turbines. The validated control strategies for the system involve using a cooling water controller to keep the compressor inlet temperature above CO2's critical temperature (304.13 K). They also adjust the circulating mass flow rate to change output power and control the exhaust gas flow rate to maintain safe turbine inlet temperatures. Performance comparisons between PI (Proportional-Integral) controllers (tuned via SIMC rule) and ADRC (Active Disturbance Rejection Control) controllers (tuned via bandwidth method) show that both effectively manage operating conditions and suppress exhaust fluctuations. However, ADRC controllers outperform by reducing settling time by 55% in load-tracking scenarios.

Dario et al. (Alfani *et al.*, 2021) optimized various sCO2 cycles for waste heat recovery. The simple recuperative cycle with bypass was selected as the optimal configuration. The authors revealed that the implementation of an external CO2 storage tank, which enables varying the CO2 inventory in the system, beneficially influences the system's efficiency during part-load operation. Specifically, when dealing with low flue gas mass flows—at 30% of the nominal rate—a 20% enhancement in power generation is achievable compared to strategies where the CO2 inventory remains unchanged.

The purpose of this paper is the theoretical analysis of various methods of power control of a fabricated gas turbine model with CO2 with supercritical parameters as the working medium, together with a review of current research on the systems considered. Three basic control methods will be compared with each other: bypass, by displacement, and by temperature. Despite numerous studies related to the subject under discussion, the problem under consideration has not yet been sufficiently addressed.

The paper presents a novel model of a supercritical carbon dioxide gas turbine designed for potential implementation in emerging Concentrating Solar Plants. The key contributions of this paper include the development of a comprehensive model using the GateCycle program, focusing on simplicity and efficiency in the recuperation model. The model integrates existing power control methods, namely bypass, inventory, and turbine inlet temperature-based controls, and evaluates their effectiveness using the Lee-Kesler real gas model for calculations. One significant contribution is the recommendation of a combined approach of bypass and inventory control systems, which balances the need for quick responsiveness and maintaining high efficiency. The paper thoroughly analyzes these control systems, providing detailed mathematical formulas and results presented in various charts. Additionally, the model's adaptability to high-pressure and high-temperature conditions using the Lee-Kesler equations marks an advancement in accurately simulating supercritical carbon dioxide gas turbine systems. This work paves the way for integrating such advanced models into next-generation renewable energy power plants, specifically those utilizing solar energy, thus contributing to the global shift towards sustainable energy solutions.

The application of closed-cycle gas turbine systems, especially those utilizing supercritical carbon dioxide as the working medium, presents a promising solution for integration into Concentrating Solar Plants (CSP). These systems are particularly suited for Solar Tower (ST) configurations, where high operational temperatures and efficiency are paramount. By leveraging the unique properties of supercritical CO_2, these turbines can achieve higher efficiencies and better performance in CSP applications compared to traditional working fluids. This makes them an ideal choice for enhancing the viability and sustainability of solar energy power generation. Consequently, the implementation of these advanced gas turbine systems in CSP not only supports the drive towards renewable energy but also addresses the need for efficient and reliable power conversion technologies in solar energy applications. The advancements in control strategies and system optimization outlined in this paper further underscore the potential of these technologies to play a critical role in the future of renewable energy infrastructure.

2 THEORETICAL BACKGROUND

One of the advantages of gas turbines in closed systems is the ability to regulate power over a large range with almost no drop in efficiency. The power of the system depends on the mass flow, the turbine inlet temperature (TIT), the gas temperature before the compressor, the compression ratio, and the efficiency. Control possibilities include the use of a storage tank, bypass, or temperature control.

The most advantageous method is to change the discharge by using a storage tank. This is called discharge control or pressure control. When decreasing from the current load, some of the gas is diverted to the storage tank. This causes a decrease in mass flow and pressure at the compressor inlet. As power increases, gas is released from the tank into the system. Within a large power range, this method allows smooth control with a relatively small decrease in efficiency—see Figure 1 , line "inventory control." The downsides of this method are the relatively slow response time and the need for large storage tanks. For this reason, this method of regulation should be abandoned in favor of the others for larger power units. A schematic of the system with applied mass flow control is shown in Figure 2 (a).

Another method is bypass control. It involves the use of a bypass of the expander and the heat exchanger with the upper heat source, reducing the flow of gas through these components while leaving the compressor unchanged. A schematic of the system with bypass control is shown in Figure 2 (b). The advantage of this solution is the ability to respond almost immediately to rapid changes in power. The major disadvantage, on the other hand, is the significant drop in system efficiency during power reduction.

The third method is upper source temperature control. This is the slowest method presented but provides higher efficiency at lower loads than bypass control.

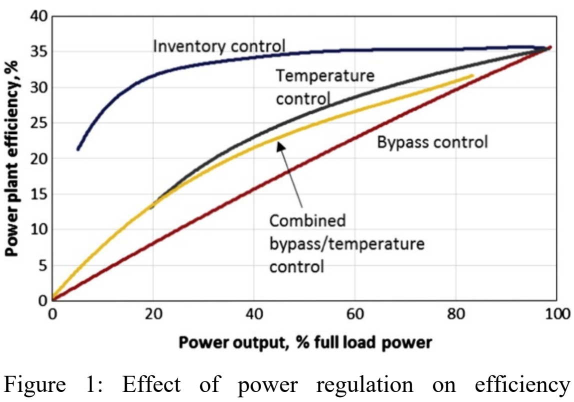

Figure 1: Effect of power regulation on efficiency (Olumayegun, Wang and Kelsall, 2016).

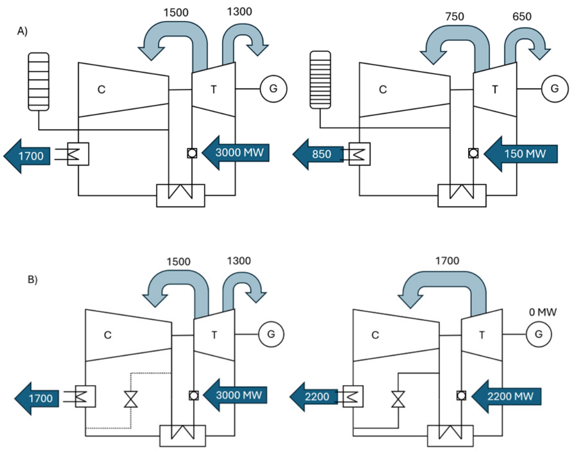

Figure 2: Schematic of the power control system: (a) method using a storage tank (b) bypass method.

2.1 Closed System of Gas Turbine

Figure 1 shows the dependence of system efficiency as a function of load for the methods described. The best solution is to use a combination of bypass regulation, which would operate when responsiveness is crucial, with regulation using a regulating tank—slower, but allowing the system to maintain a high level of efficiency.

When analyzing the operation of the system under changed conditions, it is important to note the most significant relationships describing it. When controlling the system, the discharge pressure from the compressor changes due to the forcing caused by the turbine inlet. This forcing is determined by the following relation:

$$Constant = \frac{m_{inlet}\sqrt{T_{inlet}}}{\kappa \cdot A_{inlet} \cdot P_{inlet}} \tag{1}$$

Where:

$$\kappa = \sqrt{g \cdot \frac{\gamma}{R_{gas}} \cdot \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}} \tag{2}$$

The speed of the compressor also changes, which can be determined by an algorithm using the flow equation:

$$\frac{n}{n_D} = \frac{CS}{CS_D}\sqrt{\frac{T_{inlet} \cdot Z \cdot \frac{MW_D}{MW}}{T_{inletD} \cdot Z_D}} \tag{3}$$

Where the subscript D stands for design parameters (design mode).

In situations where access to the compressor map is impossible, its isentropic efficiency can be calculated based on the equation:

$$\eta_S = \eta_{max} \left(1 - SEC \cdot \left| \frac{CS - CS_{max}}{CS_{\eta_{max}}} \right| \right) \cdot PF \cdot (1 - \alpha \cdot VEC) \tag{4}$$

The above relationships are applicable to GateCycle software and will be used to make the model.

3 MODEL DEVELOPMENT

Research is based on the mathematical model of a supercritical carbon dioxide gas turbine. The model has been constructed in the GateCycle program, which capabilities in reliability of simulations of are confirmed by many papers such as e.g. (Malinowski, Lewandowska and Giannetti, 2017; Ary Bachtiar and Maryono, 2019; Hasananto, Darmadi and Yuliati, 2021). It provides accurate thermodynamic analysis, integrates detailed component models, and allows for precise performance predictions under various operating conditions.

The model's parameters have been set so that it could be implemented in emerging Concentrating Solar Plants. The recuperation model has been created for simplicity of construction while providing high efficiency. Analyses have been performed on three types of power control systems: bypass, inventory, and turbine inlet temperature-based. Real gas equations were used in the model. For calculations in areas of very high pressures or temperatures and around the critical point, these equations may not provide sufficient quality results. According to GE Energy, the Lee-Kesler equations give the closest results for CO2 and air to VDI Warmeatlas table data over a large range of pressures and temperatures. For this reason, it was decided to use this real gas model.

The disadvantage of the model is the inability to simulate the molten salt compounds forming the upper heat source. For this reason, it was described as a water flow with the most relevant parameters the same as the actual medium.

The main assumptions for the simulation have been listed in Table 1.

Table 1: Main assumptions for model development.

Parameter	Value
Temp. at inlet to compressor, °C	31
Pressure at inlet to compressor, MPa	7.4
Temp. at turbine inlet, °C	550
Pressure at turbine inlet, MPa	24
Mass flow rate of the medium, kg/s	500
Max. upper heat source temp, °C	565
Ambient temp., °C	20
Real gas model	Lee-Kesler model

A system with heat regeneration was chosen for the simulations because of its relatively high efficiency while keeping the system simple and compact. The layout of the model implemented in GateCycle is shown in Figure 3 and the main operating parameters are shown in Table 2.

Table 2: Parameters of the examined cycle.

	T	p	M	H
	K	kPa	Kg/s	kJ/kg
1	347.82	24487.3	500	155.53
2	347.82	24487.3	500	155.53
3	569.95	24242.4	500	216.48
4	823.15	24000.0	500	535.28
5	686.25	7550.3	500	384.84
6	362.82	7474.7	500	9.11
7	362.82	7474.7	500	9.11
8	304.15	7400.0	500	187.35
9	293.15	101.3	500	4.25
10	351.82	100.31	500	63.81
11	838.14	4000.0	500	3592.77
12	669.94	3960.0	500	3208.89

The model consists of a compressor (C1 in Figure 4), which was simulated with the following settings listed in Table 2.

Table 3: C1 compressor settings.

Parameter	Value
Desired Isentropic Efficiency	0.82
Desired Outlet Pressure	24,487.297 kPa
Design Compressor RPM	3,000
Mechanical Efficiency	0.98

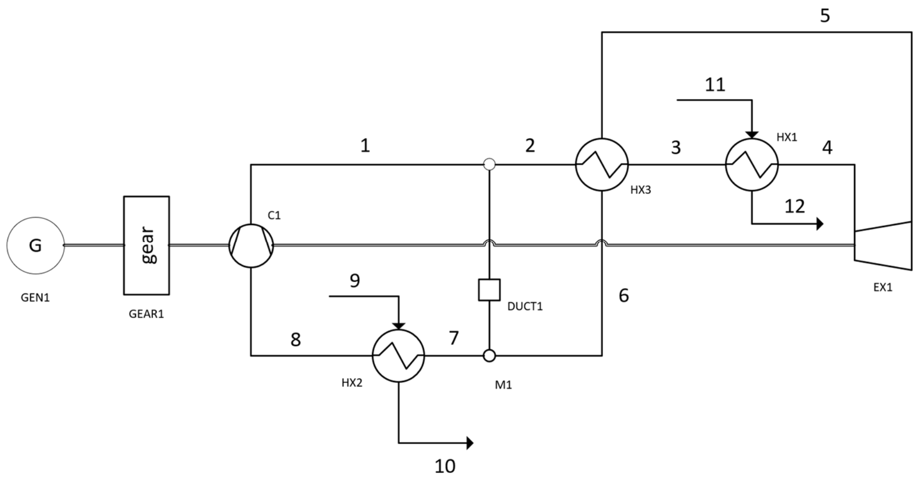

Figure 3: Scheme of the examined cycle.

The upper heat source was simulated by heat exchanger HX1. The set parameters of the added elements are given below:

- Design Method: Cold Side Outlet Temperature
- Desired Cold Side Outlet Temperature: 823.15K
- Second Design Method: Pinch (Hot Out - Cold In)
- Desired Pinch (Hot Out - Cold In): 100 K
- Demand Flow Method: Generate Hot Side Demand Flow
- Cold Side Pressure Loss: 0.01
- Hot Side Pressure Loss: 0.01
- Limits:
  - Minimum Overall Heat Transfer Coefficient: 0.01

The EX1 turbine was attached to the system, connected by a shaft to the compressor. The parameters of the gas turbine were:

- Outlet Pressure: 7,550.25 kPa
- Desired Isentropic Efficiency: 0.9

The default setting is left: Calculate Flow Area Using Inlet Pressure, Temperature, and Flow, which will determine the area of the inlet channel, allowing further off-design calculations. The definition of isentropic efficiency is:

$$\eta_s = \frac{W}{(mh)_0 + (mh)_2 + (mh)_4 - (mh)_{1s}} \quad (5)$$

where the subscripts denote: 0-inlet, 1-outlet, 2-nozzle cooling, 4-rotor cooling, s-isentropic conditions.

The lower source heat exchanger was simulated by HX2 heat exchanger with the following settings:

- Design Method: Hot Side Outlet Temperature
- Desired Cold Side Outlet Temperature: 304.15 K
- Second Design Method: Pinch (Hot In - Cold Out)
- Desired Pinch (Hot In - Cold Out): 11 K
- Demand Flow Method: Generate Cold Side Demand Flow
- Hot Side Pressure Loss: 0.01
- Cold Side Pressure Loss: 0.01
- Number of HTX Passes: 40
- Limits:
  - Minimum Overall Heat Transfer Coefficient: 0.01

A bypass control system was attached to the model, consisting of the DUCT1 flow valve, SP1 flow distributor, and M1 mixer. Duct is an element that allows modeling of pressure and temperature drops in gas streams. A similar effect can be achieved with the Flow Pressure-Temperature Modifier, which additionally allows changing the flow rate.

SP1:

- Primary Port (S12) Control Method: Specify Flow Fraction
- Primary Port Desired Fraction: 1
- Secondary Port (S13) Control Method: Remainder Port

M1:

- Equalize Inlet Pressures Method: Equalize to Pressure of Specified Port

- Controlled Inlet Port for Pressure: Primary (S9)

DUCT1:

- Pressure Method Flag: Outlet Pressure
- Desired Exit Pressure: 7,474.74 kPa

As GateCycle does not allow the insertion of an accumulation tank into the system, a Flow-Pressure-Temperature Modifier was included to simulate it behind the compressor. By forcing a specific flow value, a change in tank filling is simulated.

During off-design operation, the speed of the system changes under the influence of changes in the parameters of the working medium, so a gearbox is required to keep the generator synchronized with the grid. It has been added to the model under the name GEAR1. In the method used, power losses are calculated using equations implemented in the program. In the off-design mode, calculations of gearbox efficiency are identical to those in the design mode, so there is no need to change the settings of this element for further calculations.

To carry out off-design simulations, new model cases were created in which the following elements were set for off-design calculations: EX1, HX1, HX2, HX3, C1, GEN1. In addition, the following settings were made.

HX2:

- Second Design Method: Hot Side Outlet Temperature
- Desired Hot Side Outlet Temperature: 304.15 K
- Demand Flow Method: Generate Cold Side Demand Flow
- Minimum Pinch Limit: 10 K

C1:

- Off Design Pressure Method: Flow Equation with Flow, RPM (Revolutions Per Minute) or CS and Pressure Input
- Desired Outlet Pressure: 24,487.972 kPa
- Off Design Flow/RPM Method: Accept
- Incoming Flow
- Minimum Compressor %CS: 40
- Off-Design Efficiency Method: Use Equation to Calculate Isentropic Efficiency

HX1:

- Second Design Method: Cold Side Outlet Temperature
- Desired Cold Side Outlet Temperature: 823.15 K
- Demand Flow Method: Generate Hot Side Demand Flow

Off-design calculations were carried out to simulate the three basic methods of power control: flow change, bypass, and temperature change. Flow change was induced using the FPT1 element by changing the Desired Outlet Flow value in 5% increments of the nominal flow. This method also allowed the turbine to be topped off with a flow higher than nominal. Bypass control was achieved by changing the Primary Port Desired Fraction value at the splitter, also in 5-percentage point steps. The change in inlet temperature to the turbine was achieved by changing the Desired Cold Side Outlet Temperature in HX1, decreasing it, as in the previous cases, also in 5-percentage point increments. The results with analysis are presented in the following sections.

4 RESULTS

After three series of calculations, the results from all cases were collected and presented in graphs. Figure 4 and Figure 5 how the change in system power as a function of the change in flow through the turbine (with bypass and discharge regulation, labelled BYPASS and FLOW, respectively, on the graph) and the change in inlet temperature to the turbine (with temperature regulation labelled TEMP). As can be seen, the methodology used for flow or temperature changes allowed for a uniform distribution of post-measurement points with a thickening in the direction of decreasing power, which is a desirable phenomenon.

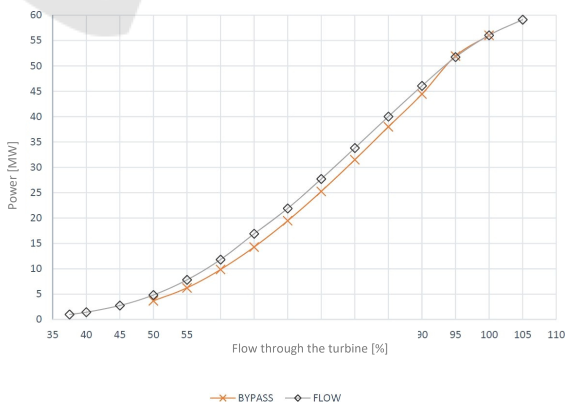

Figure 4: Power as a function of factor output through the turbine.

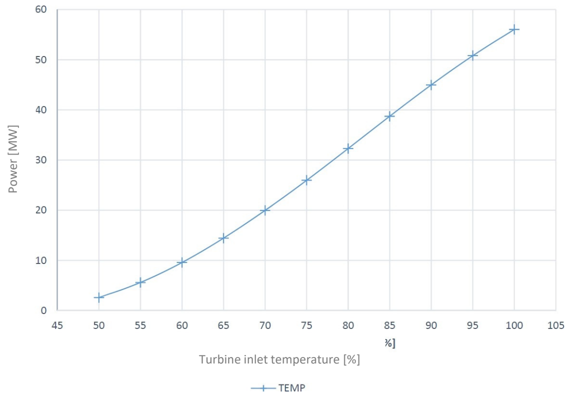

Figure 5: System power as a function of inlet temperature change per turbine.

The main factors in choosing a power control system are efficiency, response time, and cost. The stability of the system's operation under changed conditions is also an important aspect. Analyzing the graph of the efficiency of the system as a function of its power (see Figure 6), it can be concluded that the best in this respect is the flow rate control system. Slightly lower values are achieved by the bypass control system, and significantly lower by the temperature control system.

After three series of calculations, the results from all cases were collected and presented in graphs. Figure 5 and Figure 6 show the change in system power as a function of the change in flow through the turbine (with bypass and discharge regulation, labelled BYPASS and FLOW, respectively, on the graph) and the change in inlet temperature to the turbine (with temperature regulation labelled TEMP). As can be seen, the methodology used for flow or temperature changes allowed for a uniform distribution of post-measurement points with a thickening in the direction of decreasing power, which is a desirable phenomenon.

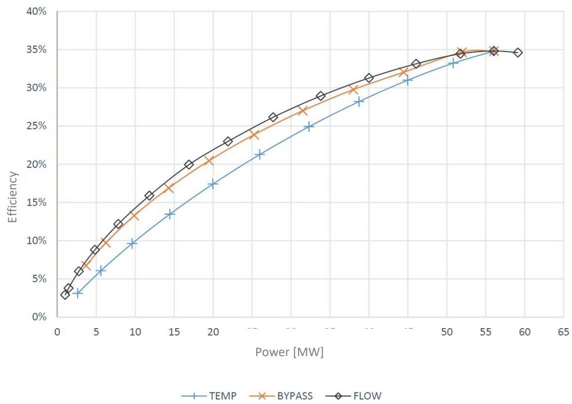

Figure 6: System efficiency as a function of generator power.

The main factors in choosing a power control system are efficiency, response time, and cost. The stability of the system's operation under changed conditions is also an important aspect. Analyzing the graph of the efficiency of the system as a function of its power (see Figure 6), it can be concluded that the best in this respect is the flow rate control system. Slightly lower values are achieved by the bypass control system, and significantly lower by the temperature control system.

The graph from the theoretical part was made for a system for which the working medium was not specified in the source, but it was most likely air or a perfect gas. For this reason, the efficiency curves obtained in the simulation have a distinctly different shape.

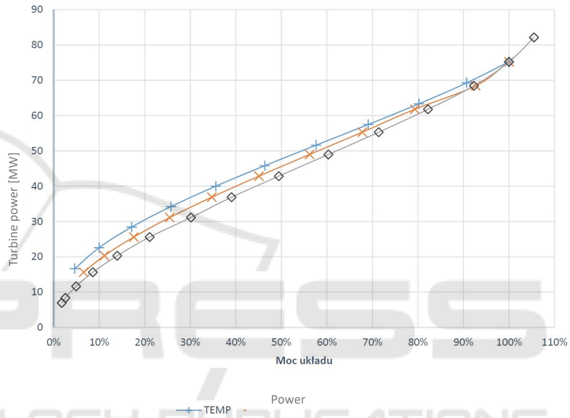

Figure 7: Power generated by the expander as a function of system power.

To further explain this phenomenon, it is worthwhile to analyze the operation of rotating machines under the changed conditions. The expander was analyzed first. Figure 7 shows a graph of the power generated on the turbine as a function of the power of the whole system. As can be seen, the expander generates the highest power with temperature control and the lowest with output control, which is quite the opposite of what would seem consistent with the efficiency graph (Figure 6). Thus, the factor determining the efficiency of the system is the power consumed by the compressor.

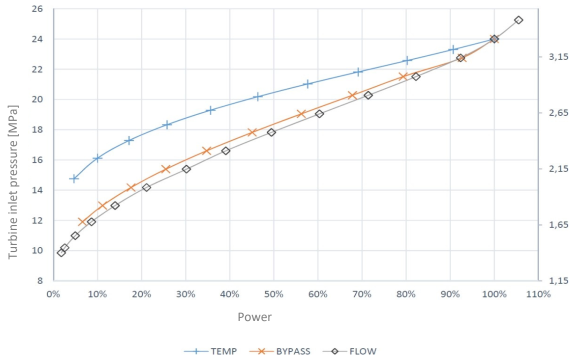

Figure 8: Power generated by the expander as a function of system power.

Considering that the inlet pressure on the compressor, which is the lowest pressure in the system, almost does not change, Figure 8 plots the values of the system compression on the auxiliary axis (on the right side of the graph).

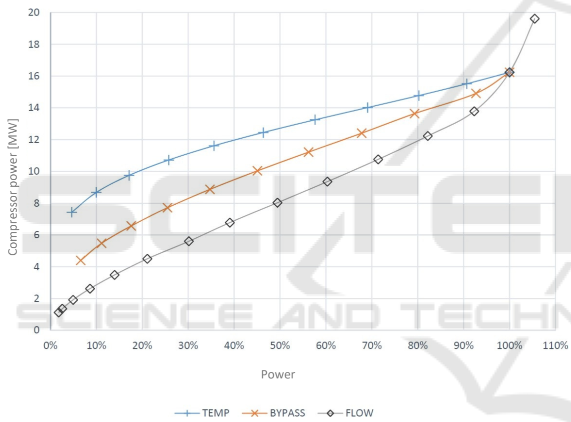

Figure 9: Compressor power as a function of system power.

A more in-depth analysis of individual results reveals significant insights. For instance, Figure 8 shows the power generated by the expander as a function of system power. The expander generates the highest power with temperature control, despite this method resulting in the lowest overall system efficiency (Figure 6). This discrepancy indicates that the power consumption by the compressor is a crucial factor. Figure 9 supports this observation, illustrating that the compressor consumes the least power under discharge control, which correlates with the highest system efficiency.

With all other parameters unchanged, an increase in system compression implies an increase in the power consumed by the compressor. Such a relationship can be seen by comparing the TEMP and BYPASS curves on the graph of compressor power as a function of system power (see Figure 9). It can also be seen that the least power is consumed by the compressor when controlling the discharge. This is due to the reduced flow rate. The graph in Figure 9 also shows that a 5 percentage point increase in flow causes a greater increase in power than a decrease in power with a corresponding decrease in flow.

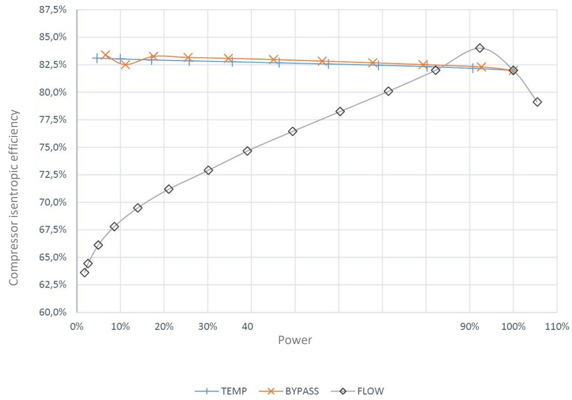

Figure 10: Compressor isentropic efficiency as a function of system power.

Additionally, Figure 10 depicts the isentropic efficiency of the compressor as a function of system power. It is evident that the efficiency is significantly influenced by the discharge method, which minimizes power consumption more effectively than other methods. This is further corroborated by Figure 11 and Figure 12, showing the compressibility factor (Z-factor) at the compressor inlet and outlet. The Z-factor remains relatively stable at the inlet but fluctuates at the outlet, especially under temperature control, indicating complex interactions between the working fluid's properties and system performance.

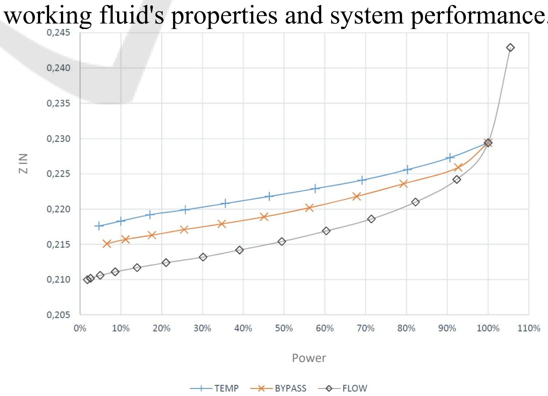

Figure 11: Compressor inlet refrigerant compressor compressibility factor as a function of system power.

A decrease in the power consumed by the compressor does not directly translate into an increase in its isentropic efficiency, as can be seen in Figure 10. The graph of the compressor's isentropic efficiency as a function of system power clearly shows that the decisive influence on this efficiency is due to the change in working refrigerant discharge. The slight increase in compressor efficiency occurring with a decrease in system power is due to the change in compression and average compressibility of the working medium. The GateCycle program does not allow you to read the average value of this factor in a given component, but it does allow you to read its value at the inlet and outlet of the compressor. Based on Figure 11, which shows the Z-factor (Compressibility Factor) at the compressor inlet, it can be concluded that its value hardly changes in this area. Thus, assuming, in accordance with engineering knowledge, the absence of local maxima and minima of this coefficient inside the compressor, it can be concluded that an increase in the value of the Z coefficient at the compressor outlet implies an increase in the average value of this coefficient throughout the component.

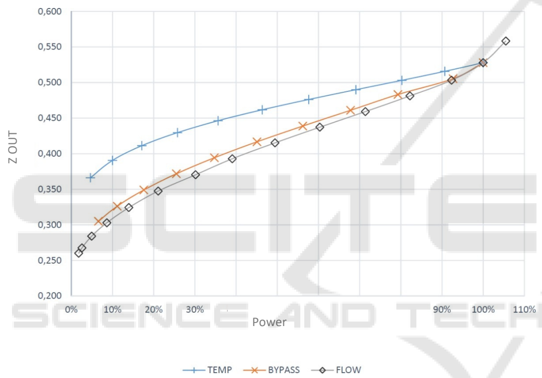

Figure 12: Compressibility factor of the working fluid at the compressor outlet as a function of system power.

Based on Figure 12, it can be concluded that higher values of the Z-factor occur with temperature control. The main effect of the compressibility coefficient on system operation is expressed differently. As already stated, GateCycle determines the off-design compressor speed. In the equation, only the compressibility factor Z changes, which helps explain the shape of the curves on the graph of compressor speed as a function of system power (see Figure 13).

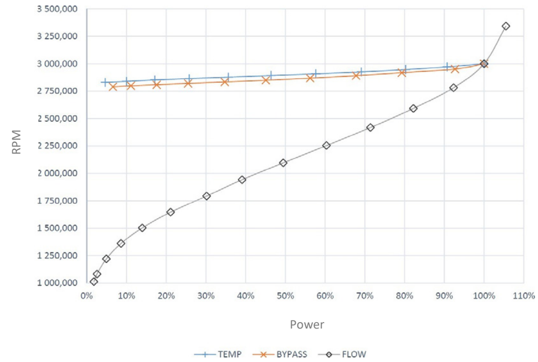

Figure 13: Rotating machine speed as a function of system power.

Figure 14 and Figure 15 highlight the importance of maintaining synchronization with the grid via a gearbox. The gearbox efficiency sharply declines with decreasing system power, emphasizing the need for efficient gearbox design and operation. Moreover, Figure 18 through Figure 20 demonstrate significant variability in heat exchanger performance under different control strategies. Notably, the efficiency of HX2 drops drastically under bypass control, suggesting that operational strategies or design modifications are necessary for optimizing heat exchanger performance.

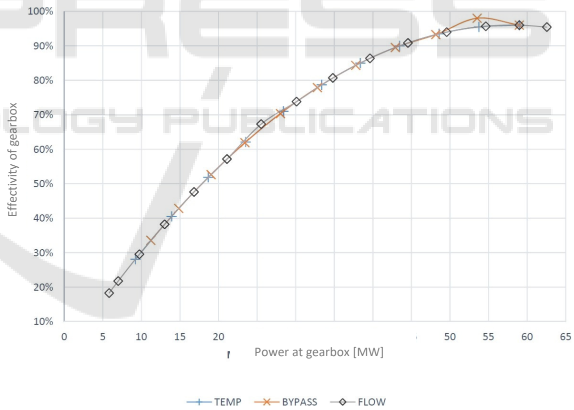

Figure 14: Gearbox efficiency as a function of power delivered at the driving shaft.

As can be seen, especially in the case of output control, the speed changes. To maintain synchronization with the grid, a gearbox is necessary in the system. Its efficiency significantly affects the efficiency of the entire system, as it significantly decreases with a decrease in the power of the system, as shown in Figure 14. Interestingly, the power of the system is the decisive factor affecting the efficiency of the gearbox, and the change in speed does not play a major role. In Figure 14, one point deviation from the plotted curves can be observed, which is probably an iteration error.

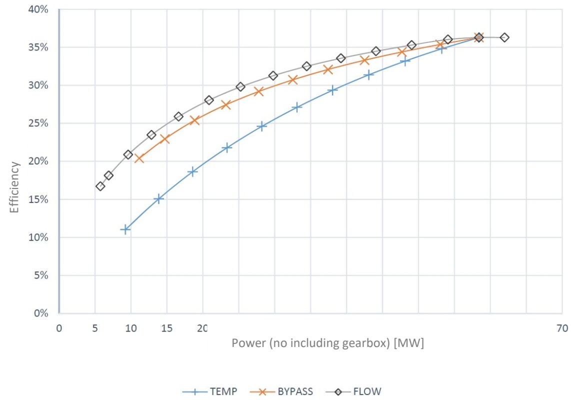

Figure 15: System efficiency as a function of system power at 100% gear efficiency.

It is also possible that the system would operate in island mode and synchronization with the grid would not be necessary. In such a case, after eliminating the gearbox from the system, its efficiency as a function of power would look like in Figure 15.

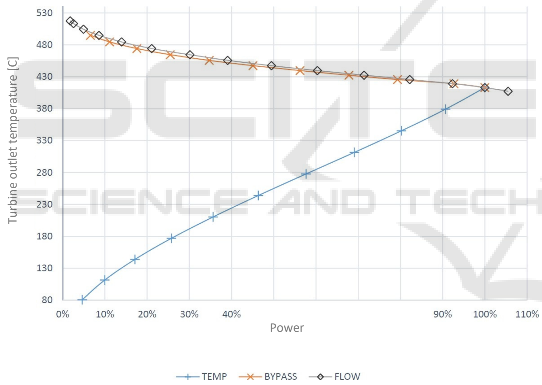

Figure 16: Turbine medium outlet temperature as a function of system power.

As stated earlier, the efficiency of the system is not the only factor in choosing the optimal power control system. The stability of operation is also very important. Figure 17 shows a graph of turbine outlet temperature as a function of system power. As can be seen from it, with bypass control and factor output, as the system power decreases, the outlet temperature increases, while in the case of temperature control, the opposite phenomenon occurs—temperature decreases at three times the speed. Such temperature changes cause changes in the efficiency of heat exchangers, as can be seen in Figure 18, Figure 19, Figure 20, which show graphs of the efficiency of heat exchangers (HX1, HX2, and HX3, respectively) as a function of system power. In the case of output control, the changes are smooth. The same happens with bypass control in exchangers HX1 and HX3 and with temperature control in exchanger HX2. Particularly noteworthy is the sharp drop in the efficiency of the HX2 exchanger with bypass regulation.

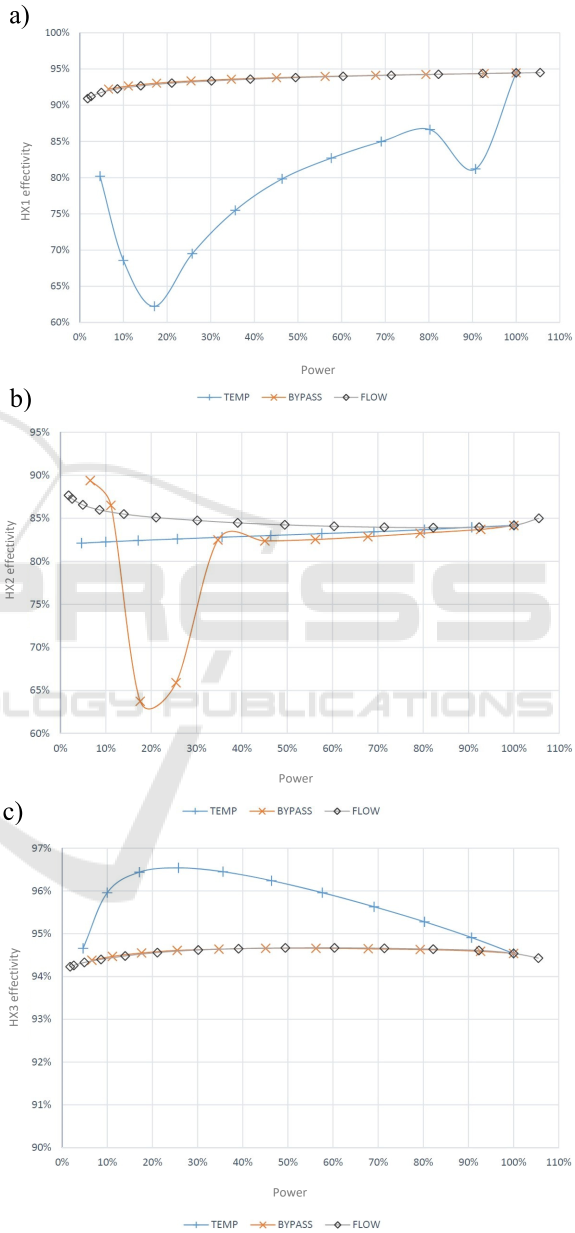

Figure 17: HX1, HX2 and HX3 heat exchanger efficiency as a function of system power, respectively a), b) and c).

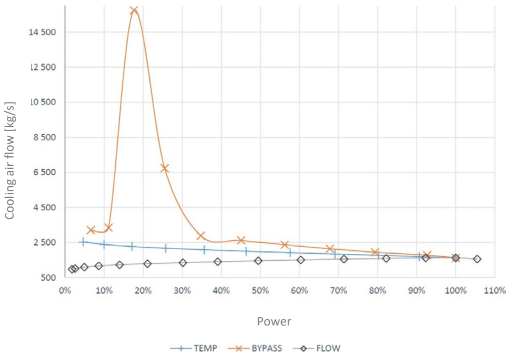

Figure 18: Cooling air flow as a function of system power.

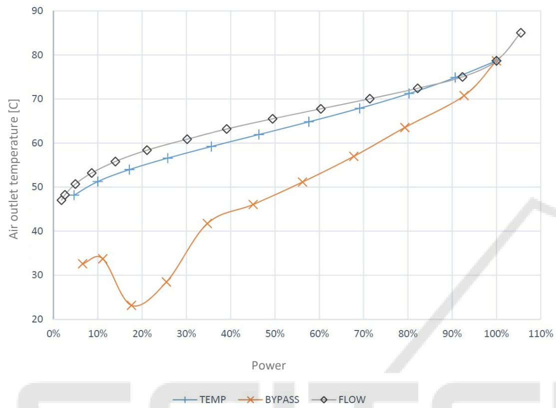

Figure 19: Graph of cooling air outlet temperature as a function of system power.

This large drop in efficiency entails either increasing the cooling airflow or using an additional cooling system, such as water. Figure 18 shows a clear jump in the 10-30% area. On the other hand, the advantage of controlling the output is that the required cooling airflow decreases with a drop in system power. Additionally, Figure 19shows a graph of the cooling air outlet temperature. It can be deduced from it that the scavenging heat is difficult to use due to the low temperature, not exceeding 80°C.

The intricate dynamics and performance of control systems in a supercritical carbon dioxide gas turbine circuit, as delineated in the results of this study, provide a profound understanding of the efficiency and operational stability under various control strategies. The discussion herein elaborates on the salient findings from the experimental data, interprets their practical implications, and delineates potential avenues for future investigations to enhance system design and efficiency.

The experimental results vividly demonstrate that flow rate control systems consistently maintain higher efficiency compared to other control mechanisms under variable operational conditions. This observation is substantiated by the data presented in Figure 6, where flow rate control systems showed superior performance in terms of system efficiency. In contrast, temperature control systems exhibited a significant decline in efficiency, suggesting their limited applicability in conditions where maintaining efficiency is critical. These findings challenge some of the conventional theories that advocate the versatility of temperature control systems across different operational parameters, suggesting a need for a more nuanced understanding of these systems in practical applications.

One of the more compelling insights from the study is the relationship between compressor power consumption and the overall efficiency of the system. Detailed in Figure 10, the results indicate an inverse relationship between flow rate reduction and power consumption by the compressor. Interestingly, this reduction in power does not correlate with an increase in the isentropic efficiency of the compressor, contrary to typical thermodynamic expectations. This anomaly points to a complex interaction between the flow rate and the thermodynamic properties of the supercritical CO_2, which may not be adequately captured by existing theoretical models. This suggests that current models may need to be revised or expanded to incorporate these nuanced interactions more accurately.

The study also highlighted the critical role of the compressibility factor (Z-factor) in system performance, particularly its minimal changes at the compressor inlet and more pronounced fluctuations at the outlet, as depicted in Figure 11 and Figure 12. These findings emphasize the delicate balance required in managing the working fluid's thermodynamic properties to optimize system performance. It also underscores the potential for significant performance enhancements through focused modifications in the handling and treatment of the working fluid under different operating conditions.

Another key area of focus is the efficiency of the gearbox, which exhibits a substantial decline with decreasing system power (Figure 14). This decline in efficiency does not strongly correlate with changes in the system's speed, indicating potential areas for improvement in gearbox design or operation. This could include exploring new materials, design modifications, or even entirely new gearbox technologies that could reduce mechanical losses and enhance overall system efficiency.

The performance of heat exchangers under different control strategies revealed significant variability, as shown in Figure 17. Particularly notable is the drastic drop in efficiency of HX2 under bypass control, necessitating a reconsideration of either the operational strategy or the design of the heat exchanger itself. Optimizing heat exchanger performance could involve detailed analysis of fluid dynamics within the exchangers or even the introduction of new materials or technologies that could withstand the operational demands of different control strategies more effectively.

Overall, these detailed analyses underscore the intricate dynamics within the supercritical CO2 gas turbine system and the necessity for carefully selecting and optimizing power control strategies to achieve the best performance outcomes.

5 DISCUSSION

Based on the findings of this study, several areas for future research emerge. First, there is a clear need for more refined theoretical models that incorporate the real gas effects observed under extreme conditions more accurately. This could involve computational fluid dynamics (CFD) simulations or experimental studies to better understand the interactions at play. Additionally, the exploration of alternative materials for compressors and heat exchangers could lead to improvements in system efficiency and durability. Finally, the integration of advanced predictive maintenance technologies could significantly enhance operational stability and extend the lifespan of system components.

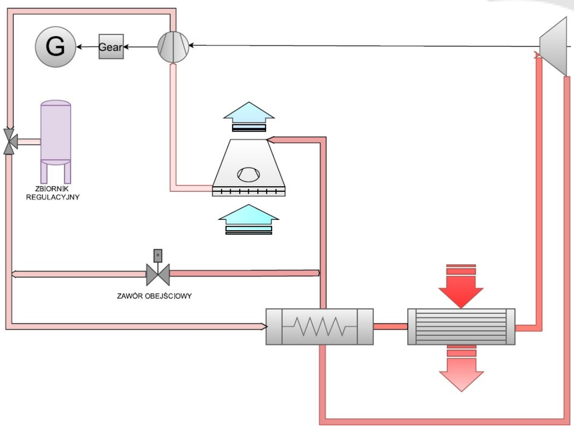

Figure 20: Schematic of the recommended layout.

The proposed hybrid approach in Figure 20 integrates both discharge and bypass controls, allowing adaptive management of power control. This dual system compensates for the limitations of each method under varying operational conditions, offering a balanced solution that enhances system responsiveness while maintaining high efficiency. The schematic in Figure 23 represents this hybrid system, which is poised to address the trade-offs observed in single-method control systems, offering an optimal path forward for supercritical CO2 gas turbine applications in Concentrating Solar Plants.

The insights from this study provide a robust framework for understanding the operational dynamics of supercritical CO2 gas turbines under different control strategies. While the current control mechanisms offer substantial benefits, there is considerable scope for improvement in system design and operational protocols. The research conducted herein sets the stage for further innovations and enhancements in this critical field of energy technology.

6 CONCLUSIONS

This investigation developed a model of a supercritical carbon dioxide gas turbine system, focusing on power modulation within the system. The model parameters were meticulously selected for application in next-generation power facilities, particularly those employing Concentrating Solar Plants technology. The model was designed for simplicity and high efficiency, incorporating a heat regeneration system.

Three distinct power control strategies were evaluated: bypass, flow rate, and turbine inlet temperature control. Extensive calculations and analyses determined that turbine inlet temperature control yielded the least favorable outcomes, with significantly lower efficiency and longer response times.

The discharge control system demonstrated marginally higher efficiencies but required significant changes in machinery speed and the addition of an accumulation tank, increasing cost and complexity. The bypass control system offered the fastest response times and ease of implementation but required increased cooling airflow at lower power outputs.

The optimal solution is a hybrid approach integrating both discharge and bypass controls. This dual system allows adaptive power management, with each method compensating for the other's limitations under varying conditions. This proposed solution is diagrammatically represented in Figure 20.

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