

Enhanced dynamic simulation approach towards the efficient mining thermal energy supply with improved operational flexibility

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Abstract: This paper presents a thermal power plant retrofitting approach focused on improvements in the operational flexibility of existing combined cycle power plants dedicated to providing thermal energy for medium and low-temperature processes in copper mining facilities. The main motivation for this research was aimed at evaluating the operational flexibility of the electrical industry through sector coupling and its effect on solving the energy sector decarbonization issues. The research evaluates the advantages of hybridization systems for supporting the electrical and mining industries to better predict operations. The proposed approach is based on a dynamic simulation scheme that finds the optimal operating parameters of the combined heat and power (CHP) system, such as location, type, and arrangement of each component of the CHP system. The power plant dynamic simulation model was validated against data available in the literature; it was also characterized by real operational data of the San Isidro II power plant installed in Chile. Several alternatives for the cogeneration plant location, as well as the splitter system design, were investigated and then compared. A cogeneration plant design with two heating modules was selected based on the comparative study performed in this work and its CHP system was evaluated for a load reduction case study. The results were compared against a reference model. The proposed CHP system exhibited improved performance: a minimum of 15% of the exhaust gases are required to supply the thermal energy demand of the electrowinning process when a full load is considered. It was also found that an average decrease of 5% of the mechanical power at each steam turbine stage noted. Finally, the proposed CHP system's average thermodynamic efficiency is found to be 19% greater than the power plant average efficiency. Consequently an average decrease of 32,500 tons of carbon dioxide emissions per year is predicted.

Keywords: Power plant retrofitting, Dynamic simulation; Operational flexibility; Combined heat and power; Copper mining; Electrowinning; OpenModelica simulations.

1. Introduction

Worldwide, electric power systems are undergoing large structural changes. Commonly, these systems have been based on centralized models in which the electric power generation using fossil fuels prevailed. Recently, electric power systems are evolving towards liberalized energy markets in which the electricity demand tends to be supplied by renewable energy sources [1]. Also, the inherent intermittency of these power generation sources, coupled with the electric power demand variability, lead to improved response and adjustment capabilities of the electric power systems. According to [2-4], these capabilities are known as “power system operational flexibility”, aiming to get a balance between

generation and demand at all times. As reported by the International Renewable Energy Agency (IRENA) [5], the operational flexibility of the electricity system must consider effective management strategies for all sectors including thermal and electrical storage, demand management, coupled systems, and operational management of generation, transmission, and distribution systems.

In this context, the generation sector plays a key role in the enhancement of electric power system operational flexibility. At the plant level, operational flexibility can be implemented [6] by using advanced simulation, optimization, and control techniques focused to minimize the operating times and maximizing the system's capabilities to work under cyclic operating conditions and peak loads. Likewise, the power system operational flexibility based on the generation sector not only must consider the traditional concept of flexibility, but should also evaluate other innovative strategies that significantly increase the electric power system flexibility, such as sector coupling. According to [5], sector coupling is the interconnection process of the electric power sector with the broader energy sector to produce heat and hydrogen or the charging of electric vehicle batteries from the energy generated in the power plants. In this sense, coupling the energy demand for the heat through power-to-heat should significantly enhance system flexibility and lead to a decrease in greenhouse gas emissions from the end-use sectors [5].

To enhance the electric power system operability, different flexibility alternatives should be considered, coupled, and evaluated. One of them is the retrofitting approach of existing power plants through cogeneration systems or combined heat and power systems (CHP) generating electricity and using heat that would otherwise be wasted to produce thermal energy for residential or industrial purposes. Likewise, capabilities to adjust demand to respond during periods of supply shortages and over-generation are both incorporated into the system. One advantage of this approach is that it can reduce emissions and operational costs, as well as increase power system reliability and thermodynamic efficiencies.

2. Background context

Research on the sector coupling for power to heat has been addressed from different points of view. Most of these studies are generally dealing with District Heating and Cooling (DHC) applications for countries with long heating periods and growing cooling requirements during summer seasons. The developments presented in these studies are interesting for both problem formulation and solutions, as well as the detection of advantages and issues of CHP plants. In this context, Sdringola et al. [7] proposed a management profile, enhancing both the operational and economic parameters of a small CHP plant. The CHP plant was designed to supply electricity, heating, and cooling through a district network based on monitored consumption of electricity, heating, and cooling, constraining the energy fluxes. Rolfman [8] presented an operational performance study of a district heating system whose aim was to maximize electricity production during periods of high electricity prices. The case study was developed using a mixed-integer linear programming model. Besides, Rakopoulos et al. [9] studied the technical, economic, and environmental performances of different configurations of CHP systems for use in district heating (DH) networks based on lignite-fired power plants. For summer and winter operational modes, the thermal cycle and plant efficiency were computed by using a thermal cycle calculation tool specifically developed for this case [9]. The performed calculations showed an important increase in electrical and thermal efficiencies in the pre-dried lignite firing case as well as considerable fuel savings and CO₂ emissions reductions.

In the same way, research such as the one presented by [10], [11], and [12] addresses the CHP plant operational management problem as an optimization problem. In [10], the

authors focused on solving the problem of optimal generation scheduling of CHP economic dispatch (CHPED) considering operational and economical constraints of the CHP plant, as well as the constraints of demand and system. Likewise, a review on the implementation of different heuristic methods for the solution of CHPED was proposed with contributions of each implemented heuristic method. The authors compared the optimal solution of the CHPED problem in terms of minimum operational cost and computational time. Regarding [11], the CHP economic dispatch problem is tackled as an optimization problem focused on minimizing the fuel cost of CHP units considering operational constraints. The whale metaheuristic optimization method is proposed and tested considering the valve-point impact of conventional thermal plants, capacity limits of the generation units, and heat power dependency constraints of the cogeneration units. Two cases were evaluated and reported results were compared with the reported solutions in recent studies, in which the improvement in the annual saving for both test systems was significant. As for the research of [12], a multiplayer harmony search method (MPHS) is proposed to solve the problem of CHPED. The MPHS method was employed on a 24-unit and 84-unit test systems to evaluate its performance in finding the optimal solution of the CHPED and handling different constraints of the problem including the valve-point loading effect of thermal plants and CHP units. Using this methodology, it was possible to identify the feasible operating zone of each type of CHP unit; results reported show an improvement in annual cost savings. Likewise, MPHS is compared with other optimization techniques obtaining optimal solutions using fewer iterations.

According to [13], existing thermal power plants that currently operate as power units can be retrofitted to deliver heat and power simultaneously, reaching efficiencies of up to 90%. This retrofit is usually aimed at extracting heat at a higher temperature in the steam turbine. The main issue to adapt this power plants to CHP systems consists in the amount of power that could be delivered. This means that the higher the required temperature of the coupled system, the lower the electrical efficiency of the system.

Looking at industrial applications, Gambini et al. [14] presented a techno-economic feasibility analysis of high-efficiency CHP plants to be coupled with the Italian paper industry. According to this analysis, gas turbines were found to be the most appropriate technology for paper industry processes among the proposed CHP alternatives. Similarly, Danon et al. [15] presented a techno-economic analysis of a CHP plant based on wood residues from the Serbian wood industry. The cost of electricity generation was assessed using five different kinds of CHP technology suitable for the wood industry. Besides, Ahmadi et al. [16] presented a feasibility study of a combined heat and power (CHP) plant in a paper mill installed in Iran. Energy efficiency, total cost rate of the system products, and CO₂ emissions of the whole plant were the main performance indicators considered for the feasibility evaluation of the proposed design of the cogeneration plant.

According to the International Copper Study Group [17], Chile has established itself as the largest copper producer in the world. Copper extraction represents the largest contribution to the Chilean economy with around 10% of gross domestic product [18]. Likewise, the Chilean mining industry is the pioneer in copper refining processes based on copper leach, solvent extraction and electrowinning technologies [19]. Through these processes, high-purity copper cathodes can be obtained without the need for prior smelting stages, using process heat at medium-low temperatures. Electrowinning is significantly important among copper processing technologies since it is used to produce high-purity metal on large-scales and at a reasonable cost [20]. Currently, almost all energy required by electrowinning processes is provided by boiler technologies based on low efficiency and highly polluting, significantly contributing to the negative carbon footprint of the country. Lately, a couple of copper mine complexes in Chile have implemented solar assisted heating systems to pre-heat the electrolyte solution before considering electrowinning cells

[20]. Similarly, some studies focusing on solar metallurgy are being conducted in order to improve production capabilities and similar applications.

On the other hand, although CHP systems based on gas turbines are increasingly being used, their operational capacity for industrial heating processes is only considered limitedly effective since it is re-using only a small fraction of its total thermal wasted energy. Similarly, the applications of CHP systems from gas turbines for the mining industry and their designs based on dynamic simulation modelling have not yet been reported in the literature. Hence, this research aims to determine the viability and applicability of CHP systems based on gas turbine applicable to industrial copper mining processes. Unlike those reported in the literature, this paper presents a dynamic simulation approach for the design of a CHP system based on energy recovery from the hot exhaust gas of an existing combined cycle power plant. The electrowinning process of a copper complex in Chile was selected to supply thermal energy since is a medium-low temperature continuous process that allows maximising the efficiency of the CHP system. The coupled system performance and behaviour are modelled through a simulator developed in the well-known OpenModelica environment. The numerical results are used to analyse the performance, determining the CHP system's optimal design and configuration. This paper is part of a comprehensive strategy to enhance the operational flexibility of conventional thermal power plants based on an integrated approach that encompasses the synthesis of optimum operating procedures of high efficiency conventional thermal power plants and the sector coupling to supply useful thermal energy. The development of this approach involves metaheuristic optimization algorithms, dynamic simulations models, response surface models, and machine learning models.

3. Proposed methodology

3.1. Problem statement

Determining the optimal location, operating parameters, and configuration of a combined heat and power (CHP) system that supplies efficiently electrical power to the grid and thermal energy for an electrowinning process of a copper mining complex is the main goal of this approach. This problem is addressed as a design problem involving the development of a CHP system dynamic simulation model capable of determining the state variables changes in time. The CHP system design aims mostly on the efficient exploitation of the gas turbine hot exhaust gas using a splitter system located between the gas turbine and Heat Recovery Steam Generator (HRSG).

Cogeneration plants or CHP systems are an integrated system that combine a thermal power plant and a process steam plant to provide electrical power and useful thermal energy simultaneously. Thermodynamically, cogeneration based on combined cycle power plants provides the most efficient use of fuel: heat remaining in the exhaust gases from the gas turbine is captured by the heat recovery steam generator to produce electricity and by the process steam plant to generate thermal energy for domestic or industrial purposes [21].

According to Al-Shemmeri [22], a typical configuration of a CHP system consists of three basic components; (a) a primary engine in which the fuel is converted into mechanical and/or thermal energy, (b) an electric generator or alternator to transform mechanical energy into electricity and (c) a heat recovery system to collect the heat produced. The operating scheme of a CHP system based on energy recovery from the hot exhaust gas is shown schematically in Figure 1.

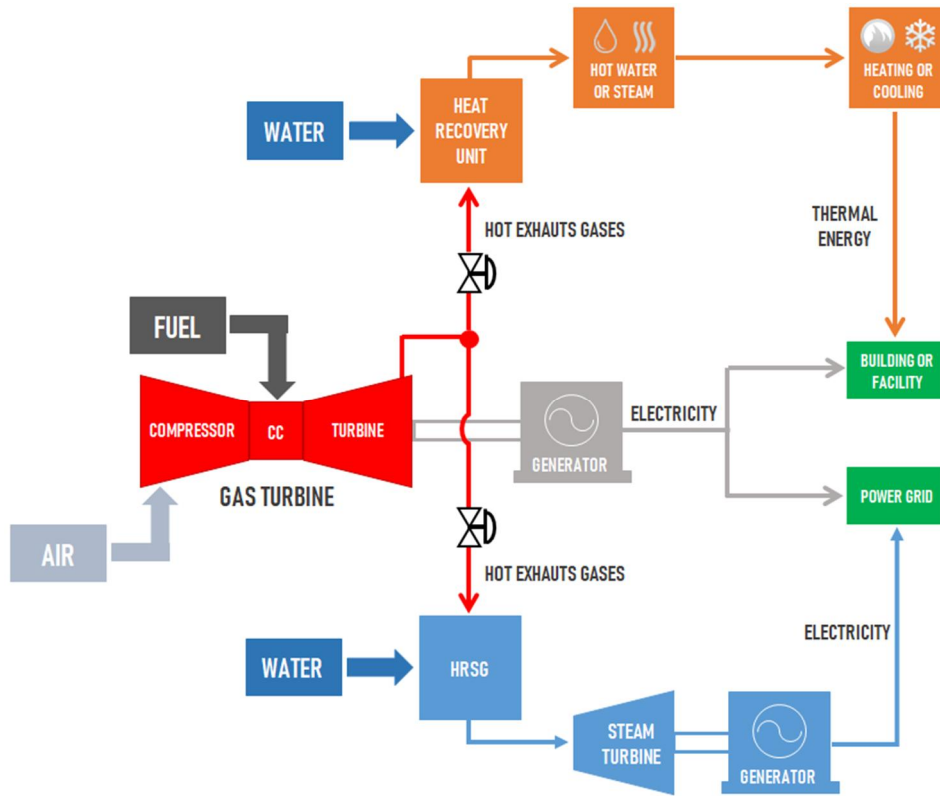


Figure 1. Operating scheme of a CHP system based on energy recovery from the hot exhaust gas.

The section corresponding to electrical power generated by a gas turbine is shown in a grey colour. The exhaust gases can be used to produce electrical power and thermal energy independently. Additional electric power can be generated from the hot exhaust gas through a HRSG system, a steam turbine, and an electric generator, which is known as a combined cycle power plant (presented in blue colour). Otherwise, thermal energy can be produced through a heat exchanger system using the hot exhaust gas available energy (showed in orange colour).

3.2. Proposed approach

The problem addressed in this research is part of a comprehensive strategy to enhance the operational flexibility of conventional thermal power plants to deal with the challenges of liberalized markets. This strategy is managed through an integrated approach that encompasses the synthesis of optimum operating procedures of high efficiency conventional thermal power plants and the sector coupling to supply useful thermal energy as shown in Figure 2.

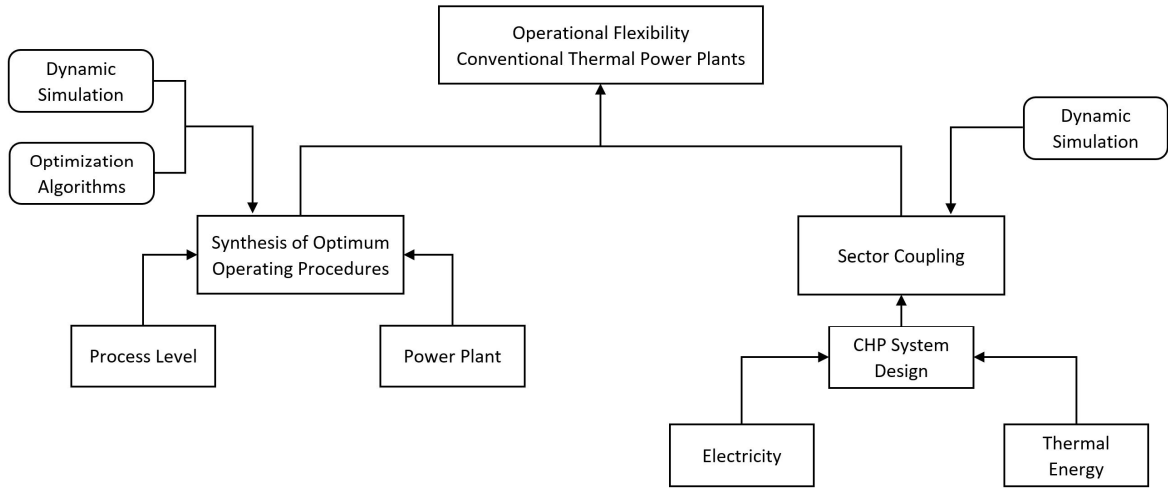


Figure 2. Integrated approach strategy to enhance the operational flexibility of conventional thermal power plants.

The sector coupling-problem has been formulated as a design problem in order to determine the optimal operating parameters of the system, location, type, and arrangement of each component of the CHP system. Likewise, considering that in the operations of both the thermal power plant and the CHP system, the state variables change in time, dynamic simulation models are proposed as design strategy. Dynamic simulations models must be modular, reconfigurable, and expandable: modular since they can use interchangeable components, reconfigurable by the capability to create new components based on existing models, and finally expandable as components or systems can be added to create more complex simulation models.

The proposed design process for the sector coupling system based on dynamic simulation models is shown in Figure 3.

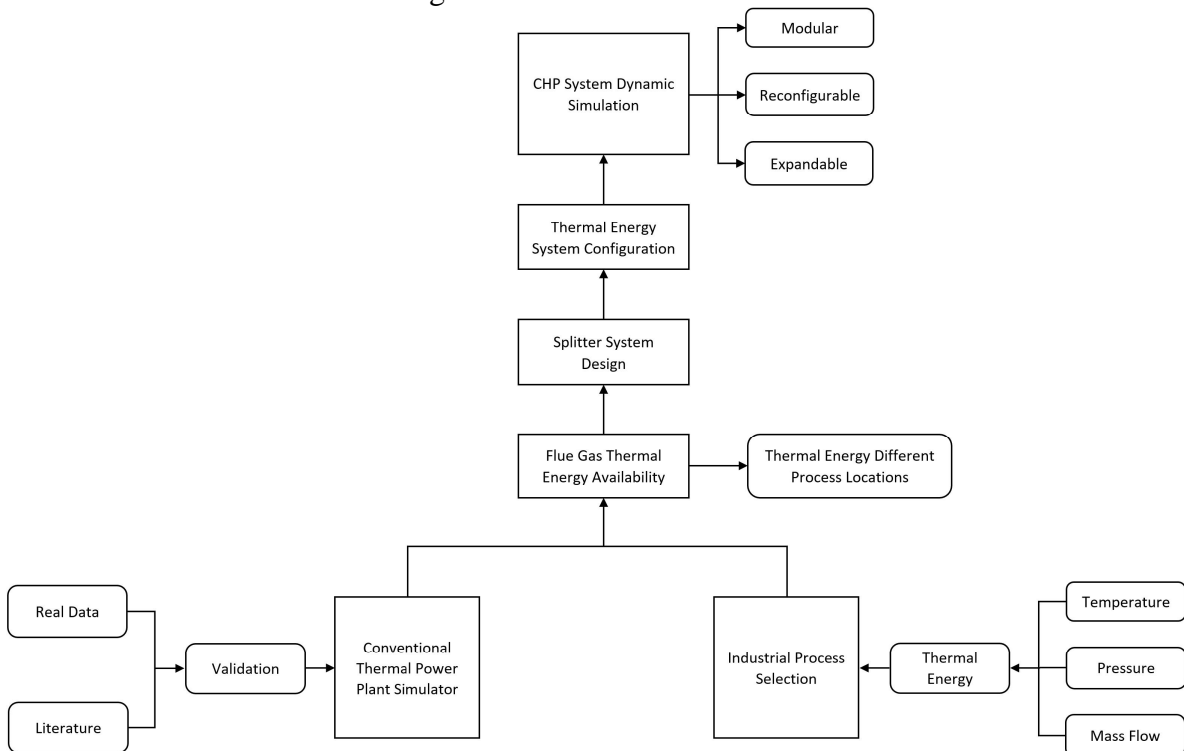


Figure 3. proposed design process for the sector coupling system based on dynamic simulation models.

The design process begins with the development of a power plant dynamic simulation model which is validated against real operational data and results available in the literature. Thereafter, industrial processes needing thermal energy and their operational features are selected. Subsequently, the availability of thermal energy from the flue gas for different locations in the process is quantified and the system location feasibility is evaluated in terms of industrial process global thermal energy requirements. In this phase of the process design, the information of the thermal energy available in the flue gas is stored for later use. In agreement with the results, the splitter system configuration is proposed in the selected location. Next, the thermal energy system simulation model with different configurations and arrangements is developed and tested. Additionally, a coupled simulation model of the power plant, splitter system, and thermal energy system is also tested. Finally, to find the optimal operational parameters and configurations of each CHP system components that satisfy the electricity and thermal energy requirements during transient operations of the system, a design iterative process based on the CHP dynamic simulation model is carried out. Since the simulation model of the CHP system is modular, reconfigurable, and expandable, based on stored information, different industrial processes or combinations of them can be evaluated using: a) modification of the thermal energy requirements, b) the location of the splitter system, and c) the configuration of the thermal energy system. In the same way, systems with different applications such as thermal energy storage could be developed by exchanging components, creating new sub-models, and adding new systems.

3.3. Simulation model

The baseline configuration of a gas turbine exhaust gas derivation CHP system in a combined cycle power plant is shown schematically in Figure 4. Module I represents the original configuration of the combined cycle power plant consisting of a gas turbine, a heat recovery steam generator, and a steam turbine. In this module, electricity is generated from fuel thermal energy by two thermodynamic cycles. The first one corresponds to a Brayton Cycle, where the combustion of natural gas produces work on the shaft in the gas turbine. Furthermore, high-temperature exhaust gases from the gas turbine are used by a heat recovery steam generator to produce high pressure and temperature steam (Rankine Cycle), generating electricity using a steam turbine. Part of the high-temperature exhaust gases at the gas turbine outlet are bypassed as the heat source to produce the required thermal energy for the needs of the industrial processes. Individual models of combined cycle power and cogeneration plants are described in sections 3.3.1 and 3.3.2.

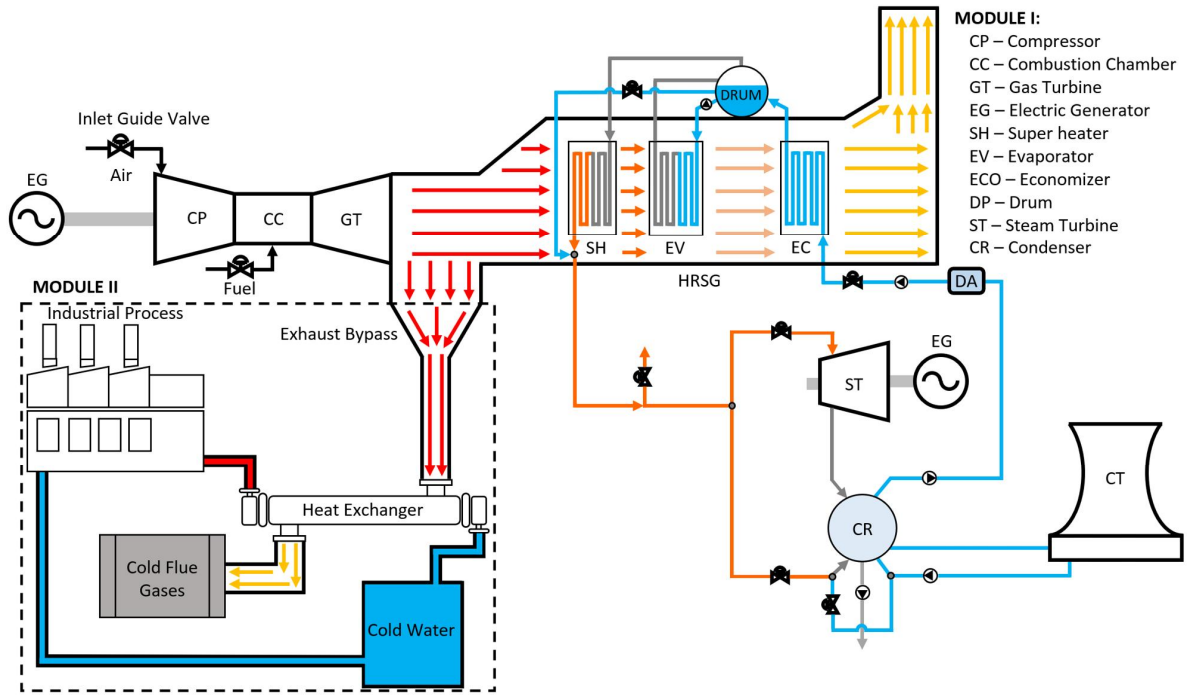


Figure 4. The basic configuration of combined heat and power systems based on high-temperature exhaust gases.

3.3.1. Combined cycle power plant

The one-dimensional models (or three-equation flow) are used to understand the physical behaviour of systems using the standard laws of physics complemented with physical correlations [23]. In power plant simulations models applications they are considered ideal as it is only required to monitor a small number of characteristic variables of the system: those variables are generally called variables of interest. The one-dimensional thermohydraulic models are represented by mass, momentum, and energy conservation equations of the mixture that describe the steady-state and dynamic behaviour of the characteristic variables [24]. The dynamic behaviour of characteristic fluid variables such as the local pressure, the total mass flux, and the temperature or the enthalpy, is described by the three-partial differential conservation equations of the mixture [24].

The mass conservation is written as:

$$\frac{\partial(\rho_m)}{\partial t} + \frac{\partial(\rho_m u_m)}{\partial z} = S_m \quad [1]$$

The symbols ρ_m , u_m and S_m represent the density, the fluid velocity, and the injection or leakage of the mixture.

The momentum conservation is defined as:

$$\begin{aligned} \frac{\partial(\rho_m u_m)}{\partial t} + \frac{\partial(\rho_m u_m^2)}{\partial z} + \frac{\partial}{\partial z} \left(\frac{X \rho_s \rho_l}{(1-X) \rho_m} V_s^2 \right) \\ = -\frac{\partial p}{\partial z} + F_{gra} + F_{wal} + f(val + form + \rho u) \end{aligned} \quad [2]$$

In this equation, the symbol X is the void fraction of the gas phase, ρ_s and ρ_l denote the density of gas and liquid phase, respectively. The V_s is the drift velocity of the gas phase concerning to the volumetric centre of the mixture. F_{gra} and F_{wal} represent the gravitational

acceleration force per volume and the friction force per volume. The function f takes into account the pressure losses due to valve and friction phenomena, ρ and u refer to the density and longitudinal velocity of the fluid.

The energy equation of the mixture is expressed as:

$$\frac{\partial(\rho_m h_{0,m})}{\partial t} + \frac{\partial(\rho_m u_m h_{0,m})}{\partial z} + \frac{\partial}{\partial z} \left(\frac{X \rho_s \rho_l}{\rho_m} V_s (h_{0,s} - h_{0,l}) \right) = \frac{\partial p}{\partial t} + q_{wal} \quad [3]$$

Here the symbol q_{wal} represent the heat flow through walls per volume, and $h_{0,l}$ and $h_{0,s}$ are the static enthalpy of the liquid and gas phase, which includes the kinetic energy of the flow.

It is important to consider that the operating transient process of the power plant involves often nonlinear dynamics with multiple steady-state operating points. The governing equations that describe the steady-state of each component of the combined cycle power plant model are presented as follows:

▪ Air Compressor

The compressor work rate is a function of air specific heat C_{pa} , air mass flow rate \dot{m}_a , the temperature a difference between the compressor inlet and outlet $T_{OC} - T_{IC}$, compressor pressure ratio $\frac{P_{OC}}{P_{IC}}$, compressor isentropic efficiency η_c and can be expressed as follow:

$$\dot{W}_C = \dot{m}_a \cdot C_{pa} \cdot \left(\left(T_{IC} + \frac{\left(\left(\frac{P_{OC}}{P_{IC}} \right)^{\frac{\gamma_a - 1}{\gamma_a}} - T_{IC} \right)}{\eta_c} \right) - T_{IC} \right) \quad (4)$$

where γ_a is the air specific heat ratio and C_{pa} represents a function of temperature according to Equation 5 [25]:

$$C_{pa}(T) = 1.048 + \left(\frac{3.83T}{10^4} \right) + \left(\frac{9.45T^2}{10^7} \right) - \left(\frac{5.49T^3}{10^{10}} \right) + \left(\frac{7.92T^4}{10^{14}} \right) \quad (5)$$

▪ Combustion Chamber

From the combustion chamber energy balance, the fuel flow required by the cycle is calculated according to the following relationship:

$$f = \frac{C_{pg} \cdot T_{OC}}{LHV \cdot \eta_{cc} + C_{pf} \cdot T_f + C_{pcg} \cdot T_{OCC}} + \frac{\dot{m}_{steam}}{\dot{m}_{air}} \cdot \frac{C_{psteam} \cdot T_{steam}}{LHV \cdot \eta_{cc} + C_{pf} T_f + C_{pcg} T_{OCC}} \quad (6)$$

where C_{pg} is the combustion gases specific heat, T_{occ} is the temperature of flue gases at the outlet of the combustion chamber, LHV is the lower heating value of the fuel, η_{cc} is the efficiency of the combustion chamber, C_{pfuel} is the specific heat capacity of the fuel, T_f is the temperature of the fuel at the inlet of the combustion chamber, \dot{m}_{steam} is the mass steam flow at the combustion chamber inlet, C_{psteam} is the specific heat capacity of the steam and T_{steam} is the temperature of the steam at the inlet of the combustion chamber.

▪ Gas Turbine

The gas turbine output power can be expressed as a function of its input and output temperatures, isentropic efficiency and pressure ratio as follows:

$$W_{GT} = \dot{m}_g \cdot C_{pg} \cdot \left(T_{OCC} - T_{OCC} \left(1 - \eta_{GT} \left(1 - \left(\frac{P_{OCC}}{P_{OGT}} \right)^{\frac{1-\gamma_g}{\gamma_g}} \right) \right) \right) \quad (7)$$

where \dot{m}_g is the gas turbine flow rate which is the summation of air mass flow rate and fuel, T_{OCC} is the temperature of combustion gases exiting the combustion chamber, η_{GT} is the gas turbine isentropic efficiency, P_{OCC} is the pressure of combustion gases exiting the combustion chamber, P_{OGT} is the pressure of the outlet hot gases exiting the gas turbine, γ_g is combustion gases specific heat ratio while the combustion gases specific heat C_{pg} is a function of temperature according to [25] and expressed by Equation 8.

$$C_{pg}(T) = 0.991 + \left(\frac{6.997T}{10^5} \right) + \left(\frac{2.712T^2}{10^7} \right) - \left(\frac{1.2244T^3}{10^{10}} \right) \quad (8)$$

▪ Heat Recovery Steam Generator

The energy balances for the water/steam cycle and combustion gases, allow to determine the gas temperature and water properties according to the expressions below:

$$\dot{m}_g C_{pg} (T_{IGSH} - T_{OGSH}) = \dot{m}_{S,SH} (h_{OSSH} - h_{ISSH}) \quad (9)$$

$$\dot{m}_g C_{pg} (T_{IGEV} - T_{OGEV}) = \dot{m}_{S,EV} (h_{OSEV} - h_{ISEV}) \quad (10)$$

$$\dot{m}_g C_{pg} (T_{IGEC} - T_{OGEC}) = \dot{m}_{S,EC} (h_{OSEC} - h_{ISEC}) \quad (11)$$

where $\dot{m}_{S,SH}$, $\dot{m}_{S,EV}$ and $\dot{m}_{S,EC}$ are the mass flow rate in the superheater, evaporator, and economizer, respectively. Meanwhile, T_{IGSH} , T_{IGEV} and T_{IGEC} are the respective temperatures of combustion gases entering the superheater, evaporator, and economizer. In addition, T_{OGSH} , T_{OGEV} and T_{OGEC} are the temperatures of the combustion gases exiting to superheater, evaporator, and economizer. On the other hand, (h_{OSSH}, h_{OSEV}) , (h_{OSEC}, h_{ISSH}) and (h_{ISEV}, h_{ISEC}) are the steam enthalpies at the inlet and outlet in the superheater, evaporator, and economizer, respectively.

▪ Steam Turbine and Condenser

For the case of the steam turbine, the output power is expressed in terms of water mass flow rate \dot{m}_w , superheated steam enthalpy entering the steam turbine h_{IST} and outlet steam enthalpy from the steam turbine h_{OST} according to:

$$W_{ST} = \dot{m}_w (h_{IST} - h_{OST}) \quad (12)$$

The condenser heat balance is described as:

$$\dot{m}_{fw} C_{pfw} (T_{OUT} - T_{IN}) = \dot{m}_{STEAM} (h_{IC} - h_{OC}) \quad (13)$$

where \dot{m}_{fw} is the power plant feedwater flow rate, C_{pfw} is the power plant feedwater specific heat, T_{IN} and T_{OUT} are the inlet and outlet temperatures of the cold fluid in the condenser, where h_{IC} and h_{OC} are the inlet and outlet temperatures of the hot fluid in the condenser.

3.3.2. Cogeneration plant

The performance of the cogeneration plant is directly related to the operational behaviour of each of its components. The plant design is carried out using mathematical models to determine the thermodynamic properties of each one of its critical components. When implementing those models, the operational parameters, sizes, and configurations of the main system components such as heat exchangers, pipelines, insulation, and pumps are determined. The governing equations for the cogeneration plant critical components are described below:

▪ Heat Exchanger (HEX)

Considering a parallel flow heat exchanger in which the cold and hot fluids enter in the same direction and position, the temperature of hot fluid is higher in relation to the cold fluid, the mass flow rate for both fluids is greater than zero, where specific heat capacities and mass flow rates are assumed to be constant over the entire length of the HEX. According to [23], the total thermal power exchanged between the hot and cold fluid is calculated according to the following expression:

$$W_{HE} = U \cdot A \cdot \left(\frac{(T_{H,O} - T_{C,O}) - (T_{H,I} - T_{C,I})}{\ln \left(\frac{T_{H,O} - T_{C,O}}{T_{H,I} - T_{C,I}} \right)} \right) \quad (14)$$

where $T_{C,I}$ and $T_{H,I}$ are the cold and hot fluid inlet temperatures, respectively, $T_{C,O}$ and $T_{H,O}$ are the cold and hot fluid outlet temperatures, U is the overall heat transfer coefficient for the heat exchanger and A is the total surface area used for heat transfer. In the same way, using the correlations suggested in [23] the mean temperature difference is evaluated as follows:

$$\Delta T_m = \frac{(T_{H,I} - T_{C,I}) - (T_{H,O} - T_{C,O})}{\ln \left(\frac{(T_{H,I} - T_{C,I})}{(T_{H,O} - T_{C,O})} \right)} \quad (15)$$

▪ Pipelines

In the design process for high pressure and high-temperature pipes, the effects of changes in temperature changes and heat transfer, as well as pressure losses must be considered. To quantify frictional pressure losses in a pipeline, the Darcy-Weisbach relation is used according to [26], correlating the characteristics length of the pipe L_{PIPE} , diameter D_{PIPE} , the velocity of the flow v_{FLOW} , acceleration due to the gravity g and Darcy's friction factor f : its value is related to the Reynolds number Re and the type of flow inside the pipeline and expressed as follows:

$$h_f = f \cdot \frac{L_{PIPE}}{D_{PIPE}} \cdot \frac{v^2}{2g} \quad (16)$$

Pipeline temperature losses are determined using heat transfer theories of conduction (Fourier's equation), convection (Newton's equation) and radiation (Stefan-Boltzman's equation), through the pipeline heat flow per unit length calculation as suggested in [27]:

$$\frac{q}{L} = \frac{T_i - T_o}{\frac{1}{2\pi r_i h_i} + \sum_{i=1}^n \frac{\ln \left(\frac{r_{i+1}}{r_i} \right)}{2\pi k_i} + \frac{1}{2\pi r_o h_o}} \quad (17)$$

where T_i and T_o are the temperatures on the pipe internal and external surfaces, respectively, r_i and r_o are the internal and external radius of the pipe, r_{i+1} is the radius of the pipe along with the variable thickness, k_i is the conductive heat transfer coefficient for each pipe material along with the thickness and h_i and h_o are the convective heat transfer coefficient inside and outside of the pipe, respectively.

To minimize the heat losses, pipeline insulation must be introduced, and its thickness must be determined. This process is carried out by matching the total pipeline heat flow to the corresponding heat flow between the surface to be insulated $T_{s,o}$ and the boundary conditions of the environment [28]:

$$\frac{T_{s,o} - T_o}{\frac{1}{2\pi r_o h_o}} = \frac{T_i - T_o}{\frac{1}{2\pi r_i h_i} + \sum_{i=1}^n \frac{\ln\left(\frac{r_{i+1}}{r_i}\right)}{2\pi k_i} + \frac{\ln\left(\frac{r_{ins}}{r_o}\right)}{2\pi k_{ins}} + \frac{1}{2\pi r_o h_o}} \quad (18)$$

where r_{ins} is the external radius of the insulation, and k_i represent the conductive heat transfer coefficient for the insulation material.

The cogeneration system design viability can also be quantified in terms of greenhouse gas emissions that will not be emitted by replacing the system that supplies thermal energy to the industrial process.

The greenhouse gas emissions amount is mainly related to the energy produced by the fuel when it is burned. In agreement with the United States, Environmental Protection Agency (EPA) [29], the greenhouse gas emission's main component evaluated in processes related to electricity generation are carbon dioxide emissions (CO_2). According to the International Energy Agency (IEA) [30], the CO_2 amount produced in a combustion process is a function of the carbon content of the fuel. The heat content or energy produced is mainly determined by the carbon (C) and hydrogen (H) content of the fuel. Natural gas has a higher energy content relative to other fuels; thus, it has a relatively lower CO_2 -to-energy content.

The emission factor is used to quantify the CO_2 emissions produced by the fuel in the combustion process to generate electricity. Mareddy [31] defines an emissions factor as a representative value that relates the quantity of a pollutant released to the atmosphere with an activity associated with the release of that pollutant. Usually, this factor is expressed as the weight of pollutant divided by a unit weight, volume, distance, or duration of the activity emitting the pollutant (e.g., kilograms of particulate emitted per Megagram of coal burned).

The fuel emission factor can be determined as a function of the calorific value of the fuel, the carbon amount in the fuel and the complete oxidation stoichiometric factor of carbon to carbon dioxide [31, 32]. The general equation for emissions estimation is expressed as follows:

$$EF = \frac{(\sum_i^n n_i \cdot X_{mol\ i})}{(\sum_i^n NCV_{mol\ i} \cdot X_{mol\ i}) \cdot A} \quad (19)$$

where EF is the emission factor, n is the number of carbon atoms in the fuel, $X_{mol\ i}$ is the molar fraction of component i in the fuel, $NCV_{mol\ i}$ is the net calorific value as an ideal gas of component i in kJ/mol , which can be obtained from Table 1 of the ASTM-D-3588 [32] standard and A is the complete oxidation stoichiometric factor.

From the fuel emission factor and the amount of fuel required by the electricity generation process, CO_2 emissions are determined by the following equation:

$$E_{\text{CO}_2} = EC \cdot EF \cdot (1 - ER) \quad (20)$$

where E_{CO_2} are the fuel carbon dioxide emissions in the period evaluated, EC is the energy consumption in the electricity generation process, EF is the emission factor, and ER overall emission reduction efficiency in percent.

4. Case Study

The proposed methodology is tested through the use of a case study that focuses on the optimal design of a CHP system based on an existing combined cycle power plant that supplies thermal energy to an electrowinning process of a copper mining complex in Chile.. From the CHP proposed design system, a study to quantify the operational flexibility impact in the whole system is conducted. Simulation models have been developed using well-known and benchmarked modelling and simulation software OpenModelica [33]. The simulation models were developed using the ThermoSysPro library which contains component models expressed in the Modelica language for the modeling and simulation of power plants and energy systems [23]. The case study is carried out in accordance with the design process described in the proposed methodology given previously in Figure 3.

4.1. CCPP Simulation model

The power plant simulation model is based on a model presented in [34], which consists of a gas turbine, a heat recovery steam generator (HRSG) with three evaporating loops (seven superheaters, three evaporators and six economizers), a three-stage steam turbine (low, intermediate, and high pressure), an electric generator, a condenser, several pumps, valves and pipes, and proportional-integral (PI) control systems to limit the drum levels. In Figure 5, the combined cycle power plant model in the OpenModelica OMEdit graphic environment is shown.

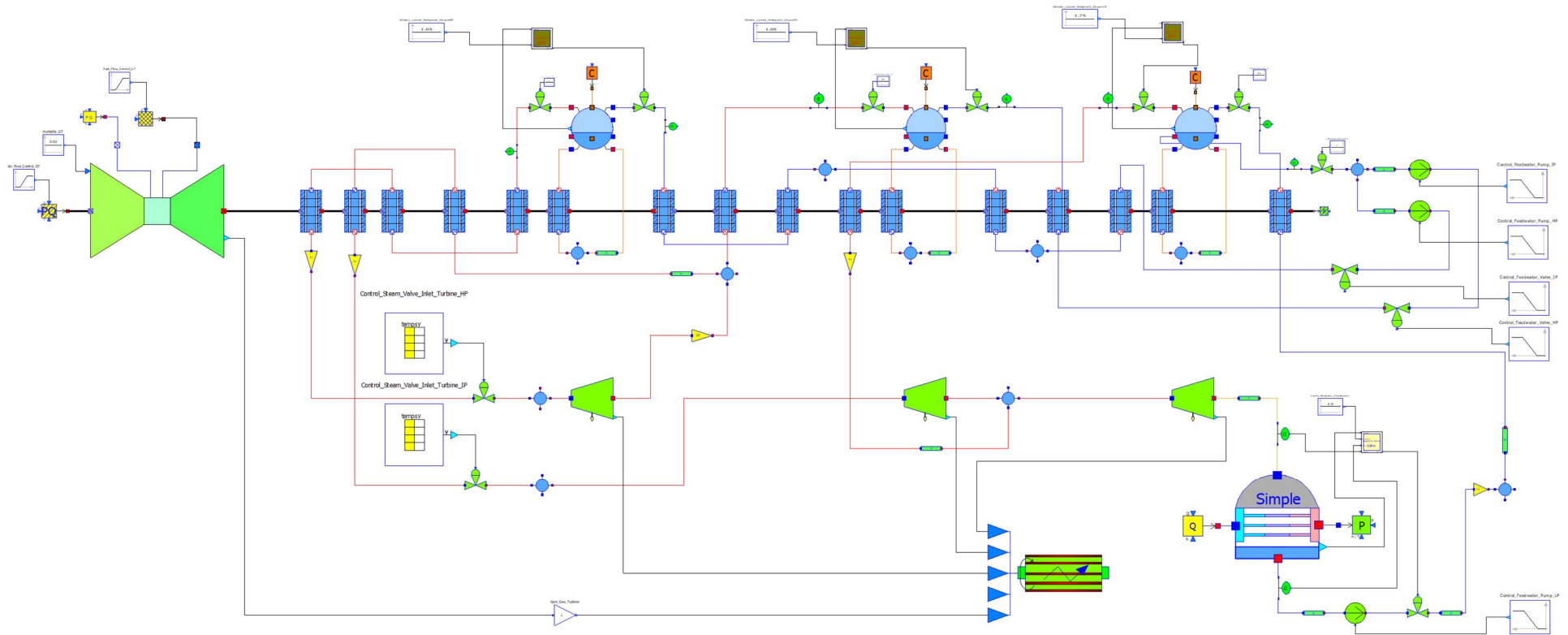


Figure 5. Combined cycle power plant simulator in OpenModelica graphical environment.

4.2. CCGP Simulation model validation

The combined cycle power plant model was validated against the load change scenario published by [34], in which the power plant goes from an initial state (100% load) to a final state (50% of load) in about 800 seconds. Similarly, mechanical power profiles in the gas and steam turbines were also compared, in terms of the level measured at the three drums. Evaluation of the derived results is focused on monitoring the electrical power generation capacity in the two cycles of the power plant, as well as the level of the HRSG drums since these can lead to steam production inefficiencies, steam quality issues, and power plant safety risks. Figures 6 and 7 show the results reported by [34] and those obtained with the model developed for the needs of this research work.

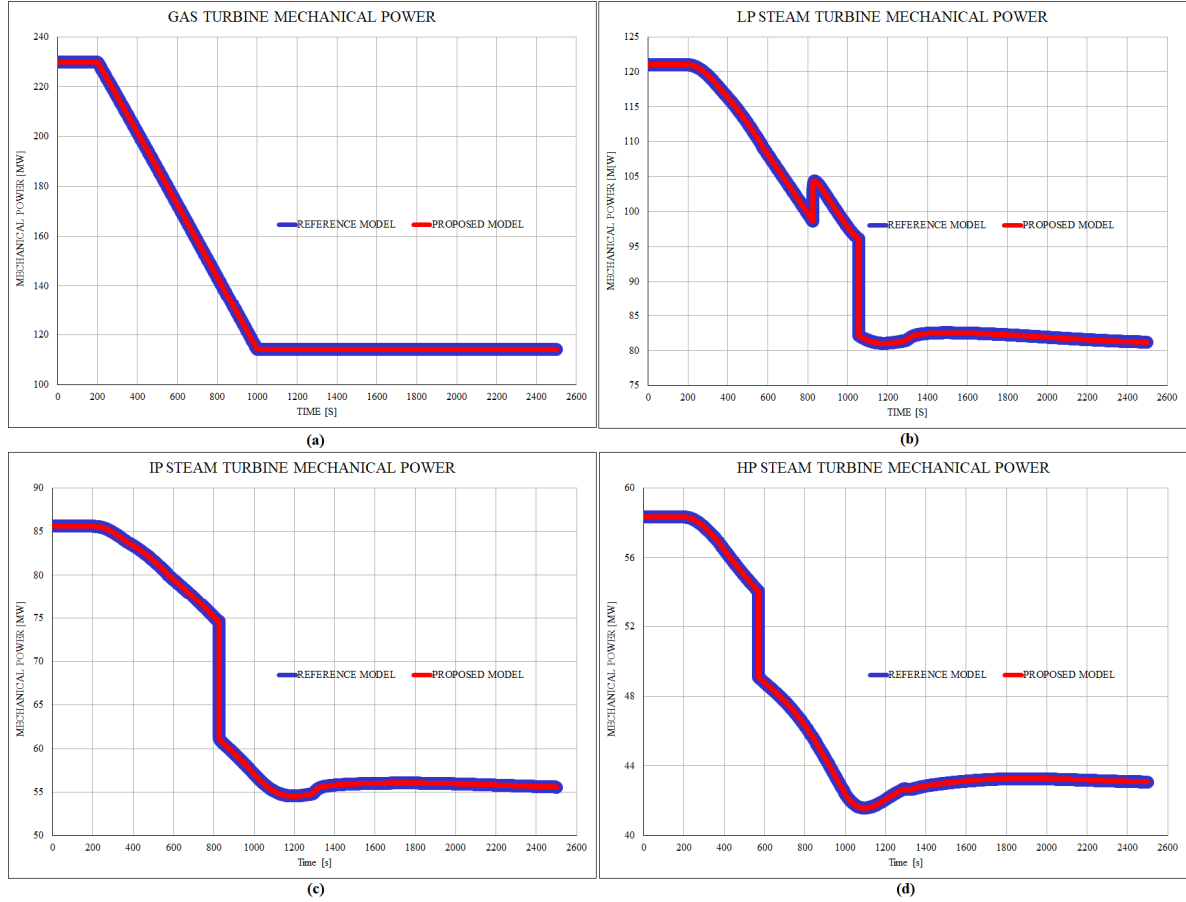


Figure 6. Results comparison between the profile reported by Hefni et al. [34] (blue line) and the proposed simulation model (red line). a) gas turbine mechanical power; b) low-pressure steam turbine mechanical power; c) intermediate pressure steam turbine mechanical power; d) high-pressure steam turbine mechanical power.

As expected, the numerically obtained results derived simulation profiles are consistent and very close to those reported in [34], with negligible numerical differences and in excellent trend agreement. Accordingly, it can be concluded that the combined cycle power plant dynamic simulation model implemented in OpenModelica has been validated since the results reported by [34] were originally endorsed with operational data of an existing power plant in Vietnam [23]. As expected, the gas turbine mechanical power decreases linearly, as time progresses, almost to an asymptotic value corresponding to the minimum load for the change load case study, as shown in Figure 6a. Regarding the steam turbines, the expected non-linear behaviour of mechanical energy can be depicted in Figures 6b, 6c, and 6d. This mechanical power response is mainly attributed to the dynamic energy

exchange between exhaust gases and water steam in the sixteen-heat exchanger of the steam generator. Figures 7b, 7c and 7d show the dynamic effect of swelling in the high, intermediate and low-pressure drums, in which the water level inside the drum increases when the corresponding drum pressure decreases.

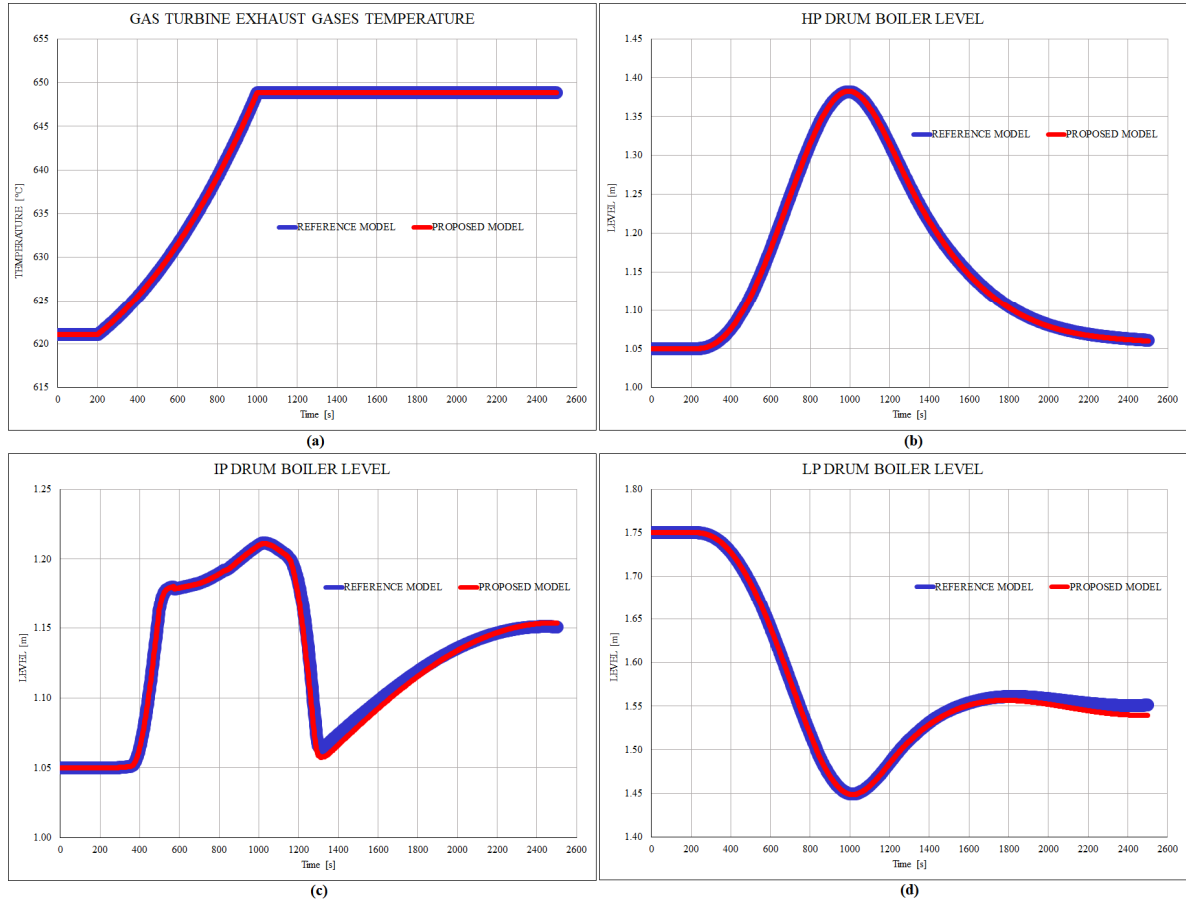


Figure 7. Results comparison between the profile reported by Hefni et al. [34] (blue line) and the proposed simulation model (red line). a) gas turbine exhaust gas temperature; b) high-pressure drum boiler level; c) intermediate-pressure drum boiler level; d) low-pressure drum boiler level.

4.3. Industrial process model

Large amounts of fuel are required by the mining industry in order to provide heat for its processes. Most mining companies use diesel power to supply those processes that require heat. This is mainly attributed to its operational readiness, reliability, and durability and ease of use bearing a low investment risk. In copper mining, one of the most important processes is electrowinning, where an external heat source is used to hold the electrolyte temperature within an operating threshold in order to achieve high purity and quality cathodes. The electrowinning process cannot tolerate any operational interruption, thus must be operating in a continuous mode [35].

The electrowinning plant that was analysed in this work corresponds to a copper mining installation currently operating in Chile [36]. This plant produces approximately 150,000 tons of fine copper per year. Currently, a diesel boiler coupled to a counterflow plate heat exchanger is used to produce the heat required by the electrowinning process. Figure 8 shows the operating parameters and the configuration of the actual heating circuit for the electrowinning plant under study. Additionally, the characteristics of the copper electrowinning process studied herein are summarized in Table 1.

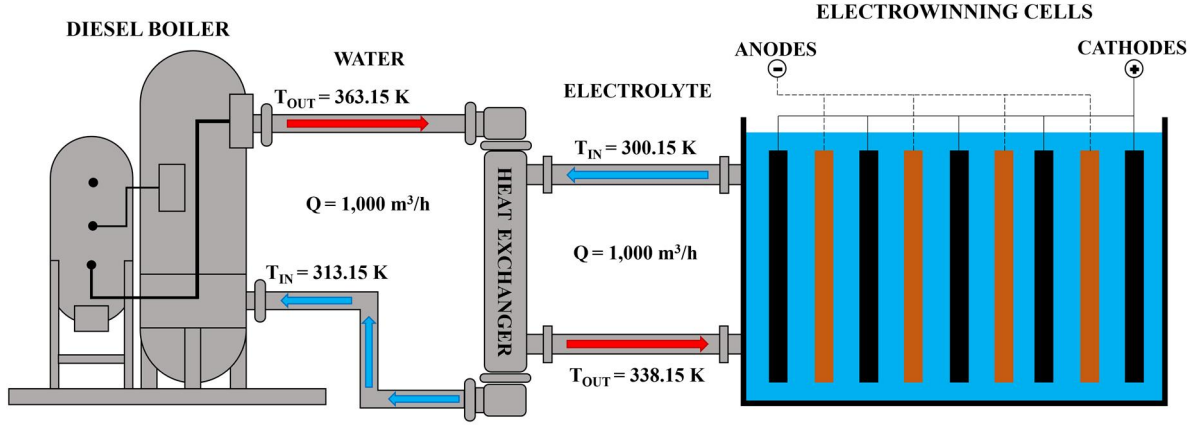


Figure 8. Heating circuit features of the electrowinning plant based on [36].

Table 1. Electrowinning process characterization [36].

| Parameter | Value |
|------------------------------------|--------|
| Inlet temperature (K) | 303.15 |
| Desired temperature (K) | 338.15 |
| Flow rate in the inlet (m³/h) | 1,000 |
| Electrolyte density (kg/m³) | 1,225 |
| Electrolyte specific heat (J/kg K) | 3,480 |

4.4. Modified flow chart

A modified flow chart is developed and analysed herein, which consists of a splitter and two flow regulating valves that are synchronously operated to adjust the exhaust gas flow needed by each component of the CHP system. Between the gas turbine outlet and steam generator inlet, a splitter is installed distributing the exhaust gases in three different directions: (a) to the heat recovery steam generator, (b) to the cogeneration plant; and (c) to the flue gas stack. In addition, two control valves are added to regulate the hot gas flows. A control system for every valve regulates the optimal amount of flows needed by the steam generator and cogeneration plant, prioritizing thermal energy generation for the electrowinning process.

The viability of the exhaust gases splitter system proposed in this research work is evaluated through the energy balance of the water/steam cycle and exhaust gases in the cogeneration plant according to the following expression:

$$\dot{m}_{EG,CS} C_{peg} (T_{IEGCS} - T_{OEGCS}) = \dot{m}_{E,CS} C_{pe} (T_{IECS} - T_{OECS}) \quad (21)$$

where C_{peg} , C_{pe} , $\dot{m}_{EG,CS}$ and $\dot{m}_{E,CS}$ are the exhaust gases specific heat, electrolyte specific heat, exhaust gases mass flow rate and electrolyte mass flow in the cogeneration system, respectively. Furthermore, T_{IEGCS} and T_{OEGCS} represent the temperatures of exhaust gases entering and exiting the cogeneration system. Likewise, T_{IECS} and T_{OECS} are temperatures of electrolyte entering and exiting the cogeneration system, respectively.

From this energy balance, it is possible to evaluate if the cogeneration system capacity to produce thermal energy reaches the levels necessary to bring the electrolyte solution to the electrowinning process operating conditions as described in Figure 8 and Table 1. Figure 9 shows the implementation of the splitter system for the gas turbine exhaust gases in the proposed combined cycle power plant simulation model.

4.5. Cogeneration System Configuration and Implementation

Based on the typical electrowinning process operational parameters, the location and design of the splitter system were determined. Therefore, the next step is to determine the configuration and design of the cogeneration plant. In accordance with the mining map of the National Mining Society of Chile [37] and the installed power generation data of the National Electric Coordinator of Chile [38], the reference power plant can be geographically positioned close to the electrowinning process plant that was investigated in this research work.

In these terms and considering the security and operational efficiency of the system, the proposed cogeneration system operates as follows:

- Hot water close to saturation conditions is generated in the power plant.
- The design of the hot water circuit involves a water tank, several pipes and pumps, and a counterflow heat exchanger. The gas turbine exhaust gases derive from the hot fluid, while the water of the tank is the cold fluid.
- A pumping system and insulated pipes are used to transport hot water to the electrowinning plant.
- In the electrowinning plant, a counterflow heat exchanger is installed in order to heat the electrolyte solution to the required operating conditions. Hot water from the power plant is the hot fluid, while the electrowinning cells electrolyte is the cold fluid.
- Coupling is performed of both systems close to the circuit of the water/steam cycle of the cogeneration plant.

The proposed design and implementation of the cogeneration system coupled with the combined cycle plant simulation model is shown in Figure 9.

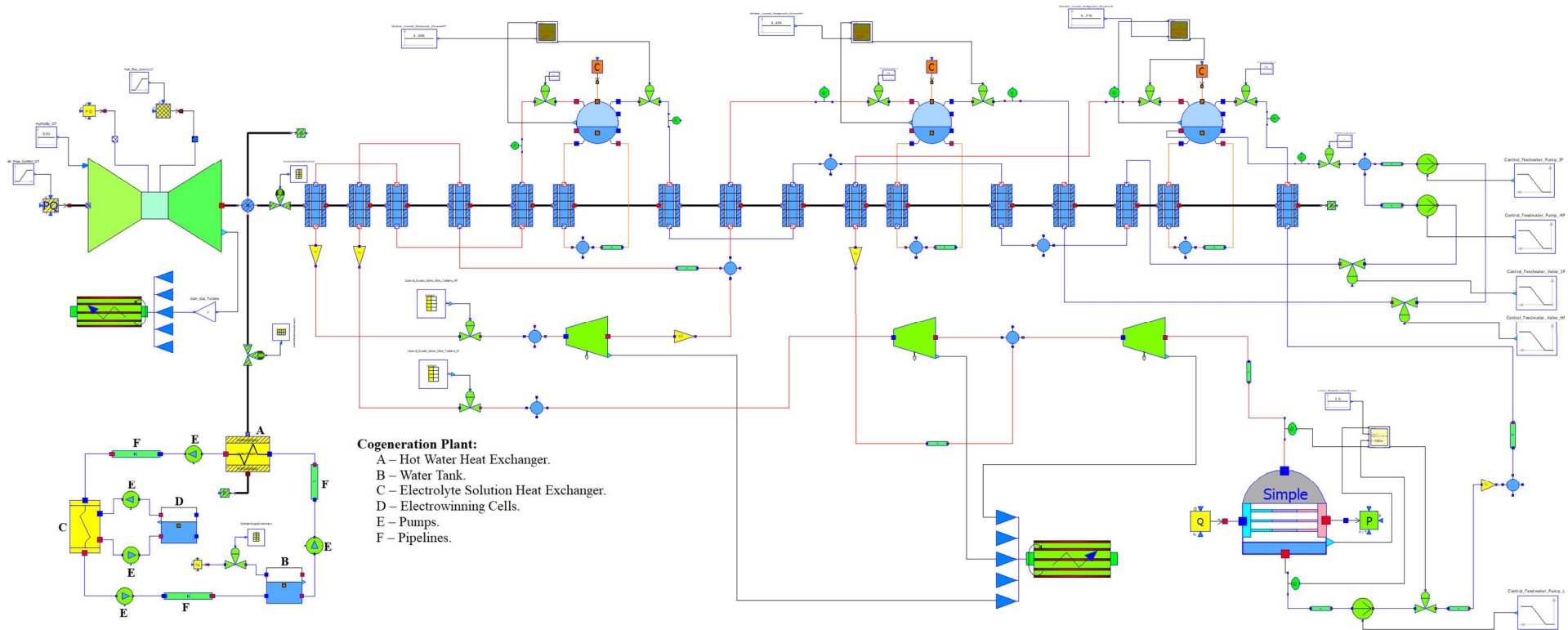


Figure 9. Proposed combined heat and power simulation model – OpenModelica.

5. Numerical experiments, results, and discussion

To solve the design problem, the combined cycle power plant simulation model was adjusted in terms of the gas turbine mechanical power according to operational data of the San Isidro II power plant operating in Chile. For this study, the steam power cycle configuration between the reference model and a real power plant are considered to be equivalent since both have a heat recovery steam generator with three evaporating loops and sixteen heat exchangers, a three-stage steam turbine (low, intermediate, and high pressure), a condenser, centrifugal pumps, control valves, and pipes. The steam turbine mechanical power is regulated according to new gas turbine operating parameters.

A splitter location feasibility study was evaluated according to the exhaust gases operating parameters shown in Table 2 that were obtained from a power plant load change simulation.

Table 2. Exhaust gases operating parameters for a load change simulation.

| GT Load [%] | Exhaust Gases | | | | |
|----------------|---------------|-----------------------|--------------------------|--------------------------|--------------------------|
| | Q [kg/s] | T _{HRSG} [K] | T _{HRSG-HP} [K] | T _{HRSG-IP} [K] | T _{HRSG-LP} [K] |
| 100 | 606.207 | 894.22 | 606.41 | 509.56 | 395.45 |
| 90 | 546.002 | 900.40 | 600.99 | 505.35 | 404.54 |
| 80 | 484.661 | 909.66 | 593.03 | 497.86 | 392.34 |
| 70 | 424.456 | 922.02 | 585.32 | 491.46 | 389.70 |
| 60 | 395.442 | 948.74 | 585.14 | 491.04 | 387.95 |
| 50 | 302.428 | 974.24 | 579.71 | 488.10 | 385.53 |

Four alternative scenarios were selected: HRSG inlet, HRSG-HP output, HRSG-IP output, and HRSG-LP output at different load levels ranging from 100% to 50% of the gas turbine. The results evaluation is presented in Figure 10, where all cases are compared quantifying the energy consumption and exhaust gas temperature decrease. An energy balance between the exhaust gases and electrolytes is also performed, where the electrolyte receives thermal energy from the hot exhaust gases.

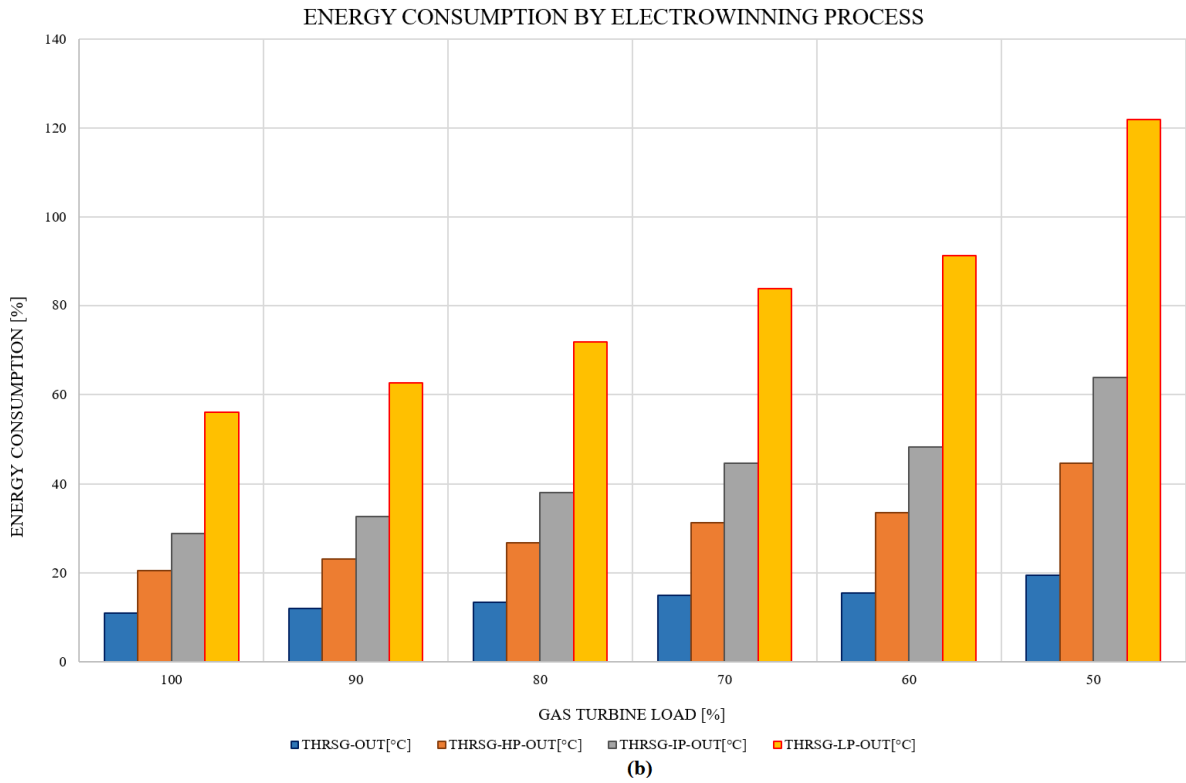
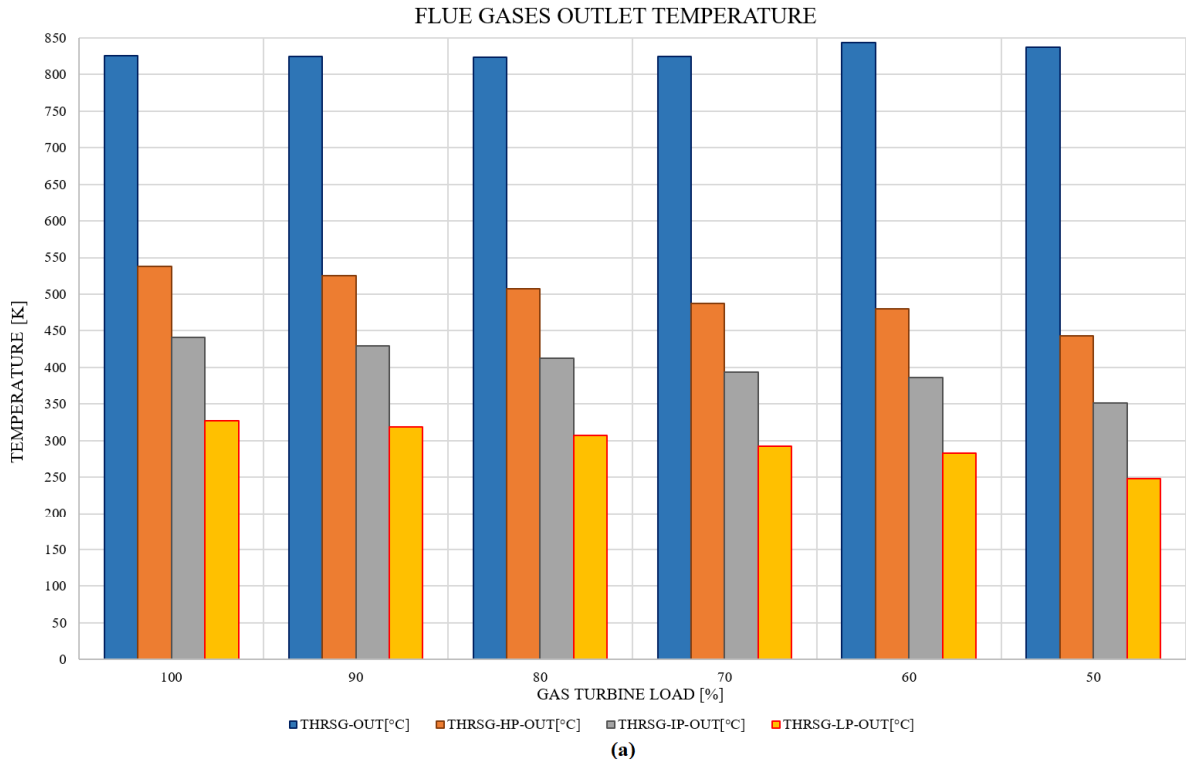


Figure 10. Comparison between the flue gas temperatures (a) and energy consumption for different splitter location (b).

According to the results shown in Figures 10a and 10b, the alternative HRSG-LP output indicates that it is not a feasible solution for cases where the heat load is less than 60% in the steam cycle. This response is attributed to the thermal energy at this location, which is less than the thermal energy required by the electrowinning process. Concerning the exhaust gases exiting temperature, for load cases below 80% do not provide useful thermal energy, since the exhaust gases are driven at temperatures equal to or less than the

initial temperature of the water that needs to be heated. For the HRSG-HP and HRSG-IP output alternatives, it is required to extract about 30% and 45% of the available energy, respectively, in order to generate electricity in the intermediate and low-pressure turbines given that these turbines are the ones that produce the most energy within the steam cycle. This energy loss would represent an average decrease in the generation capacity of the steam cycle close to 30%.

Concerning the exhaust gas temperatures for load levels of 70% and lower, it was found that the system could not supply steam at temperatures required by intermediate and low-pressure turbines. For the splitter system located at the HRSG inlet, there is a 15% average decrease in the exhaust gas temperature for all load levels and the operating temperatures are in the order of 900 Kelvins. With respect to the thermal energy required for the electrowinning process, this is matched by the exhaust gases mass flow that is found to be around 15% at the inlet of the HRSG scenario. These changes are the ones affecting the least the overall performance of the system, thus achieve optimum operational management of the CHP system. Therefore, this location would be the most feasible and viable location for assuring an easy installation process. Based on this finding, it is recommended to implement the exhaust gases splitter system in the steam generator inlet. A summary of the main geometric and operational features of the proposed design of the cogeneration plant are presented in Table 3.

Table 3. Proposed cogeneration system operating parameters for full load gas turbine.

| Component | Value |
|---|---------|
| Heat exchanger (hot water) | |
| Exhaust gases mass flow (kg/s) | 90 |
| Exhaust gases inlet temperature (K) | 894.15 |
| Water mass flow (kg/s) | 278 |
| Water inlet temperature (K) | 313.15 |
| Global heat transfer coefficient ($\text{W/m}^2 \text{ K}$) | 3,500 |
| Global heat exchange surface (m^2) | 150 |
| Pipeline (hot water) | |
| Pipe internal diameter (m) | 0.508 |
| Wall thickness (m) | 0.03175 |
| Thermal insulation thickness | 0.120 |
| Heat exchanger (electrolyte) | |
| Water mass flow (kg/s) | 278 |
| Water inlet temperature (K) | 363.15 |
| Electrolyte mass flow (kg/s) | 278 |
| Electrolyte inlet temperature (K) | 300.15 |
| Heat transfer coefficient for the hot side ($\text{W/m}^2 \text{ K}$) | 7,250 |
| Heat transfer coefficient for the side ($\text{W/m}^2 \text{ K}$) | 1,800 |

The cogeneration plant design was implemented into the combined cycle power plant simulation model, while the CHP system performance was evaluated for a power plant decrease load change case study. The respective results comparisons were performed against a reference model [34]. In Figures 11a and 11b the exhaust gases flow deviated from the cogeneration plant are shown and the relative impact on the mechanical power profiles of a steam turbine is quantified accordingly.

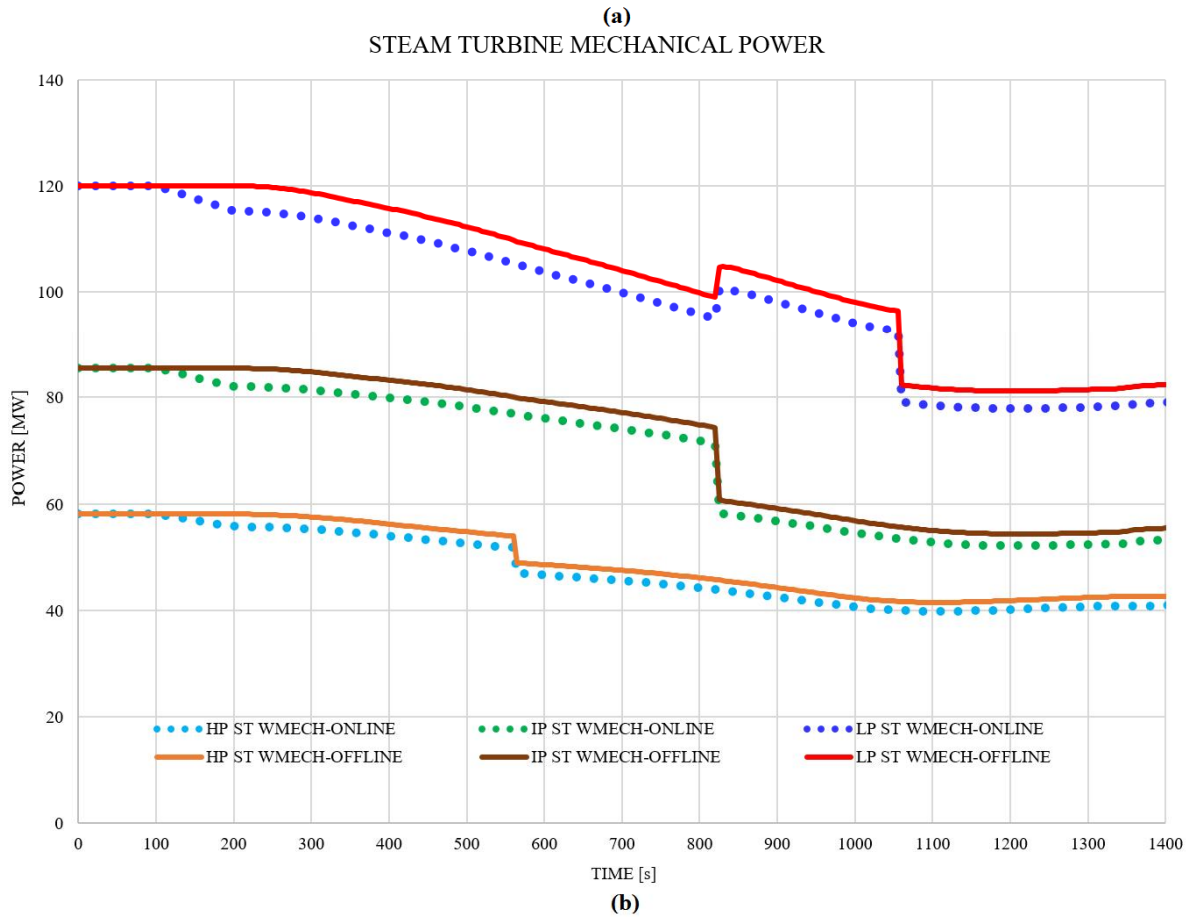
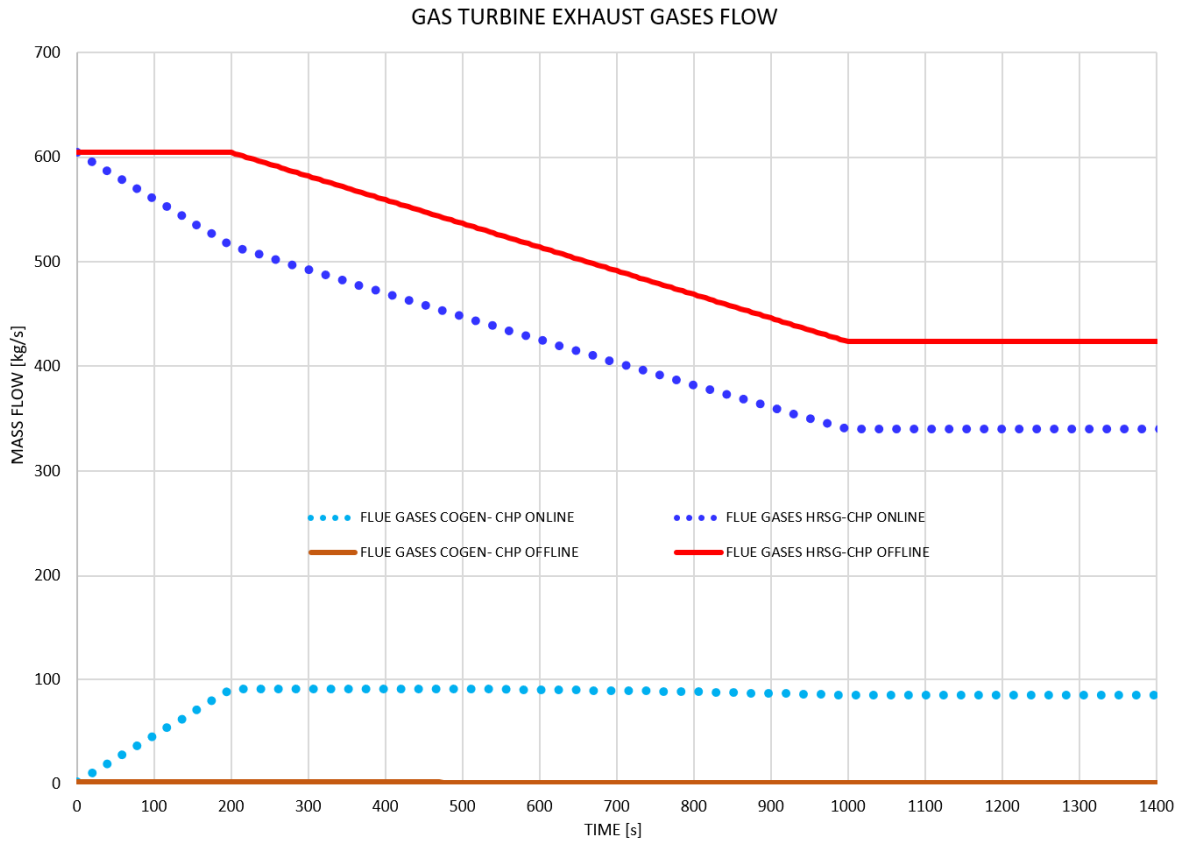
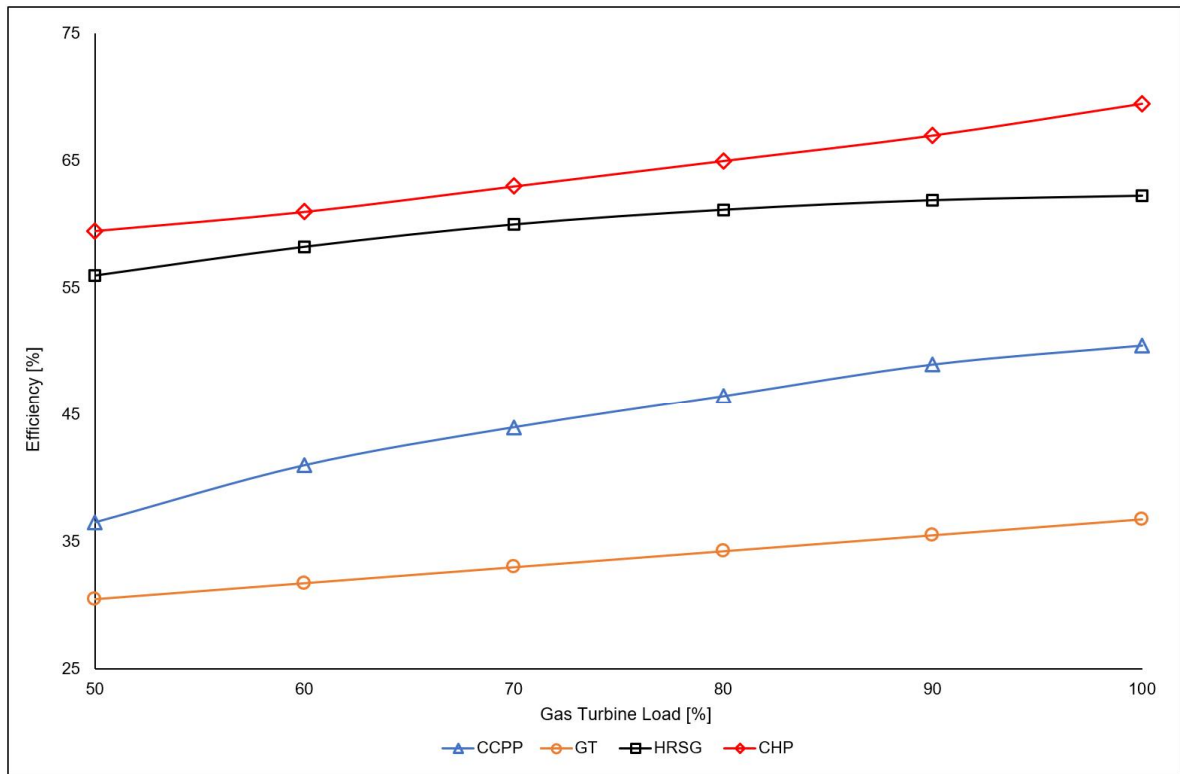


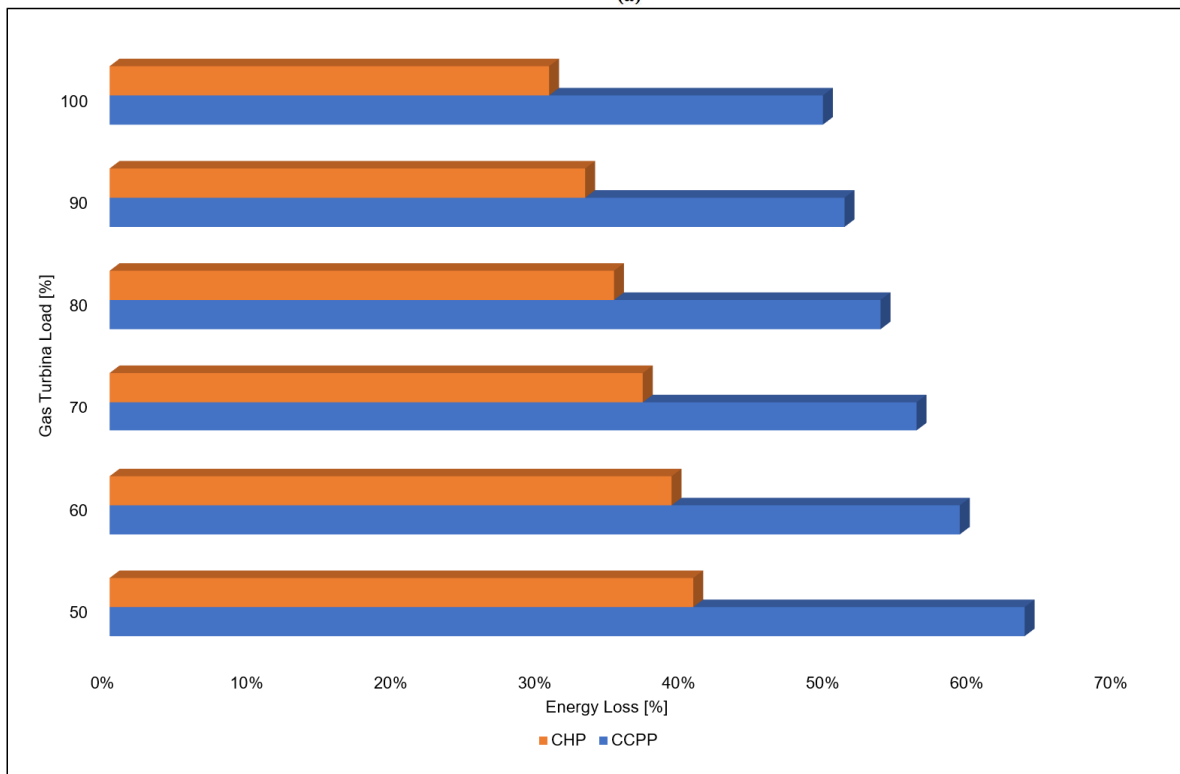
Figure 11. Mechanical power profiles (a) and exhaust gases flow profiles (b) for the Combined Cycle Power Plant and CHP System.

In order to supply thermal energy that is required by the electrowinning process, around 15% of exhaust gases must be diverted from the cogeneration plant for the gas turbine full-load case. While equivalent mass flows of 20%, 19%, 18%, 17% and 16% must be directed to the cogeneration plant for partial loads of 50%, 60%, 70%, 80% and 90%, respectively. It was noted that in regard to the steam turbine mechanical power, there is an average decrease of 5% and a maximum decrease of 17% per turbine stage (HP, IP, and LP) for the cases considered. In contrast, it is observed that the proposed CHP system's average thermodynamic efficiency is 19% greater than the CCPP's average efficiency, increasing from 50% to 69%. In addition, the proposed design of the CHP system causes an average decrease of 32,500 tons of carbon dioxide emissions per year by replacing the diesel boiler that currently provides thermal energy to the electrowinning process in the mining complex.

By results shown in Figure 12a, for cases where the Gas Turbine load is less than 65%, the thermodynamic efficiency of the power plant is affected by the amount of flue gas required by the electrowinning process. For these load cases, the efficiency change ratio is 2.5 higher compared to a load change of 10% in loads greater than 70% in the Gas Turbine. Regarding the energy loss, the Combined Cycle Power Plant presents 24% more loss concerning the CHP system for a 50% load in the Gas Turbine, while for full load the energy loss ratio between CCPP and CHP system is 19% as shown in Figure 12b. In this sense, the feasible operation region of the CHP system will be limited to loads greater than 65% in the Gas Turbine, in order to have a suitable efficiency of the Rankine cycle power block and stability in the electrical power generation.



(a)



(b)

Figure 12. Thermodynamic efficiencies (a) and energy loss (b) for different components of the CHP System.

6. Conclusions

This research has focused on the development of a base simulation model useful for developing more complex systems involving the design and optimization of mining districts, where several thermal power plants and mining complexes requiring thermal energy for heating processes are integrated. A Chilean mining industry case has been studied since mining represents around 10% of the GDP of the country and it represents by far the highest greenhouse gas emission. A designed based dynamic simulation models of a combined heat and power system for the efficient supply of thermal energy for a copper mine electrowinning processes have been developed and operational simulations have been out in detail.

The proposed methodology in fact aims at improving the operational capabilities of an existing thermal power plant by designing a cogeneration plant coupled with an industrial process of high thermal energy demand. To achieve accurate results, the CCPP simulation model was validated against results available in the literature and compared with real operational data. In order to increase the efficiency of the combined heat and power generation, several alternatives for the cogeneration plant location and splitter system design were evaluated. An exhaust gas splitter system placed in the steam generator inlet was found to be the most feasible solution with a splitter junction and two flow regulating valves: those components are synchronously operated to adjust the exhaust gas flow needed by each component of the CHP system. To supply thermal energy to the electrowinning process, around 15% of exhaust gases must be diverted to the cogeneration plant for the gas turbine full load case, and the exhaust gas flow should be increased by an average of 1% percent when the gas turbine load decreases in the order of 10%. Regarding to the steam turbine mechanical power, there is an average decrease of 5% and a maximum decrease of 17% per turbine stage (HP, IP, and LP) for the cases considered. A cogeneration plant with two heating modules was proposed. In the first module, hot water close to saturation conditions is produced transferring heat from exhaust gases to the feed water using a counterflow heat exchanger. Hot water produced in the power plant is transported by an insulated pipeline to the second heating module, in which hot water transfers thermal energy to the electrolyte solution through another counterflow heat exchanger.

The numerical models were developed using the well-known software OpenModelica. The corresponding CHP system performance was evaluated for a power plant decrease load change case study and relevant results comparisons were performed against a reference model. The dynamic simulation model for the proposed CHP system was found to provide promising results in terms of power generation, thermodynamic efficiency, and a relative CO₂ emission reduction. Since the CHP system's average thermodynamic efficiency is 19% greater than the CCPP's average efficiency, increasing from 50% to 69%, and the proposed design of the CHP system causes an average decrease of 32,500 tons of carbon dioxide emissions per year. Finally, based on the thermodynamic efficiencies and the loss energy analysis of the CCPP and CHP system, the feasible operation region of the CHP system will be limited to loads greater than 65% in the Gas Turbine: this range preserves and acceptable efficiency of the Rankine cycle power block and helps stability in the electrical power generation.

According to the research findings, it is safe to conclude that from the power plant's operational strategy point of view, a design that is based on dynamic simulation models can improve and diversify the CHP system's applications. Conclusively, this can significantly improve the CHP system's operational flexibility to face the new challenges of liberalized markets in the short and medium-term.

As future work, an economic analysis of the cogeneration plant proposed design will be performed and the integration of a thermal energy storage system to optimize the operability of the CHP system will be further investigated.

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Data availability statement: The data that support the findings of this study are available in [Sociedad Nacional de Minería] at [<https://www.sonami.cl/mapaminero/>] and [Coordinador Eléctrico Nacional | Servimos a Chile con Energía] at [<https://www.coordinador.cl/>].

Conflicts of Interest: The authors declare no conflict of interest.

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