Energy and exergy assessment of scenarios utilizing a comprehensive combined heat and power and district heating systems

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A new comparative study between different district heating scenarios fed by a combined power plant from an energy and exergy point of view has been presented in this research. The proposed scenarios provide the flexibility of the power plant cycle in different environmental and geographical conditions to produce balanced electricity and heat. The proposed scenarios have been divided into design and off-design conditions when the maximum electricity or heat is the purpose of system operation. Results show that by relying on off-design scenarios, affordable outcomes are achievable. The heat efficiency in an off-design scenario has increased more by than 30% compared with the base case by controlling the exact demands on the equipment in the maximum power generation scenario. The total energy efficiency varies between 56.6 to 87.6 in different scenarios. It has been similarly observed for the energy and exergy efficiencies by growing 2% in the maximum heat production scenario. In addition, preparing a balance between the highest power of heat production has been fulfilled by proposing a medium model that can be supplied in an acceptable range. Eventually, the highest value of exergy destruction has been observed in the combustion chamber of the gas cycle (about 68%) due to thermochemical irreversibility.

keywords: *Energy modelling, power generation, energy efficiency, exergy destruction, heat recovery.*

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CHP	Combined Heat and Power
ex	Feature Specific exergy [$kJkg^{-1}$]
E_x	Exergy flow [kW]
GC	Gas Cycle
GT	Gas Turbine
HRSG	Heat Recovery Steam Generator
m	Mass [kg]
P	Power [kW]
ST	Steam Turbine
T	Temperature [°C] centers.
ΔE_x	Exergy change in a flow [kW]
η	Efficiency
ϕ_m	Mass flow [kgs^{-1}]

subscript

E_x	Exergy
f	Functional
flue gas	Flue gas
fuel	Fuel
in	Inlet
out	Outlet
p	Primary
product	Product
oxidant	Oxidant
s	Secondary
source	Source

superscriptsshaft

ch Chemical
tm Thermo-mechanical

1. INTRODUCTION

The growth of apartment housing in cities and the rise of smaller buildings have led many people and builders to use heating packages. At least, the independence of the residential unit is one of the advantages of this heating system. Still, the irreparable risks of using the package far outweigh the concerns about the increase in gas prices or the monthly charge of the building [1].
In recent years, the use of appliances such as home heating systems such as heaters, water heaters, and wall-mounted packages has added to the problems of energy carriers. In many cases, it has turned healthy and safe homes into sick unsafe dwellings [1, 2].
In a district heating system, the thermal energy required for heating the building, heating water consumption, and heating processes needed by commercial and industrial consumers is provided through the central powerhouse or recovering heat generated in power plants. Steam pipes or heating water pipes will transfer this energy. The application of heat recovery from power plants in the surrounding areas to increase welfare and reduce urbanization costs seems necessary. Reducing greenhouse gas emissions is another achievement of district heating [1, 2].
District heating systems are part of the decarbonization of hot sections because they allow the integration of essential and clean energy sources, which can be at the level of individual buildings in hot areas. However, almost many cities are currently implementing low-carbon regional heating solutions, 90% of which are still based on fossil fuels [1, 2].
In the zero-emission scenario by 2050, the total share of cash and electricity resources in the global, regional heat supply will increase from 8% today to about 35% in the current decade, helping to reduce carbon dioxide emissions by one-third [1].
Global district heating production was 16 EJ of heat in 2020, which jumped 30% from the 2000 level with an annual compound growth rate of 1.3% (or 2.4% if normalized for weather conditions). The dramatic 2.3% increase from 2019 to 2020 was driven mainly by China and, to some extent, by Korea (7% growth each) [1].
The share of countries and regions of the world in district heating by type of self-consumption in the energy production units, industrial, construction, or agriculture is indicated in Fig. 1 [1].
Different generations of district heating are shown in Table 1. In this Table, the development trend of such systems has been expressed over the past years [2].
Cogeneration plants, often known as combined heat and power (CHP) plants, are also critical identifiers of the region's energy systems due to the simultaneous generation of usable electricity and heat [3]. A combined heat and power plant designed as a topping cycle can be used for various purposes for district heating applications. For example, desalination, heating of buildings, or hot water production [4–8].
In a study, Kelly and Pollitt examined the benefits of using district heating in the UK. They showed that this would significantly reduce greenhouse gas emissions [9]. They obtained the connection between combined heat and power, and district heating systems could be the best option for governments to reduce costs and increase energy efficiency in cold regions.
Kılkış, in a paper-based district heating system built in Östra Sala backe in Uppsala Municipality in Sweden, showed that according to 5 scenarios considered, the amount of exergy correlation between supply and demand when the annual electrical load is 46 GW h^{-1}, and the annual heat load is 67 GW h^{-1}, is almost 84 %. In this research, more energy sources for use in district heating are also recommended [10].
About 50% of electricity demand is generated through CHP systems in Denmark. Thus, it is possible to store energy properly in different units and have high flexibility for other electricity or heat generation priorities in each seasonal procedure [11].
Ahmadi et al. [12] evaluated the optimization of CHP systems. The results indicated that the highest exergy destruction is related to the combustion chamber in a cycle designed for district heating. They also showed the destructive effect of increasing the inlet air temperature of the cycle on overall efficiency.
The utilization of waste heat in industries for district heating has been studied by Fang et al. [13]. They presented that by transferring the recovered heat of several factories with temperatures between 20 and 90°C to a more remote area, 390,000 GJ of thermal energy could be saved annually. It also prevents the annual emission of 35,000 tons of CO2 and saves 150,000 tons of water.
Fazlollahi et al. [14], in their research, proposed that by setting up a district heating unit, adverse environmental effects can be reduced by up to 65%. In the system they introduced, the total energy efficiency can be increased up to 75%.
The positive effects of operating a district heating system, in the long run, have been presented in a study by Siuta-Olcha et al. [15]. The results show that energy consumption for hot water production in buildings has decreased by more than 29% over ten years. This is equivalent to a reduction of 0.13 $GJ.m^3$ of annual thermal energy consumption. In addition, the amount of heat consumed for space heating in various buildings has decreased from 17.8 to 23%.
Seasonal optimization of a 4 MW district heating system in Finland was performed in a study by Kousa et al. [16]. The results of this study based on the ambient temperature prediction model showed that creating flexibility in the system by increasing the temperature by 15 degrees leads to saving 185 MWh of energy and reducing 40 tons of CO2 emissions.
The characteristics of thermal storage systems in a district heating system were evaluated in a study by Siddiqui et al. [17]. Their research demonstrated that the optimal thermal storage should have a capacity equal to 1% of the required annual heat.
Turunen et al. [18] studied the determination of the appropriate capacity for a thermal sink in a district heating system. Their results showed that by changing the heat load consumption of the CHP system, a significant amount of energy could be saved during peak energy consumption periods.
Corradi et al. [19] investigated the differences between district heating and individual heating in Italy. Their results showed that the current thermal power plant configuration with two boilers and a CHP unit reaches a yearly primary energy saving of 21.3% and 1099 tCO2 avoided.
One of the essential issues in designing and evaluating district heating systems based on supplying heat from the waste recovery of power plants is considering off-design conditions for electricity or heat generation. Researchers in previous studies have less considered this. Therefore, this study investigates a CHP system to provide the required heat for a building complex in different conditions of off-design full load, off-design part

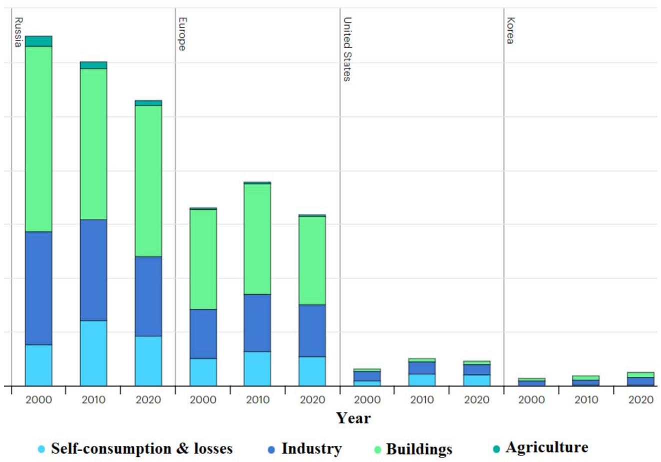

Fig. 1. District heating development in the different parts of the world [1].

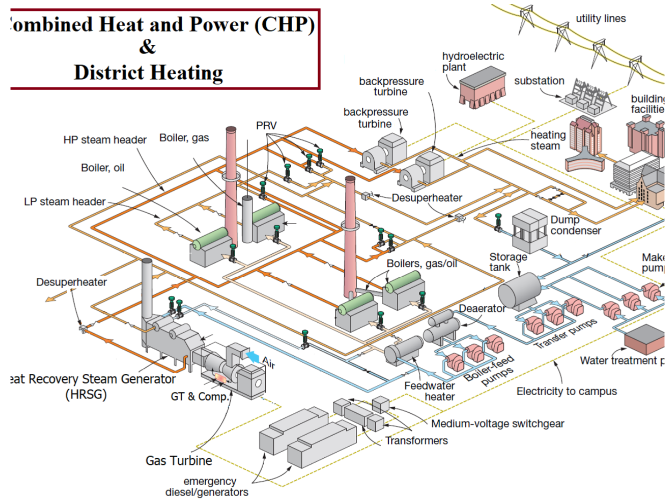

Fig. 2. The proposed CHP system network.

Table 1. Development trend of district heating technologies [2].

	1st Generation	2nd Generation	3rd Generation	4th Generation
Peak Technology Period	1880-1930	1930-1980	1980-2020	2020-2050
Heat Production	Steam boilers	CHP and heat only boilers	Large-scale CHP	Heat Recycling
Energy Source	Coal	Coal and oil	Biomass, waste, and fossil fuels	Renewable sources

load, and off-design medium load as a novel approach.

Thus, the objective of this study is to perform an accurate estimate of the heat that can be recycled flexibly based on heat or electrical load demand. Such an approach will improve the operating conditions of power plants and their multi-purpose services. In particular, this approach can effectively increase the level of competitiveness for distributed generation power plants. Energy and exergy analyses were performed on the proposed system in this study, and the energy consumption and efficiencies for each of the scenarios were obtained. In addition, the energy management conclusion of using this system from the electricity or heat point of view has been done, as well as in comparison with the situation of buildings heating supply by a specific capacity boiler.

2. CHP SYSTEM CONFIGURATION

The network structure of the CHP system considered in this study is shown in Fig 2. This system consists of a gas turbine to generate electricity, which in this study applies the heat recovered to generate electricity and heat in steam turbines. Therefore, the network has two steam turbines, one to generate electricity and another one to supply the required heat for the district heating system. Fossil-fueled boilers have also been presented as an old source of buildings heating in the selected area. In this energy network, a diesel generator is also provided to generate electricity in an emergency time. Also, hot and cold water transmission pipelines, control valves, water treatment and pumping systems, cycling water cooling systems, and power transmission line equipment are displayed in this configuration.

The model demonstrated in Fig. 3 shows that the steam supply network is fed solely by a heat recovery steam generator at the gas turbine exhaust. Fig. 3 shows a power generation system with a steam and hot water side network to provide district heating. Each of the subsystems is divided and named separately. The relationship between mass and energy flows of these subsystems is conceptually considered.

A gas turbine cycle consisting of air and fuel inlets, combustion chamber, compressor, turbine, and generator is considered in the topping gas cycle. The cycle capacity is 100 MW, and the exhaust heat is used in the next subsystem to generate steam.

Exhaust heat from gas turbines is used in HRSG heat exchangers to produce superheated steam. This flue gas is then directed to the hot water cycle heat exchanger and comes out of the stack. In this study, the HRSG is a single-stage type. It includes three heat exchangers, an economizer, evaporator, and superheater, located in the flue gas path, respectively, from last to first. The output of this system is steam with a temperature of 530°C and a pressure of 40 bar to produce work in the steam turbine.

Superheat steam expands in turbine No. 101 to generate power and then leaves it. The outlet steam is divided into two parts at connection point No. 102, part of which is directed to the heat exchanger No. 115 to supply the demanded heat of the district heating system, and the other part is expanded in turbine No. 104. The expanded output flow from this turbine is cooled in the water-cooling cycle and then transferred to the deaerator with the return water from the district heating system for recovery in the comprehensive system. This forms a closed water cycle.

The required heat in the district heating system is supplied from the top steam cycle and the hot water cycle. This heat is transferred to the circulating water in this system and is consumed in heatsink No. 401 by the intended parts, and then this water flow that its temperature has been reduced is returned to the closed cycle of heat supply by the pump.

3. DISTRICT HEATING SCENARIOS

One of the most critical factors related to the utilization of CHP systems in buildings is estimating the heat and electrical load demand model to manage energy supply. In this regard, the average annual consumption of electrical and thermal energy demand of the buildings in Tehran has been calculated and demonstrated in the diagrams in Fig. 4.

Therefore, considering the dependence of the amount of electricity and heat required in the areas around the power

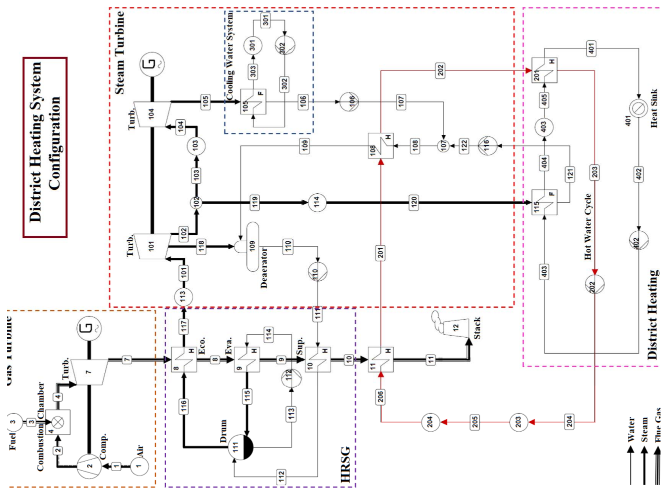

Fig. 3. The proposed model of CHP and district heating in this study.

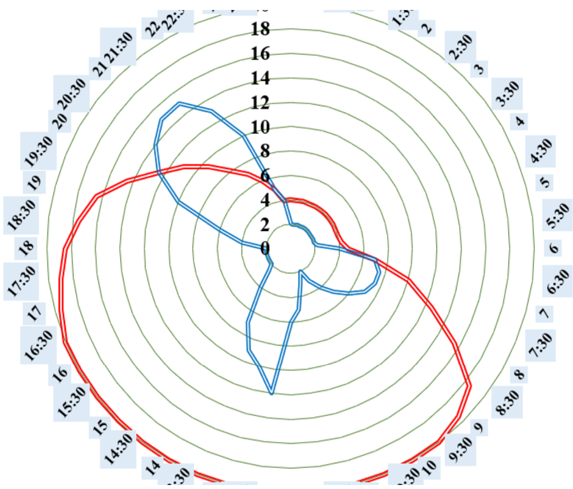

Fig. 4. The average hourly electricity and natural gas consumption pattern in a residential area [20].

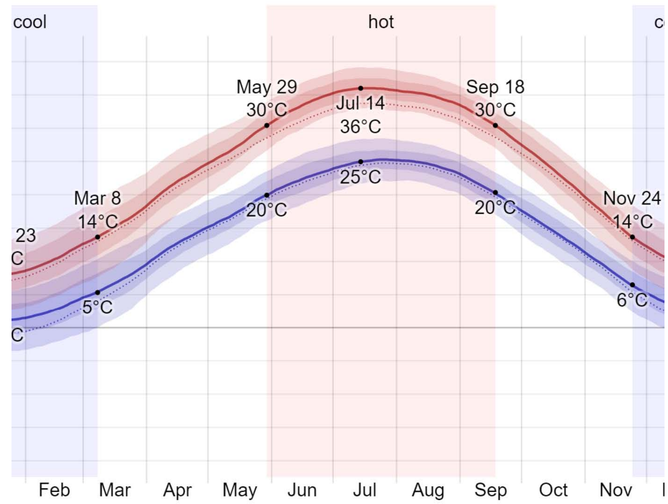

Fig. 5. The average annual temperature of Tehran [21].

plant to the hours of the day and seasons, in this study, five scenarios have been considered to exploit the potential in bottoming systems. Of these five scenarios, two are related to the CHP power plant's performance under design conditions, and three are off-design conditions. Off-design strategies have been investigated to be flexible in changing electrical or thermal energy demand.

A. Design scenarios

The methodology of this study is based on power plant operation scenarios in design and off-design conditions. In this way, the capacity of the power plant can be designed to generate maximum electricity or heat or simultaneously produce a certain amount of heat or electricity constantly; While novel scenarios of flexible operation of the power plant in accordance with the conditions of consumption, daily hours and different seasons of the year have been examined as an alternative strategy in this study.

Scenario ♯1.
In this scenario, the target is to provide the conditions for maximum power generation. Therefore in the heat exchanger at the junction of the steam cycle and the district heating system, no heat is transferred, and all steam coming out of turbine No. 101 is directed to turbine No. 104 to generate power.
Scenario ♯2.
In this scenario, to generate maximum heat in the district heating system, most of the steam leaving the steam cycle is transferred to heat exchanger No. 115. Technically, the proper amount of steam is passed through turbine blade No. 104.

B. Off-Design scenarios

Scenario ♯3.
In this scenario, the main goal is also generating maximum power, but with the difference that the capacities of the comprehensive system equipment were used to provide more heat to the district heating system.
Scenario ♯4.
In this scenario, the primary purpose is also generating maximum heat, but with the difference that by making changes to the capacity of the comprehensive system equipment to provide more power in the steam cycle.
Scenario ♯5.
In this scenario, the objective is to utilize the experience gained in modeling scenarios ♯3 and ♯4. The average heat consumption in the district heating system is calculated, and a more balanced system is created.

4. SYSTEM MODELING AND ASSUMPTIONS

Modeling of all scenarios has been fulfilled by Cycle-Tempo software. As mentioned earlier, the power generated in the gas cycle is equal to 100 MW and is constant in all scenarios considered in this study. Therefore, as the topping cycle, the fuel consumption in this study will be constant.
In the steam cycle, the high-pressure turbine is responsible for generating power. The outlet steam is used for two purposes: to generate additional power in the low-pressure turbine or provide the required heat for the district heating system.

The hot water stream is operated in the district heating system with a temperature of 75°C, and at its output, the temperature will decrease by 20 degrees. Due to the pressure drop equal to 2 bars in the pipelines, the pump circulates the flow with constant pressure.

The difference in the amount of thermal energy absorbed in the different scenarios proposed in this research is in mass flow in the district heating system. Therefore, calculating this value in each scenario based on adjustable parameters in the master plan is essential. Also, some surrounding variables such as temperature, pressure, and humidity at the inputs and outputs of the comprehensive cycle are considered the same in all scenarios to allow comparison and differentiation.

The uniqueness of each scenario was based on changes in some of the equipment so that the specific modeling of each case could be implemented more accurately. These changes include:

- Since the outlet temperature of the stack is known in design scenarios and unspecified in off-design scenarios, and this is effective in determining the mass flow of hot water cycle fluid, the performance of equipment No. 11 is different in these two categories.
- Calculating the mass flow of equipment No. 201 in scenarios in which maximum power generation is the primary goal is different from scenarios in which maximum heat consumption in the district heating system. Because calculating the mass and energy balance of the system is directly related to the specific percentage of fluid passing through the heat exchanger No. 105.
- The calculation of the water mass consumption in the cooling water cycle varies in different scenarios due to the dependence on the amount of steam output from turbine No. 104. This value is directly related to the set temperature for the circulating water in this cycle.

Exergy is defined as the amount of work (= entropy-free energy) a system can perform when brought into thermodynamic equilibrium with its environment.

Exergy analysis is an effective tool for district heating system analysis. By running it, we can connect the system with a general environment. The generalized exergy equation is written as follows [22].

$$Ex_{in} = Ex_{out} \rightarrow (m \times ex)_{in} + \left(1 - \frac{T_0}{T_{in}}\right) Q_{in} + W_{in} = (m \times ex)_{out} + \left(1 - \frac{T_0}{T_{out}}\right) Q_{out} + W_{out} \quad (1)$$

A. Turbines

In a turbine, shaft work occurs due to the expansion of the working fluid containing energy. Thus, the result of energy conversion in the production turbine is work that can be defined and measured as a system product [22].

$$Ex_{product} = P_{shaft} \quad (2)$$

To calculate the exergy in a turbine, the exergy changes caused by its thermomechanical activity must consider. For this purpose, it is necessary to estimate the exergy content of all turbine output flows. This is shown in (2):

$$Ex_{source} = Ex_{in}^{tm} - \sum Ex_{out}^{tm} = Ex_{in} - \sum Ex_{out} \quad (3)$$

The functional efficiency of the turbine is presented as (4):

$$\eta_{Ex,f(turbine)} = \frac{P_{shaft}}{Ex_{in} - \sum Ex_{out}} \quad (4)$$

B. Heat exchangers

Since the main purpose of heat exchangers is to provide the heat required for the primary fluid flow, the change in the exergy of the inlet and outlet of this fluid can be described as the exergy content of the heat exchanger products [22].

$$Ex_{product} = Ex_{p,out} - Ex_{p,in} \quad (5)$$

The same is true for the secondary fluid flow that is considered the source of the exergy supply in the heat exchanger. Thus, the change in fluid exergy at the inlet and outlet must be evaluated.

$$Ex_{source} = Ex_{s,in} - Ex_{s,out} \quad (6)$$

The heat exchanger's functional exergy efficiency could be written by (7):

$$\eta_{Ex,f(heat\,exchanger)} = \frac{Ex_{p,out} - Ex_{p,in}}{Ex_{s,in} - Ex_{s,out}} \quad (7)$$

C. Pumps

Exergy change in pumps can be estimated based on the amount of shaft work applied to the fluid for compression. Hence, the evolution of fluid exergy content at the inlet and outlet must be considered [22].

$$\eta_{Ex,f(pump)} = \frac{Ex_{out} - Ex_{in}}{P_{shaft}} \quad (8)$$

Since the electric motor provides the shaft work in the pump, so in calculation related to the pump exergy, the motor power can be written as follows:

$$\eta_{Ex,f(pump)} = \frac{Ex_{out} - Ex_{in}}{P_{electric}} \quad (9)$$

D. Deaerator

Since the primary fluid in the deaerator is the output fluid of the condenser, heat is transferred to it by the secondary fluid. Hence, to calculate the exergy, we can do it almost the same way as a heat exchanger [22].

$$Ex_{product} = \phi_{m,p} \times (ex_{out} - ex_{in}) = \phi_{m,p} \times ex_{out} - Ex_{p,in} \quad (10)$$

Where $\phi_{m,p}$ is the condensate heated mass flow. Similar to the previous one, the exergy calculation relation for the source side is written as follows:

$$Ex_{source} = \sum (\phi_{m,s} \times ex_{s,in}) - ex_{out} \times \sum \phi_{m,s} = \sum Ex_{s,in} - ex_{out} \times \sum \phi_{m,s} \quad (11)$$

Likewise, the functional exergy efficiency is presented as follows:

$$\eta_{Ex,f(deaerator)} = \frac{\phi_{m,p} \times ex_{out} - Ex_{p,in}}{\sum Ex_{s,in} - ex_{out} \times \sum \phi_{m,s}} \quad (12)$$

Table 2. Input parameters of the proposed CHP cycle [4–6].

Parameter	Value	Unit
Gas Cycle		
Compressor inlet temperature	288.15	[K]
Compressor inlet pressure	1.013	[bar]
Compressor pressure ratio	11	-
Compressor isentropic efficiency	0.86	-
Combustion chamber isentropic efficiency	0.99	-
Combustion chamber pressure loss	0.22	bar
Gas turbine isentropic efficiency	0.87	-
Fuel temperature in GC	15	[C]
Turbine inlet temperature	1100	[C]
Steam Cycle		
Temperature of superheated steam in HRSG	530	[C]
Pressure of superheated steam in HRSG	40	[bar]
Feedwater temperature	134.15	[C]
Pressure of feed water	46.5	[bar]
Circulation ratio of the drum	5	[%]
Inlet Pressure of turbine no. 104	1.5	[bar]
District Heating		
Outlet temperature of heat sink no. 401	55	[C]
Outlet temperature of heat exchanger no. 201	75	[C]
Outlet pressure of the pump no. 202	10	[bar]
Pressure loss in heat exchanger no. 201	0.5	[bar]

E. Drum

The output of the drum is saturated steam, which can be considered a product in exergy evaluations. The difference between the exergy amount in steam and the feed water is measured. Also, the exergy source of the drum is the difference between the input and output of the evaporator, which is applied to calculate the efficiency [22].

$$\eta_{Ex,f(drum)} = \frac{Ex_{steam,out} - Ex_{feedwater,in}}{Ex_{exaporator,out} - Ex_{evaporator,in}} \quad \textbf{(13)}$$

F. Combustion chamber

Fuel and oxidant react in the combustion chamber and transfer heat through the exhaust fluid. In this apparatus, exergy is of thermomechanical type, and to obtain its efficiency, the exergy content of flue gas, fuel, and input oxidant must be considered [22].

$$\eta_{Ex,f(combustion\ chamber)} = \frac{Ex^{tm}_{flue\,gas} - Ex^{tm}_{fuel} - Ex^{tm}_{oxidant}}{Ex^{ch}_{fuel} + Ex^{ch}_{oxidant} - Ex^{ch}_{flue\,gas}} \quad \textbf{(14)}$$

Input data of the main components of the integrated cycle are presented in Table 2.

5. USE OF TEMPERATURE ESTIMATES

To perform energy balance in the presented model subsystems in this study, it is crucial to evaluate the operating temperature in the equipment and the temperature of the hot and cold fluids flow. For this purpose, except scenario ♯1, the exact temperature of the outlet or inlet of apparatus Nos is in all scenarios. 403 and 206 have been estimated. To do this, an initial value is usually used to start the computational steps.

To achieve convergence in the iterative steps, the values for the previous and next flows must be considered with each other. For this purpose, each piece of equipment's mass balance and energy balance are fulfilled sequentially to obtain the correct values in the appropriate temperature range for the cycle performance.

Specific temperature estimates are considered in each of the scenarios, which are as follows:

Scenario ♯2: In this scenario, due to the importance of calculating the temperature of steam / hot water inlet to the district heating system, the inlet temperature to device 403 is assumed.

Scenario ♯3: In the hot water cycle, the temperature of flow No. 206 is calculated using the outlet temperature of device No. 204. This determines the amount of temperature that can be transferred from heat exchanger No. 11.

Scenarios ♯4 and ♯5: To regulate the heat receivable from the top cycle in the district heating system, the temperature value of equipment No. 403 has been estimated to determine the temperature values of the equipment before and after it in the calculation steps from the correct point.

6. RESULTS AND DISCUSSION

The principal modeling results and their pertained discussions have been presented. As mentioned before, determining the exact and final temperature of the inlet flow to the district heating system has been done in scenarios ♯ 2, ♯ 4, and ♯ 5, which is done by considering the amount of heat exchangeable in exchangers Nos. 115 and 201 in the proposed model. The same is done for the flows introduced in the hot water cycle in scenarios ♯ 3, ♯ 4, and ♯ 5.

The results of comprehensive cycle modeling presented in this research are widely dependent on the demand for electrical and thermal loads from power and heat generators. For this purpose, based on the daily hours and according to the conditions of the hot and cold seasons of the year, it is possible to manage the production of electricity or heat. Therefore, the present system can be well adapted to different conditions to maximize its performance benefits.

Obtaining maximum power or heat in a CHP system may be necessary in some cases. However, one of the models presented in this research is developed based on the average annual values of energy demand, which can have a better competitive advantage in energy management in terms of sustainable energy supply. The composition of air, fuel, and flue gas in the top cycle modeling is mentioned in Table 3.

The results obtained from modeling for system power and efficiencies in the maximum power design scenario are presented in Table 4.

According to the results shown in Table 4 and the explanations provided in section 3, the maximum amount of steam is consumed to generate electricity in the turbine. Therefore the operation of pump number 116 is zero, and the net electrical efficiency is obtained at 50.57%.

The estimated exergy values of the main equipment for scenario ♯ 1 have been shown in Table 5.
T-S diagram of the cycle has been demonstrated in Fig 6.
The modeling results of the proposed system for the maximum heat generated scenario are shown in Table 6. In Scenario ♯ 2, instead of a splitter No. 102, a control valve (Fig 7) is considered that provides a minimum of 2% steam for the operation of the turbine No. 104, and the rest of the steam flow to generate heat is forwarded directly to the heat exchanger No. 115.
As can be seen, the amount of heat received in heat sink No. 401 can be significantly increased, which has modified the CHP system's heat efficiency by more than 43%.
The estimated exergy values of the main equipment for scenario ♯ 2 have been shown in Table 7.
Based on what was mentioned in sections 3 and 4, in scenario ♯ 3, the exact amount of heat energy is received from heat exchanger No. 11 before the stack to heat the mass flow of water in the hot water cycle. It has been performed by considering a control valve in the hot water cycle, as presented in Fig 8.
As can be seen in Fig. 4 and 5, it is not always necessary to generate maximum power or heat during the hours of the day, in different seasons of the year, as well as in other parts of the world.
For example, heat production is significantly higher in cold geographical areas than in the tropics or in the warm seasons of the year; electricity generation is maximized to provide the required energy by cooling systems. Therefore, design scenarios can not uniformly meet the different needs of CHP power plants. Therefore, the off-design strategies in this study are evaluated as follows.
Mass flow in this scenario is similar to other cases, and thus at the lowest temperature, the same enthalpy required by the district heating system is achieved. Hence, heat sink No. 401 with a higher mass flow will supply the necessary hot water, and the total thermal efficiency will increase up to 58%. The results of this scenario are shown in Table 8.
The estimated exergy values of the main equipment for scenario ♯ 3 have been shown in Table 9.
In Scenario ♯ 4, both control valves 102 and 203 are applied to regulate the amount of steam used in the district heating system. Thus, while generating heat for more mass flow in heatsink number 401, the remaining capacity can generate power in the steam turbine. The results of this scenario are presented in Table 10.
In Scenario ♯ 5, utilizing the results obtained in the field of electricity and heat generation in the previous scenarios, the average required heat for the district heating system is provided by placing the control valve No. 203 in the hot water cycle. On the other hand, the produced steam capacity in the system is used to generate electricity. Therefore, in this scenario, the desired electrical efficiency is obtained simultaneously with the district heating system's minimum required heat. The results of this scenario are shown in Table 12.
The estimated exergy values of the main equipment for scenario ♯ 4 have been shown in Table 11.
Therefore, the results of this study show that controlling the output of electricity and heat of CHP power plants with the aim of optimal use in district heating can be severely effective in increasing the utilization of the power plant, appropriate to different operating conditions. Of course, it should be noted that both supply and demand sides of energy must be equipped with control and predicting systems to achieve maximum utilization.
The estimated exergy values of the main equipment for scenario

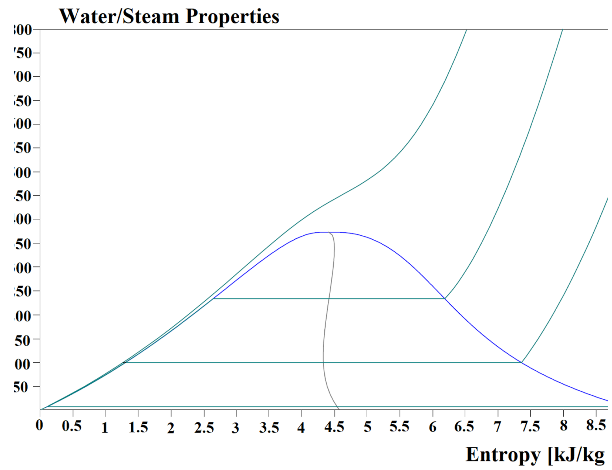

Fig. 6. Temperature-Entropy diagram of the proposed cycle.

♯ 5 have been shown in Table 13.
As shown in Fig. 9, the best balance in terms of electrical, thermal, and total efficiencies among the scenarios proposed in the present study is established in Scenario ♯ 5. The reason for this is the priority of generating electricity as the power plant's output. Therefore, the control of consumption in the district heating system leads to an increase of about 2% in the electrical efficiency of the power plant, which is a significant amount.
Variations in electrical, thermal, and total efficiencies of the proposed cycle (Scenario ♯ 5) for different annual temperature conditions in Tehran are shown in Fig. 10. According to the results, the thermal efficiency of the cycle decreases in the hot seasons of the year, and consequently, the efficiency of the district heating system also declines. This reduces the total efficiency of the integrated system during the warmer months of the year.
Also, based on the results presented in Table 14, the maximum possible energy efficiency in the proposed cycle is compared with the results of several samples from previous research. As can be seen, the proposed cycle efficiency at maximum heat consumption for district heating is an acceptable value compared to the results of other studies.
The amount of exergy destruction in the main components of the CHP and district heating system proposed in this study is displayed in Fig. 11. According to the results, the highest rate of exergy destruction is related to the combustion chamber of the gas turbine in the topping cycle. Similarly, turbines and compressors in the gas cycle are the second and third places of exergy destruction. The reason for this is the irreversibility of the conversion of fuel chemical energy into shaft work in this system. Heat recovery steam generator is another place where exergy destruction occurs. Among its heat exchangers, the evaporator contains a more significant amount due to the incompleteness of converting saturated liquid to saturated vapor in this equipment. By applying the intercoolers before the combustion chamber or heat regenerators in advanced power cycles, the exergy destruction in the system can be reduced.

7. CONCLUSIONS

This study established a comparison between different district heating scenarios classified in two design and off-design states.

Table 3. Air, natural gas, and flue gas compositions in the proposed system [23].

Component	Air	Natural Gas	Flue Gas
N2	0.7729	0.1432	0.7486
O2	0.2075	0.0001	0.1334
H2O	0.0101	-	0.075
Ar	0.0092	-	0.0088
CO2	0.0003	0.0089	0.0341
CH4	-	0.8129	-
C2H6	-	0.0287	-
C3H8	-	0.0038	-
C4H10	-	0.0015	-
C5H12	-	0.0004	-
C6H14	-	0.0005	-
Average Molemass [kg kmol-1]	28.85	18.64	28.45
LHV [kJ kg-1]	0	37994.81	0
HHV [kJ kg-1]	0	42104.15	0

Table 4. Power, heat, and efficiencies of scenario ♯ 1.

	Apparatus	Energy [kW]	Totals [kW]	Exergy [kW]	Totals [kW]
Absorbed	Fuel Source	304,780.70		318,368.80	
Power			304,780.70		318,368.80
Delivered	GT Generator	100,000		100,000	
Gross Power	ST Generator	55099.7		55099.7	
			155,099.70		155,099.70
Auxiliary Power	Pump (No. 106)	27.94		27.94	
	Pump (No. 110)	270.56		270.56	
	Pump (No. 112)	120.74		120.74	
	Pump (No. 116)	0		0	
	Pump (No. 202)	121.56		121.56	
	Pump (No. 302)	332.1		332.1	
	Pump (No. 402)	91.8		91.8	
			964.69		964.69
Net Power			154,135.09		154,135.09
Heat	Sink (No. 401)	18502.6		2769.7	
Total			172,637.70		156,904.80
Efficiency	Gross	50.88%		48.71%	
	Net	50.57%		48.41%	
	Heat	6.07%		0.87%	
	Total	56.64%		49.28%	

Table 5. Exergy values of scenario ♯ 1.

Apparatus	No.	Exergy transmitted [kW]	Exergy efficiency (%)
Turbine	7	217,494.90	95.14
Turbine	101	37415.2	93.64
Turbine	104	25772.7	81.43
Heat Exchanger	8	3379	84.11
Heat Exchanger	10	1642.3	84.05
Heat Exchanger	108	2076.2	58.28
Deaerator	109	51.3	93.38
Pump	106	-18	64.57
Pump	110	-225.7	83.43
Pump	112	-100.1	82.9
Pump	116	0	0
Pump	202	-95.3	78.47
Pump	302	-263.3	79.3
Pump	402	-69.1	75.34
Heat Exchanger	11	2964.3	75.69
Heat Exchanger	9	8408.8	81.14
Heat Exchanger	201	1637.4	62.35
Combustor	4	101,214.70	67.57
Drum	111	11.48	99.97
Compressor	2	-98071.1	93.05

Table 6. Power, heat, and efficiencies of scenario ♯ 2.

	Apparatus	Energy [kW]	Totals [kW]	Exergy [kW]	Totals [kW]
Absorbed	Fuel Source	304,780.70		318,368.80	
Power			304,780.70		318,368.80
Delivered	GT Generator	100,000		100,000	
Gross Power	ST Generator	35060.2		35060.2	
			135,060.20		135,060.20
Auxiliary Power	Pump (No. 106)	1.74		1.74	
	Pump (No. 110)	270.56		270.56	
	Pump (No. 112)	120.74		120.74	
	Pump (No. 116)	26.49		26.49	
	Pump (No. 202)	121.56		121.56	
	Pump (No. 302)	17.56		17.56	
	Pump (No. 402)	640.57		640.57	
			1199.12		1199.12
Net Power			133,861.11		133,861.11
Heat	Sink (No. 401)	132,971.75		19905.32	
Total			266,832.88		153,766.50
Efficiency	Gross	44.31%		42.42%	
	Net	43.92%		42.04%	
	Heat	43.62%		6.25%	
	Total	87.54%		48.29%	

Table 7. Exergy values of scenario ♯ 2.

Apparatus	No.	Exergy transmitted [kW]	Exergy efficiency (%)
Turbine	7	217,494.90	95.14
Turbine	101	37415.2	93.64
Turbine	104	1138.96	53.63
Heat Exchanger	8	3379	84.11
Heat Exchanger	10	1642.3	84.05
Heat Exchanger	108	513.8	76.75
Deaerator	109	51.3	93.38
Pump	106	-0.8	46.03
Pump	110	-225.7	83.43
Pump	112	-100.1	82.9
Pump	116	-17.01	64.2
Pump	202	-95.3	78.47
Pump	302	-12.59	71.68
Pump	402	-497.02	77.59
Heat Exchanger	11	2965.4	75.68
Heat Exchanger	9	8408.8	81.14
Heat Exchanger	201	2257.7	67.31
Combustor	4	101,214.70	67.57
Drum	111	11.48	99.97
Compressor	2	-98071.1	93.05

Table 9. Exergy values of scenario ♯ 3.

Apparatus	No.	Exergy transmitted [kW]	Exergy efficiency (%)
Turbine	7	217,494.90	95.14
Turbine	101	37396.3	93.61
Turbine	104	25303.6	81.44
Heat Exchanger	8	3379	84.11
Heat Exchanger	10	1642.3	84.05
Heat Exchanger	108	1738.3	58.16
Deaerator	109	119.16	90.63
Pump	106	-17.76	64.53
Pump	110	-225.6	83.43
Pump	112	-100.1	82.9
Pump	116	0	44.21
Pump	202	-93.5	77.85
Pump	302	-263.3	79.3
Pump	402	-94.2	76.69
Heat Exchanger	11	3894	70.62
Heat Exchanger	9	8410.7	81.14
Heat Exchanger	201	1602.2	69.75
Combustor	4	101,214.90	67.57
Drum	111	11.59	99.97
Compressor	2	-98071	93.05

Table 8. Power, heat, and efficiencies of scenario ♯ 3.

	Apparatus	Energy [kW]	Totals [kW]	Exergy [kW]	Totals [kW]
Absorbed	Fuel Source	304,780.70		318,368.80	304,780.70
Power			304,780.70		
Delivered	GT Generator	100,000		100,000	100,000
Gross Power	ST Generator	35060.2		54698.3	35060.2
			154,698.30		
Auxiliary Power	Pump	27.52		27.52	27.52
	(No. 106)				
	Pump	270.5		270.5	270.5
	(No. 110)				
	Pump	120.74		120.74	120.74
	(No. 112)				
	Pump	0		0	0
	(No. 116)				
	Pump	120.22		120.22	120.22
	(No. 202)				
	Pump	332.1		332.1	332.1
	(No. 302)				
	Pump	122.86		122.86	122.86
	(No. 402)				
			993.94		
Net Power			153,704.38		
Heat	Sink	25209.78		3773.81	25209.78
	(No. 401)				
Total			178,914.16		
Efficiency	Gross	50.75%		48.59%	50.75%
	Net	50.43%		48.27%	50.43%
	Heat	8.27%		8.27%	8.27%
	Total	58.70%		58.79%	58.70%

Table 10. Power, heat, and efficiencies of scenario ♯ 4.

	Apparatus	Energy [kW]	Totals [kW]	Exergy [kW]	Totals [kW]
Absorbed	Fuel Source	304,780.70		318,368.80	
Power			304,780.70		318,368.80
Delivered	GT Generator	100,000		100,000	
Gross Power	ST Generator	42977.7		42977.3	
			142,977.70		142,977.70
Auxiliary Power	Pump	1.75		1.75	
	(No. 106)				
	Pump	270.31		270.31	
	(No. 110)				
	Pump	120.73		120.73	
	(No. 112)				
	Pump	26.63		26.63	
	(No. 116)				
	Pump	121.13		121.13	
	(No. 202)				
	Pump	332.11		332.11	
	(No. 302)				
	Pump	606.27		606.27	
	(No. 402)				
			1478.92		1478.92
Net Power			141,498.81		141,498.81
Heat	Sink	125,754.45		18824.99	
	(No. 401)				
Total			267,253.25		160,323.80
Efficiency	Gross	46.91%		44.90%	
	Net	46.42%		44.44%	
	Heat	41.26%		5.91%	
	Total	87.68%		50.35%	

Table 11. Exergy values of scenario ♯ 4.

Apparatus	No.	Exergy transmitted [kW]	Exergy efficiency (%)
Turbine	7	217,494.90	95.14
Turbine	101	46994.3	92.96
Turbine	104	376.6	3.28
Heat Exchanger	8	3380.2	84.11
Heat Exchanger	10	1641.4	84.06
Heat Exchanger	108	485	80.9
Deaerator	109	37.3	94.21
Pump	106	-0.8	45.54
Pump	110	-225.5	83.43
Pump	112	-100.1	82.9
Pump	116	-17.1	64.19
Pump	202	-94.8	78.32
Pump	302	-263.3	79.3
Pump	402	-470	77.53
Heat Exchanger	11	3182.1	74.48
Heat Exchanger	9	8414.6	81.14
Heat Exchanger	201	2107.7	69.2
Combustor	4	101,214.90	67.57
Drum	111	11.43	99.97
Compressor	2	-98071	93.05

Table 12. Power, heat, and efficiencies of scenario ♯ 5.

	Apparatus	Energy [kW]	Totals [kW]	Exergy [kW]	Totals [kW]
Absorbed	Fuel Source	304,780.70		318,368.80	
Power			304,780.70		318,368.80
Delivered	GT Generator	100,000		100,000	
Gross Power	ST Generator	48593.6		48593.6	
			148,593.60		148,593.60
Auxiliary Power	Pump (No. 106)	14.8		14.8	
	Pump (No. 110)	270.33		270.33	
	Pump (No. 112)	120.73		120.73	
	Pump (No. 116)	13.5		13.5	
	Pump (No. 202)	120.82		120.82	
	Pump (No. 302)	332.1		332.1	
	Pump (No. 402)	363.56		363.56	
			1235.92		1235.92
Net Power			147,357.69		147,357.69
Heat	Sink (No. 401)	75,000		11227.23	
Total			222,357.69		158,584.92
Efficiency	Gross	48.75%		46.67%	
	Net	48.34%		46.28%	
	Heat	24.60%		3.52%	
	Total	72.95%		49.81%	

Table 13. Exergy values of scenario ♯ 5.

Apparatus	No.	Exergy transmitted [kW]	Exergy efficiency (%)
Turbine	7	217,494.90	95.14
Turbine	101	42708.2	93.19
Turbine	104	11649.7	82.45
Heat Exchanger	8	3380.1	84.11
Heat Exchanger	10	1641.5	84.06
Heat Exchanger	108	1183.5	68.14
Deaerator	109	68.2	92.67
Pump	106	-9.4	63.34
Pump	110	-225.5	83.43
Pump	112	-100.1	82.9
Pump	116	-8.5	62.99
Pump	202	-94.4	78.16
Pump	302	-263.3	79.3
Pump	402	-280.3	77.11
Heat Exchanger	11	3406.1	73.26
Heat Exchanger	9	8414.1	81.14
Heat Exchanger	201	1615.3	71.71
Combustor	4	101,214.90	67.57
Drum	111	11.48	99.97
Compressor	2	-98071	93.05

Table 14. Comparison of the maximum CHP efficiency of the present study with some previous research.

Reference No.	[24]	[25]	[26]	Present Study
Energy Efficiency	74.7	76%	90%	87.50%

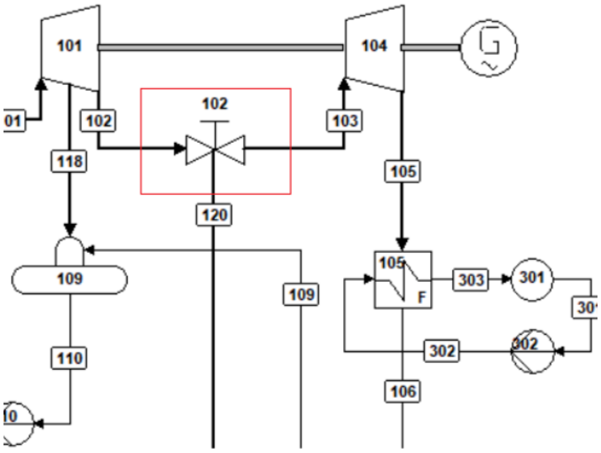

Fig. 7. The steam control valve of scenario ♯ 2 to allocate maximum heat.

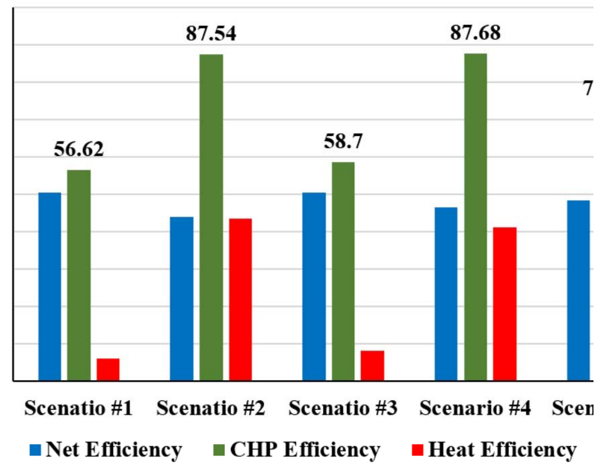

Fig. 9. The net electrical efficiency, heat, and total CHP efficiency for different scenarios of the proposed cycle.

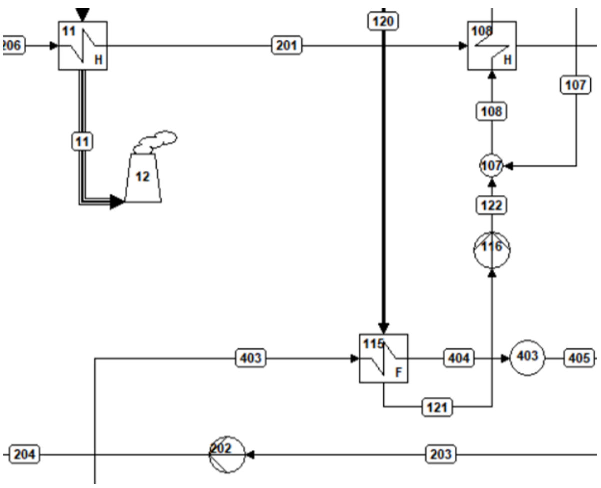

Fig. 8. The hot water control valve of scenario ♯ 3 to allocate the exact required heat.

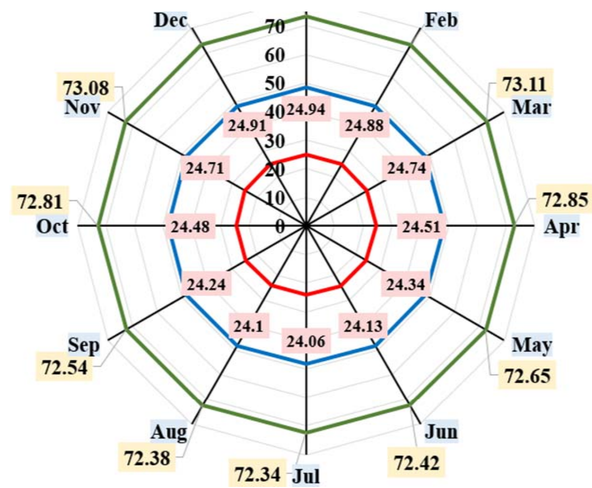

Fig. 10. The net electrical efficiency, heat, and total CHP efficiency change with the annual average temperature.

Both scenarios consist of maximum electrical or heating production cases to demonstrate how loads management is possible in various times and seasons to make an efficient local energy distribution system. This study uses a combined-cycle power plant as the proposed district heating system source.

The design approach achieved a maximum power generation of about 155 MW by scenario ♯ 1, where the electrical exergy efficiency is 48.71%. On the opposite side, the highest heat efficiency, more than 43%, will be obtained when the heat sink absorbs the maximum feasible heat in scenario ♯2.

The system could operate more flexibly in the off-design scenarios due to the considered controllers. For this reason, in scenario ♯ 3, although the maximum power is approximately available, the heat efficiency increased more than 30% compared with scenario ♯ 1. Also, the electrical efficiency has been improved by more than 2% in scenario ♯ 4 compared with scenario ♯ 2 while the maximum targeted heat was supplied. Moreover, the proposed model in scenario ♯ 5 indicates the feasibility of producing a high amount of heat and electricity at the same time that might be attractive for developers and consumers to find a reasonable balance for the most operational cases.

Finally, the most exergy destructive equipment in the proposed cycle are combustor, turbines, and compressor, respectively, mainly because thermomechanical irreversibility occurs in these parts compared with steam or hot water streams.

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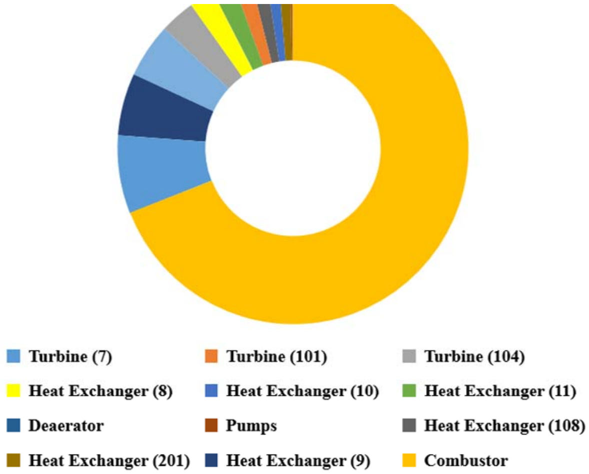

Fig. 11. The share of exergy destruction in the significant components of the proposed cycle.

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