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Energy Analysis of a CCHP-GSHP Hybrid System

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ABSTRACT

Simultaneous generation of electricity (CHP) and cogeneration systems (CCHP) are an attractive technology according to the principle of cascading energy. When combined with renewable energy, these systems can reduce the intensity of fossil fuel consumption and environmental impact. In this research, the combination of systems for simultaneous generation of heat and cold electricity and geothermal heat pump as two heat and cold load supply systems has been studied. Each of these systems can be used alone as a supplier of heat and refrigeration demand, but their combination can provide higher capabilities in the supply of heat and refrigeration and is more economical. In a study conducted on a hospital, it was found that the use of a combination of systems for the simultaneous production of electricity, heat and cold and geothermal heat pump will have more appropriate energy results. Simultaneous generation of electricity (CHP) and cogeneration systems (CCHP) are an attractive technology according to the principle of cascading energy. When combined with renewable energy, these systems can reduce the intensity of fossil fuel consumption and environmental impact.
Keywords: CCHP, GSHP, Energy Analysis, Geothermal Heat Pump

1. INTRODUCTION

Recently, some CHP / CCHP systems have been integrated with renewable energy. Wang et al. [1] proposed a CCHP system with biomass liquefaction gas and examined the annual operation of the project in terms of energy and exergy. Li et al. [2] proposed a CHP system based on natural gas and geothermal heat pump (GSHP) in series, and compared the performance of the proposed system with the CHP and GSHP systems separately. The results of this analysis showed that the proposed system has improved the efficiency of GSHP system and energy efficiency compared to separation systems. Bai et al. [3] proposed a new hybrid power generation system with a two-stage solar biomass degassing process, which adopted two different solar collectors as a heat source to meet the biomass degassing heat needs. In addition, the thermodynamic properties of the design were examined in both day and night modes, the proposed system showing some advantages over the single-stage degassing mode. Zhang et al. [4,5] proposed a new cogeneration system using biomass and solar energy as a power supply, and analyzed the system's performance in terms of energy and economics. Ezzat and Dinser [6] presented energy and exergy analysis of a geothermal energy-based system for power generation, refrigeration, hot air, hot water, and food drying, and the effect of system parameters on energy and exergy efficiency. brought. Tempesti et al. [7]

1.1.Equipment and systems

1.1.1. Simultaneous generation of electricity and heat systems

In designing and selecting cogeneration systems, the purpose of using this equipment must first be considered, ie due to the non-uniformity of demand in the electrical and thermal sectors (and refrigeration if needed), the system can not take into account the demand fluctuations of all sectors. Optimal response, so the main purpose of system design should be defined on only one of the system outputs. For example, in order to use the cogeneration system in a residential town, and due to the lack of heat generation support system, it is necessary if the cogeneration system is used, the system is designed to be able to maximize the heat load of the town in winter. To provide. However, if there is a heat supply support system, the cogeneration system can be designed in such a way that it can meet the basic or average heat load demand of the town, and in case of increased heat demand (on cold winter days), the support system can be used. Be. In this case, the operating efficiency of the system will certainly be higher than the case without using a backup system. Accordingly, the first step in designing the systems of simultaneous generation of electricity and heat is to study the demand for electrical and thermal energy of the place to be used to determine the most optimal scenario of energy supply.
The next step in designing simultaneous production systems is the selection of equipment and their type of arrangement. Simultaneous production systems are generally selected from the Topping cycle type. In this arrangement, the fuel is first burned in a gas turbine or internal combustion engine and then in a downstream heat exchanger converter of the exhaust gas flow of heat energy is recycled. Exhaust gases from gas turbines have a higher temperature than exhaust gases from internal combustion engines, so in applications that require the production of medium or high pressure steam, gas turbines are a more suitable option than internal combustion engines. The following figure shows two production systems simultaneously with the gas turbine power actuator and the internal combustion engine.

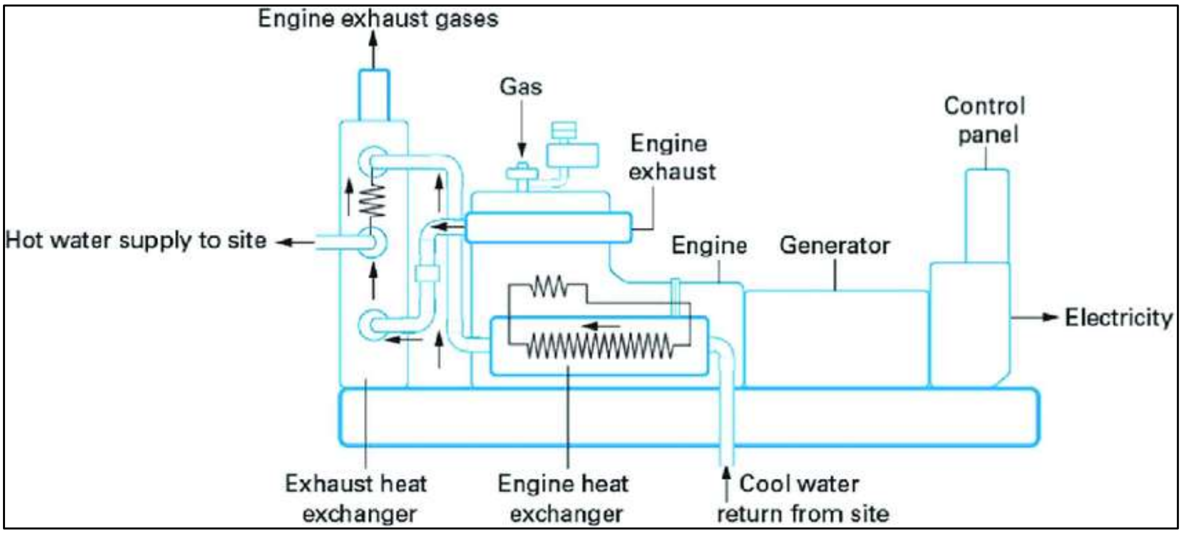

Fig. 1. Simultaneous generation of electricity and heat using an internal combustion engine

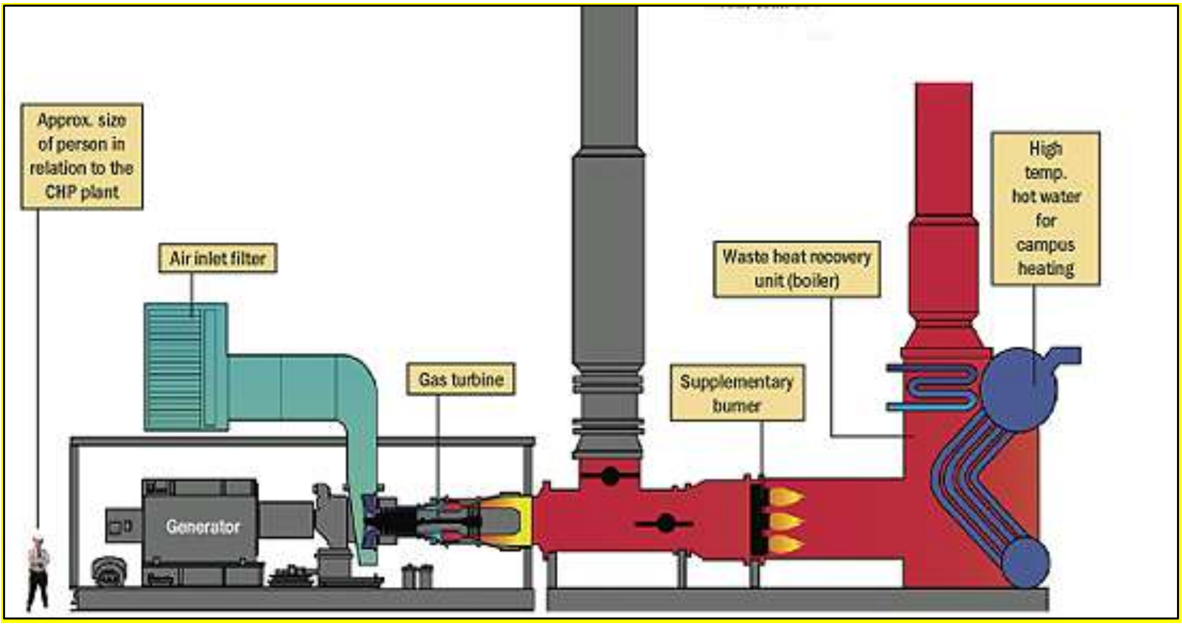

Fig. 2. Simultaneous generation of electricity and heat using a gas turbine.

The heat recovery section in cogeneration systems can have a simple or complex structure depending on the type of output fluid. If the output current from these systems is hot water, the recycled heat exchanger will be one or more tube heat exchangers. If the output fluid from these systems is steam, this section should include a superheater, evaporator, economizer and water preheater. In addition, peripherals such as gas strippers will be added to the system. These systems are called heat recovery boilers. Due to the high cost of heat recovery boilers in cogeneration systems, these equipments are commonly used in larger scale systems.

From the perspective of the first law of thermodynamics, the efficiency of a cogeneration system can be defined as follows:

$$\eta = \frac{P+Q}{F} \tag{1}$$

In the above relation, P is the electric power generated by the cogeneration system, Q is the heat recovered from the heat recovery system and F is the fuel energy entering the system.

Sometimes, in order to express the efficiency of a cogeneration system, the quality of the energy produced (electrical and thermal) is not considered equal and thermal energy is considered lower in quality than electrical energy. For example, the Public Service Regulatory Policy Act of 1978 in the United States defines the efficiency of a cogeneration system as follows:

$$\eta = \frac{P+\frac{Q}{2}}{F} \tag{2}$$

Another definition of cogeneration system efficiency is effective electrical efficiency, which is known as fuel efficiency. This measurement expresses the efficiency of the cogeneration system as the ratio of net electrical output to pure fuel consumption. This means that the net fuel consumption is equal to the amount of fuel entering the system, except for the part of the fuel that is used to generate useful heat, assuming a suitable efficiency for the boiler (usually 80%). This efficiency is as follows:

$$\eta = \frac{P}{F-\frac{Q}{\eta_q}} \tag{3}$$

In the above relation η_q is the average efficiency of a heat generation system such as a boiler. Fuel economy is one of the parameters used when discussing the advantages of simultaneous generation of electricity and heat over heating operations and the production of electrical power. Compares the fuel economy with the difference between the fuel consumption of the production system used simultaneously with the heating system and the production of power separately (eg boiler and electric-only production). The percentage reduction in fuel consumption can be estimated from the following equation:

$$S = 1 - \left[\frac{P}{\frac{P}{Eff_P}+\frac{Q}{Eff_Q}}\right] \tag{4}$$

In the above equation, the term inside the bracket defines the fuel used to generate electricity and steam in a cogeneration system. The denominator also describes the fuel used to generate electricity (P / EffP) and thermal energy (Q / EffQ) in separate heating and power operations.

Positive values indicate fuel savings, while negative values indicate that the CHP system uses more fuel than generating separate heat and power.

1.1.2. Gas turbine

In any cogeneration system, one piece of equipment is used as a power generator. The most widely used of these equipments are gas turbines and internal combustion engines. Gas turbines provide higher recycling temperatures than internal combustion engines and are therefore considered more suitable for many applications of cogeneration systems. Gas turbines are a combination of the three main components of a compressor, a combustion chamber and a turbine that create an open power generation cycle.

Another way to improve the efficiency of gas turbines is to increase the efficiency of cycle components by using other equipment such as cooling coolers between compressors, re-combustion of liquid gas inside the turbine after several stages of the turbine, heat exchanger between turbines or heat exchangers to heat the air. .

The compression process in an ideal compressor is an isentropic process, so the following isentropic relationship should be used to model the compressor.

$$T2 = T1 \ * \left(\frac{P2}{P1}\right)^{\frac{k-1}{k}} \tag{5}$$

In this regard, P1 and T1 will be the pressure and temperature of the gas before compaction and P2 and T2 will be the pressure and temperature of the gas after compaction.

The thermodynamic efficiency of a compressor can be estimated from the polytropic efficiency. As the polytropic efficiency is known, Equation (5) is modified as follows.

$$\mathrm{T2} = \mathrm{T1} \ * (\mathrm{P2/P1})^{\frac{\mathrm{k-1}}{\mathrm{k\,\eta poli}}} \tag{6}$$

As mentioned, the variable K is temperature dependent. Its average value for gas turbine cycle calculations before combustion is 1.4 and after combustion is 1.333.

To calculate the value of K in a turbine, one must first calculate Cp at the average temperature of the turbine. To calculate the Cp of combustion products, the fuel to air ratio must also be specified. CP Gases from burning liquid fuel (diesel) in dry air are as follows:

$$\mathrm{Cp} = \sum_{\mathrm{n=0}}^{8} An * Tz\hat{}(n) + \frac{\mathrm{FAR}}{\mathrm{FAR}+1}(\sum_{\mathrm{n=0}}^{7} Bn * Tz\hat{}(n)) \tag{7}$$

In this regard, the FAR ratio of fuel to air, coefficients A0 to A8, coefficients of Table 1 for dry air and constants B1 to B7 are as follows.

Table 1. Equation constants

B0	B1	B2	B3
-0.718874	8.747481	-15.863157	17.254096
B4	B5	B6	B7
-10.233795	3.081778	-0.361112	-0.003919

The Cp value for gas fuel combustion products is as follows.

$$Cp(gas) = (1.0001 + 0.9248 * FAR - 2.2078 * FAR^2) * Cp(liquid) \tag{8}$$

Using Equation (7), the Cp of combustion products in humid air can be calculated.

Combustion efficiency is a measure of the incompleteness of combustion. Incomplete combustion directly affects fuel consumption. Combustion efficiency is the ratio of the increase in actual enthalpy of combustion products to the heat of the inlet fuel theory.

$$\eta_{(com)} = \frac{\Delta h_{(actual)}}{\Delta h_{(theoretical)}} = \frac{\left(m^{\cdot}_{(air)} + m^{\cdot}_{(fuel)}\right)h_3 - m^{\cdot}_{(air)} * h_2}{m^{\cdot}_{(fuel)} * LHV} \tag{9}$$

By knowing the efficiency of the combustion chamber, the inlet fuel flow can be obtained by calculating the enthalpy of the inlet and outlet flows of the combustion chamber. To calculate the enthalpy of flow, the composition of its components must be known. During combustion, the composition of the gases changes. Table (3) presents the calculated values of the chemical reaction of one mole of fuel according to the following combustion equation:

$$CxHySz + \left(x + \frac{y}{4} + z\right) O2 \rightarrow xCO2 + \frac{y}{2}H20 + zSO \tag{10}$$

Table 2. Mole change of gases due to combustion of one mole of fuel

Fuel	The rate of change of moles of gas due to combustion					
	N2	O2	CO2	H2O	Ar	SO2
CH4	0	-2	1	2	0	0
C2H6	0	-3.5	2	3	0	0
C3H8	0	-5	3	4	0	0
C4H10	0	-6.5	4	5	0	0
C5H12	0	-8	5	6	0	0
C6H14	0	-9.5	6	7	0	0
N2	1	0	0	0	0	0
CO	0	-0.5	1	0	0	0
CO2	0	0	1	0	0	0
H2O	0	0	0	1	0	0
H2	0	-0.5	0	1	0	0
H2S	0	-1.5	0	1	0	1
He	0	0	0	0	1	0
O2	0	1	0	0	0	0
Ar	0	0	0	0	1	0

The pressure drop in the combustion chamber is an important issue because it affects both fuel consumption and output power. The pressure drop in the combustion chamber is in the range of 2 to 8% of static pressure. The uniformity of the outlet temperature profile affects the useful size of the inlet temperature to the turbine because the average inlet temperature to the turbine is limited by the maximum gas temperature. The average inlet temperature of the turbine affects both fuel consumption and output power.

1.1.3. Chiller

Absorption chillers are commonly used as cooling generators in co-generating systems for heat and cold. In this type of chillers, instead of using power, thermal energy is used to produce cold and they have less moving parts than compression types.

1.1.4. Geothermal heat pump systems

A heat pump or heat pump is a device that transfers energy from the point of origin (low temperature) to the point of destination that has a higher temperature. Heat is always moving from a warmer body to a colder body, but a heat pump picks up heat from a colder body or space and transfers it to a warmer body or space.

The heat pump can be used for cold and heat generation purposes. The following figure shows the T-S diagram of a heat pump cycle along with the diagram of a working cycle of a heat pump.

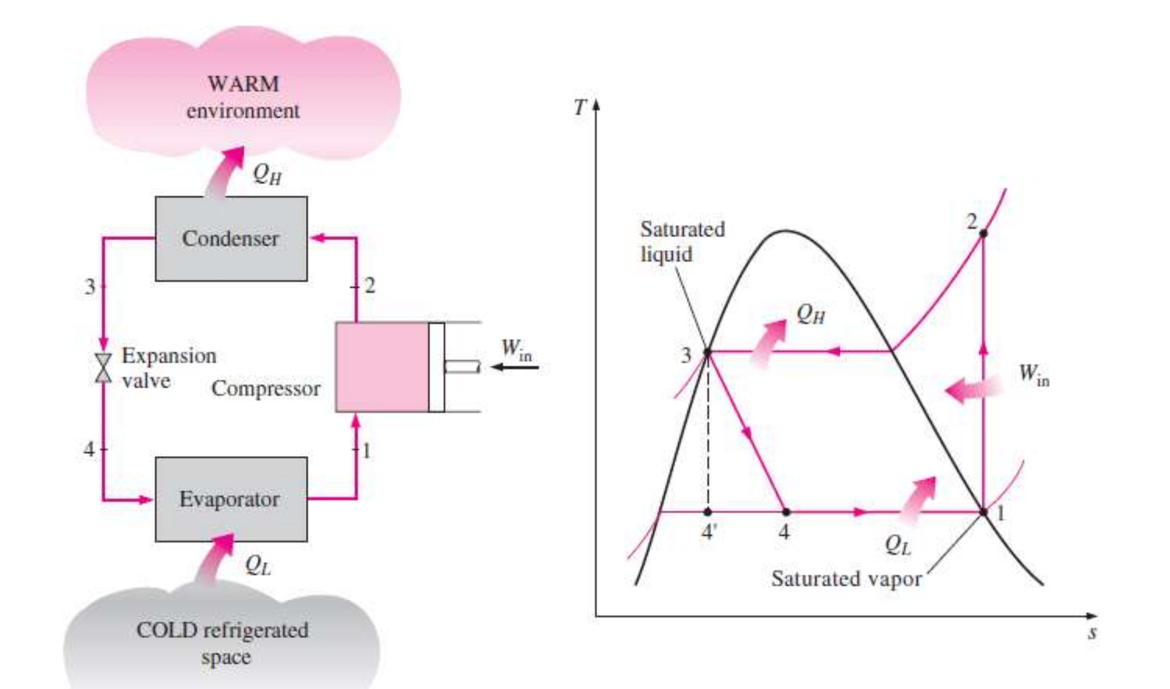

Fig. 3. T-S diagram and work diagram of heat pump or refrigerator cycle

In cold regions, the use of heat pumps is more for heating. Geothermal heat pumps are central devices that extract heat from the depths of the earth; And they use the ground in winter as a source of heat supply in summer as a source of heat dissipation.

1.2.Coupling of geothermal systems and simultaneous production of electricity and heat

Simultaneous generation of electricity and heat system and geothermal heat pump can be used in the supply of heat and refrigeration in parallel or in series. The higher the temperature difference between the hot and cold source temperatures, the higher the efficiency. Therefore, if the system heat demand is at a high temperature, the heat pump system can be used in series with the heat recovery system. In this case, we can expect the heat produced to have a higher quality than the separate operating mode of the two systems. In parallel mode, the two systems supply the heat and cold demand of the system in parallel with each other. In this case, the energy propositions and constraints of the system determine the priority of using either of these two systems.

1.3. Presenting hypotheses and basic information

In this research, modeling of coupling of two systems of simultaneous generation of electricity, heat and cold and geothermal heat pump has been done. Electricity generation is provided only in the cogeneration cycle but heating and cooling using the recycled energy of the turbine exhaust gas in the cogeneration cycle and the geothermal heat pump. The general layout of the equipment based on the original design is as follows.

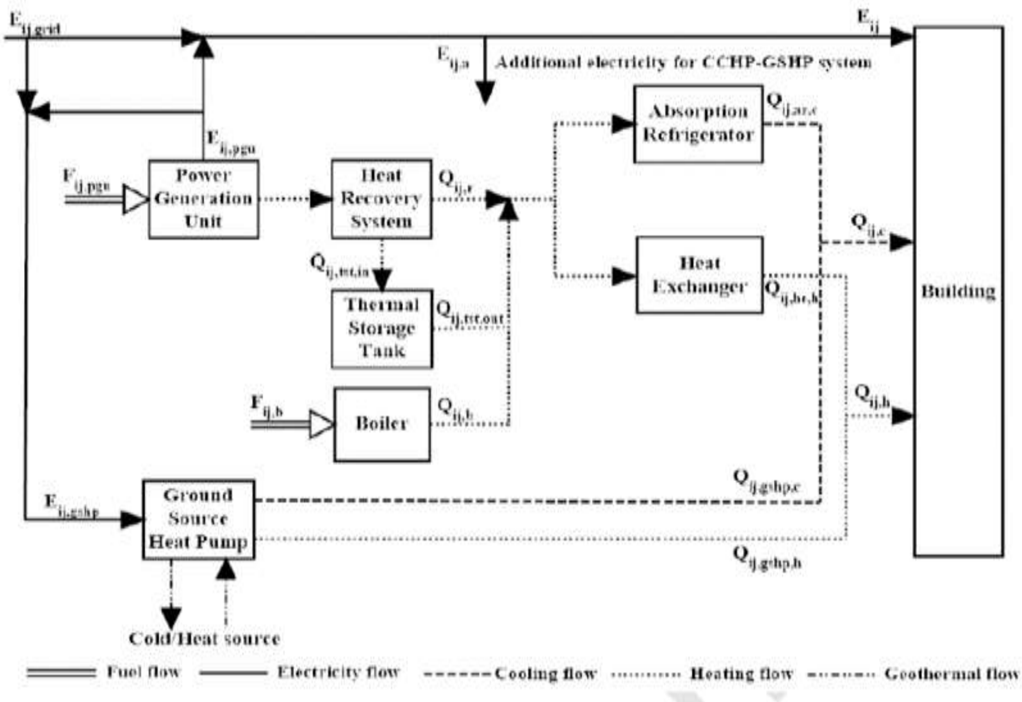

Fig. 4. Diagram of the studied cycle

In the above figure, in order to provide thermal and refrigeration energy, heat storage tank and boiler are also used. These equipments are not among the main equipments of the cycle, but they can be used in case of no need for electric load demand or increased demand for thermal and refrigeration load. The combination of use or capacity selection of each equipment in the above cycle depends on economic studies. In any engineering design or simulation process, appropriate assumptions must be made in order to create the right design.

Table 3. Specifications of gas turbines used in the cogeneration cycle

Compressor pressure ratio		12.7
Turbine inlet temperature	C	1038
Gas turbine outlet temperature	C	525
Compressor inlet air flow	Kg/s	11
Output power	kW	2000
Gas turbine efficiency	%	24.6

-Design of heat recovery system: based on pinch temperature equal to 20 ° C.
- Auxiliary boiler capacity: equal to the steam production capacity in the power cycle.
- Geothermal system design capacity: thermal and refrigeration power equal to the thermal power of the power system.
- Geothermal system heat pump: compression cycle, and the temperature of water returning from the earth is equal to 12 degrees Celsius.
-Absorption chiller: single effect using low pressure steam as heat source and COP equal to 0.67.

1.3.1. Simulated cycle

The simulated cycle in Thermoflow software consists of 4 parts. The first part is the gas turbine and its heat recovery system, the second part is the heat boiler, the third part is the absorption chiller and the end part is the geothermal heat pump system.

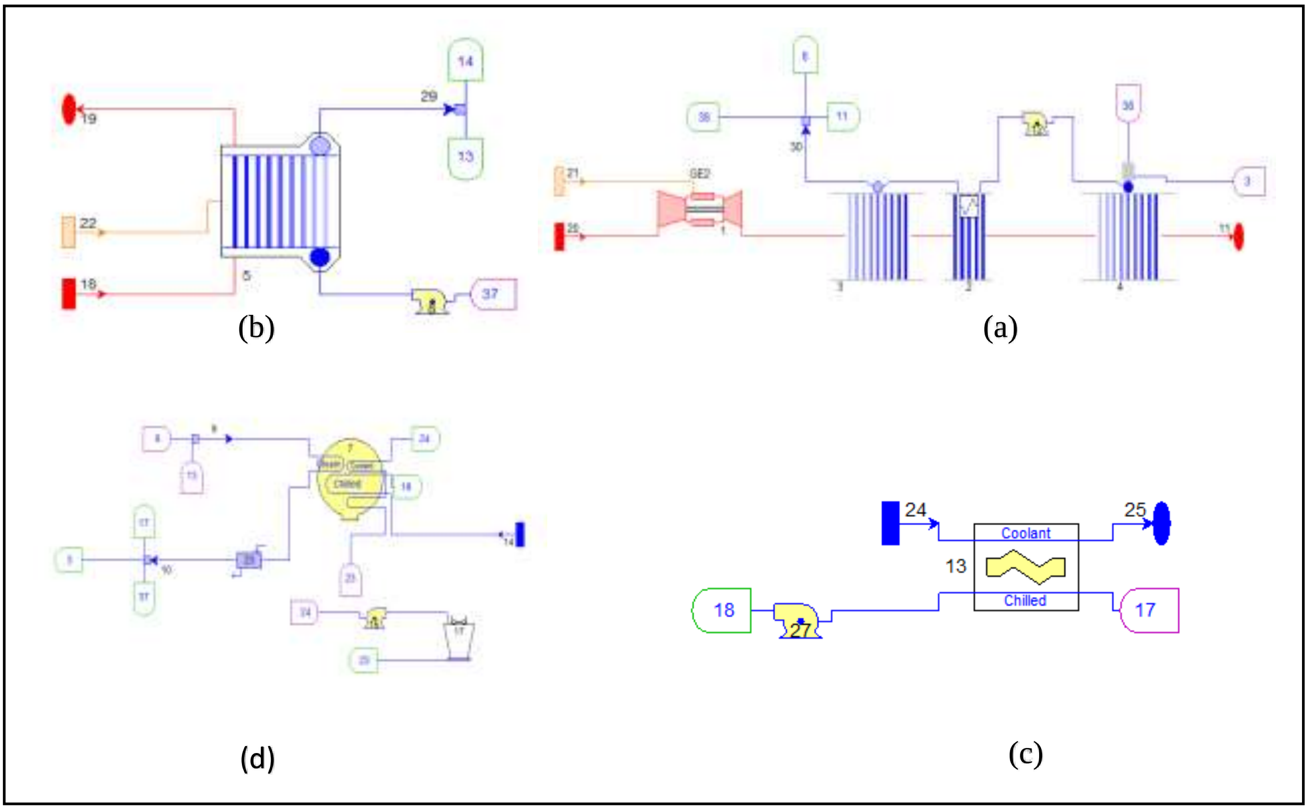

Fig. 5. Different parts of the simulated cycle: a) Power system and heat recovery (b Boiler system c) Geothermal heat pump system d) Absorption chiller system.

In the figure above, the identical number symbols indicate the communication flow between these systems. For example, the signs 13 and 8 in the above figures indicate the heat flows entering the absorption chiller.

1.3.2. Simulation results

Table 2 summarizes the cycle report in design mode. Empowerment services for different parts of the cycle, efficiency of all cycles, capability of cycle components, price, price and various parameters in the summary of the advanced program.

Table 3. Summary of cycle results in design mode

<table>
<tr><td colspan="8">SYSTEM SUMMARY</td></tr>
<tr><td colspan="8">Steam Property Formulation - IFC-67</td></tr>
<tr><td colspan="8">Ambient pressure = 1.013 bar Temperature = 15 C RH = 60 %</td></tr>
<tr><td colspan="8">Program revision date: February 25, 2013</td></tr>
<tr><td colspan="8"></td></tr>
<tr><th></th><th colspan="2">Unit</th><th colspan="3">LHV</th><th colspan="2">HHV</th></tr>
<tr><td>Net fuel input</td><td colspan="2">[kW]</td><td colspan="3">8055</td><td colspan="2">8918</td></tr>
<tr><td>Gross heat rate</td><td colspan="2">[kJ/kWh]</td><td colspan="3">14913</td><td colspan="2"></td></tr>
<tr><td>Net heat rate</td><td colspan="2">[kJ/kWh]</td><td colspan="3">19140</td><td colspan="2">21190</td></tr>
<tr><td>Gross electric efficiency</td><td colspan="2">[%]</td><td colspan="3">24.14</td><td colspan="2"></td></tr>
<tr><td>Net electric efficiency</td><td colspan="2">[%]</td><td colspan="3">18.81</td><td colspan="2">16.99</td></tr>
<tr><td>CHP efficiency</td><td colspan="2">[%]</td><td colspan="3">24.09</td><td colspan="2"></td></tr>
<tr><td>PURPA efficiency</td><td colspan="2">[%]</td><td colspan="3">21.45</td><td colspan="2"></td></tr>
<tr><td colspan="8"></td></tr>
<tr><td>Gross power</td><td colspan="2">[kW]</td><td colspan="3">1944.4</td><td colspan="2"></td></tr>
<tr><td>Net power</td><td colspan="2">[kW]</td><td colspan="3">1515.1</td><td colspan="2"></td></tr>
<tr><td>Total auxiliaries</td><td colspan="2">[kW]</td><td colspan="3">429.4</td><td colspan="2"></td></tr>
<tr><td>Net process heat output</td><td colspan="2">[kW]</td><td colspan="3">425.2</td><td colspan="2"></td></tr>
<tr><td colspan="8">POWER DEVICE(S)</td></tr>
<tr><th>Generator</th><th>Component</th><th>Shaf t No.</th><th>Component/ Shaft [kW]</th><th>Eff [%]</th><th>Mult iplier</th><th>Gen [kW]</th><th>Account ed [kW]</th></tr>
<tr><td></td><td>Gas Turbine (GT PRO)[1]</td><td></td><td>2042.2</td><td></td><td></td><td></td><td></td></tr>
<tr><td>Generator [1]</td><td></td><td>1</td><td>2042.2</td><td>95.2 1</td><td>1</td><td>1944. 4</td><td>1944.4</td></tr>
<tr><td>Total Generator (s)</td><td></td><td></td><td></td><td></td><td></td><td>1944. 4</td><td>1944.4</td></tr>
<tr><td colspan="8">AUXILIARY DEVICE(S)</td></tr>
</table>

Component	**Component/Shaft [kW]**	**Multiplier**	**Aux [kW]**	**Accounted [kW]**
Cooling Towers(various) fan/pump	7.5	1	7.5	7.5
Electric Chiller(PCE) aux	395.6	1	395.6	395.6
Gas Turbine(GT PRO)[1]: aux		1	4	4
Package Boiler(PCE)[5]: aux	0	1	0	0
Pump(PCE)	0.1	1	0.1	0.1
Pump(PCE)	0	1	0	0
Pump(PCE)	1.9	1	1.9	1.9
Pump(PCE)	0.9	1	0.9	0.9
Total components auxiliaries			409.9	**409.9**
Total miscellaneous auxiliary				**19.4**
Total plant auxiliary				**429.4**

The table above introduces three efficiencies for the cycle. Electrical efficiency, cogeneration efficiency and PURPA efficiency. Figure 3 simulates the cycle diagram and Table 3 presents the characteristics of all cycle currents including temperature, flow, pressure and enthalpy.

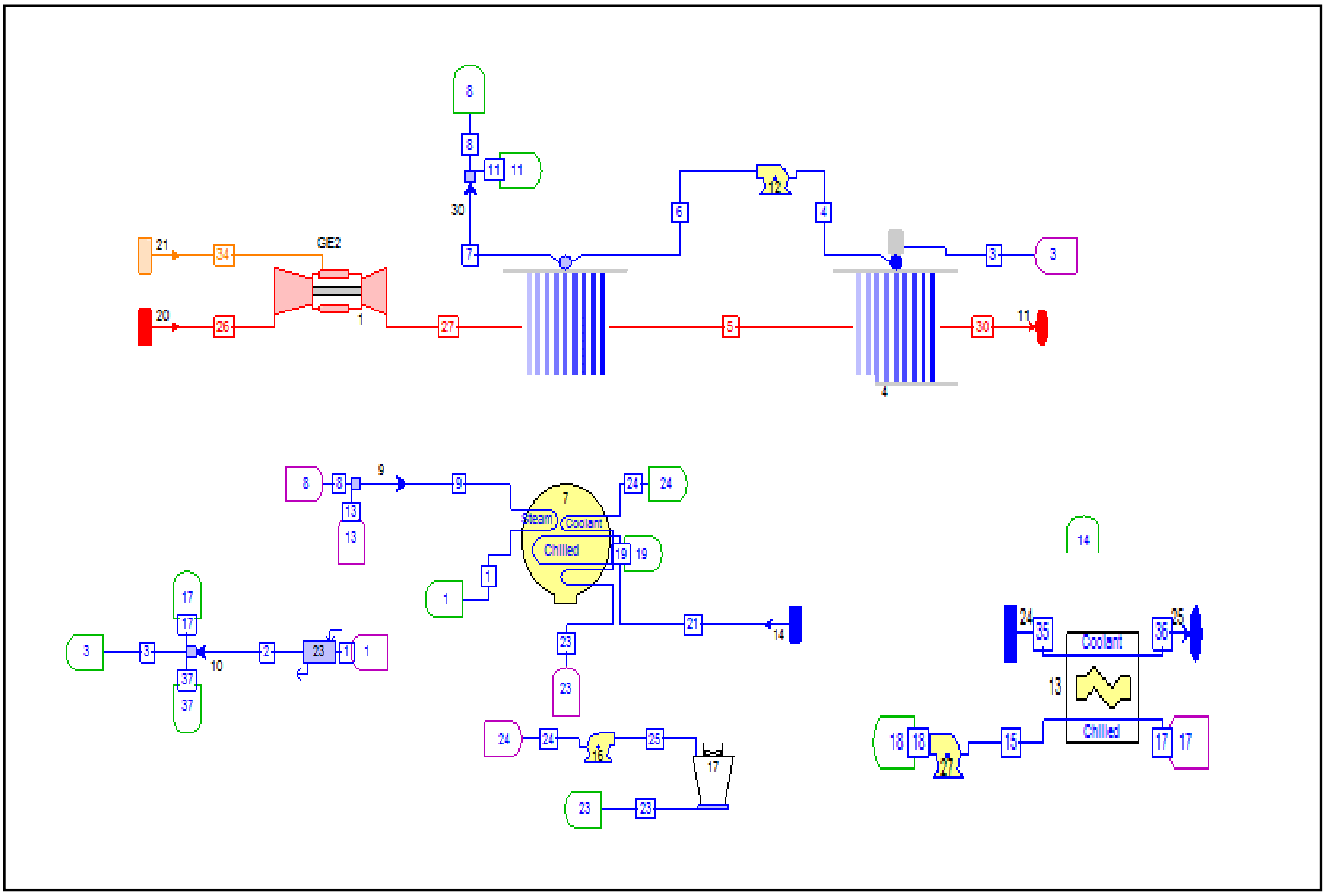

Fig. 6. Simulated cycle flow diagram

Table 4. Simulated cycle flow characteristics

Stream	Fluid	P	T	M	H*	H
		bar	C	kg/s	kJ/kg	kJ/kg
1	Water	0.1967	39.73	0.191	-2381.19	166.3
2	Water	0.1967	59.73	13.34	-2297.56	249.93
3	Water	0.1967	59.73	1.929	-2297.56	249.93
4	Water	0.1967	59.73	1.929	-2297.56	249.93
5	Gas/Air	1.0132	124.44	10.59	103.9	
6	Water	1.185	59.74	1.929	-2297.43	250.06
7	Water	1.185	104.44	1.91	135.22	2682.71
8	Water	1.185	104.44	0.191	135.22	2682.71
9	Water	0.1967	98.19	0.191	135.22	2682.71
10	Water	1.185	104.44	0	135.22	2682.71
11	Water	1.185	104.44	1.719	135.22	2682.71
12	Water	2.564	59.68	0	-2297.56	249.93
13	Water	1.185	104.44	0	135.22	2682.71
14	Water	1.185	104.44	0	135.22	2682.71
15	Water	0.0983	7	11.41	-2518.07	29.41
16	Water	0.5068	7	22.83	-2518.03	29.46
17	Water	0.1967	59.73	11.41	-2297.56	249.93
18	Water	0.5068	7	11.41	-2518.02	29.47
19	Water	0.5068	7	11.41	-2518.03	29.46
20	Water	1.185	104.44	1.719	135.22	2682.71
21	Water	1.0135	15	11.41	-2484.46	63.03
23	Water	1.0132	15.82	20.74	-2481.01	66.48
24	Water	0.5066	25.82	20.74	-2439.24	108.24
25	Water	1.0132	25.83	20.74	-2439.18	108.31
26	Gas/Air	1.0132	15	10.42	-10.13	
27	Gas/Air	1.0193	528.16	10.59	546.05	
30	Gas/Air	1.0132	124.44	10.59	103.9	
31	Gas/Air	1.0132	15	0	-10.13	
32	Gas/Air	1.0132	184.44	0	177.46	
33	Fuel	1.724	25	0	46280.22	
34	Fuel	20.68	25	0.174	46280.22	
35	Water	2.014	12	68.79	-2496.92	50.57
36	Water	1.325	22	68.79	-2455.15	92.34
37	Water	0.1967	59.73	0	-2297.56	249.93

The following figures show the modeling results for some of the main cycle equipment such as gas turbines and absorption chillers

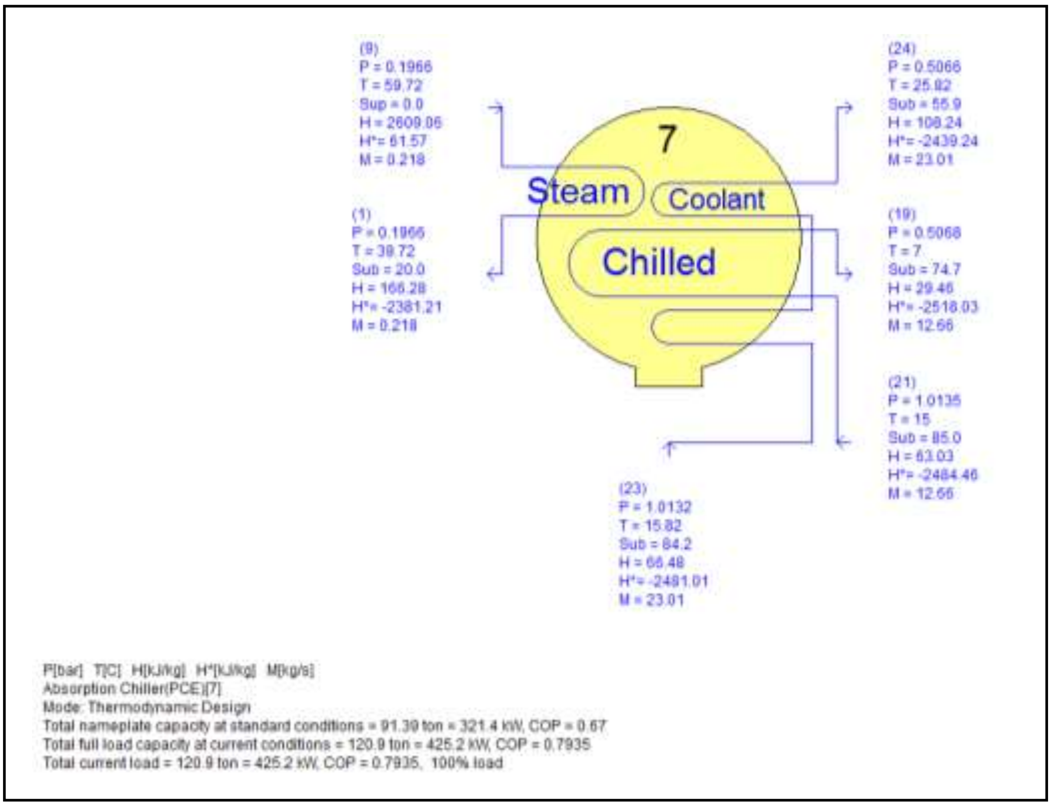

Fig. 7. Design specifications of absorption chiller in thermoflow software

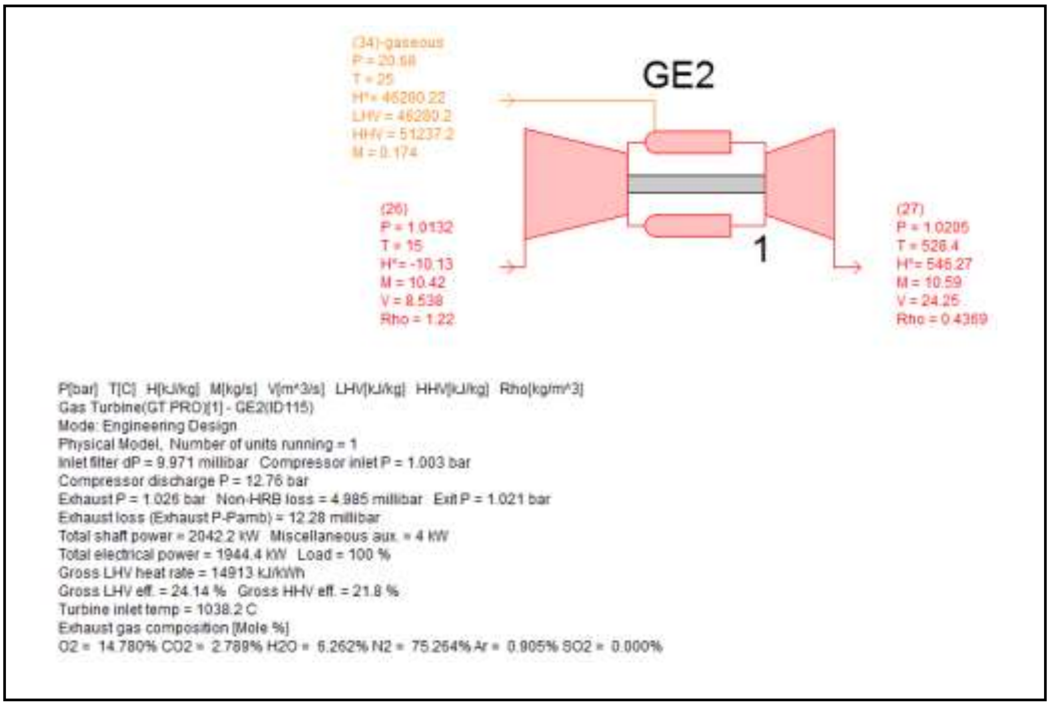

Fig. 8. Gas turbine design specifications in Thermoflow software

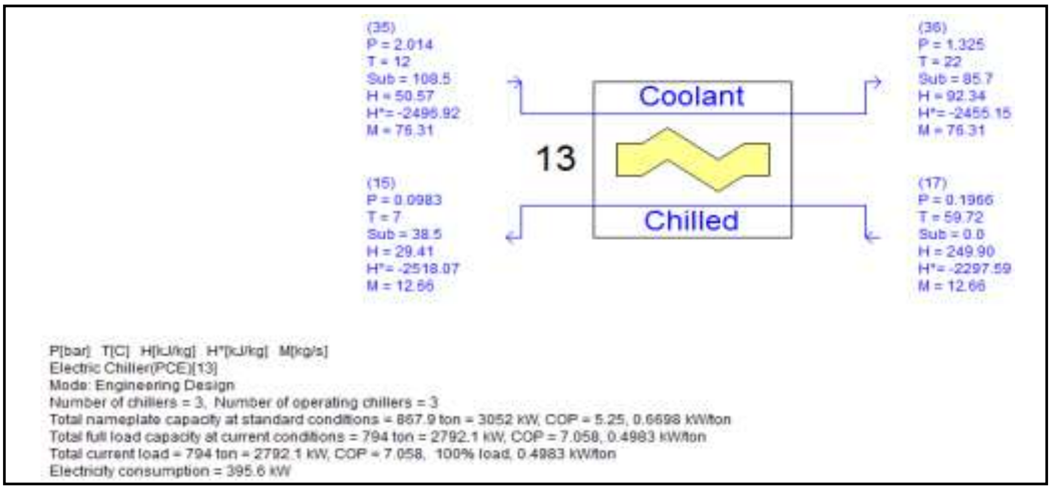

Fig. 9. Design specifications of heat pump in thermoflow software (summer operation)

1.3.3 . Gas turbine operating load

Due to the fact that in the above cycle, only the gas turbine plays the role of generating electricity, the amount of operating load of this equipment can affect the efficiency of the cycle. In the sensitivity analysis, the cycle efficiency in different turbine loads from 50 to 100% in both use and non-use of auxiliary boiler has been investigated and reported.

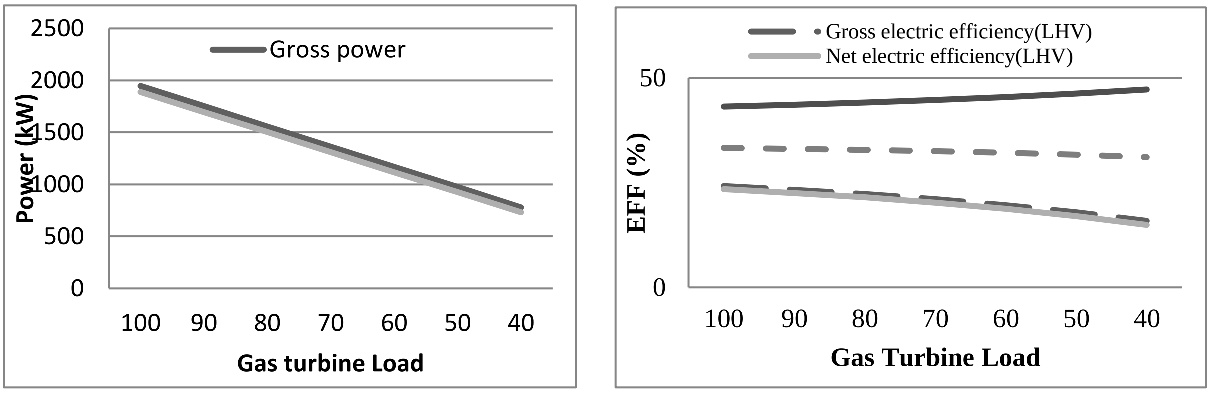

Fig. 10. Cycle efficiency at different loads of gas turbines when no auxiliary boiler is used.

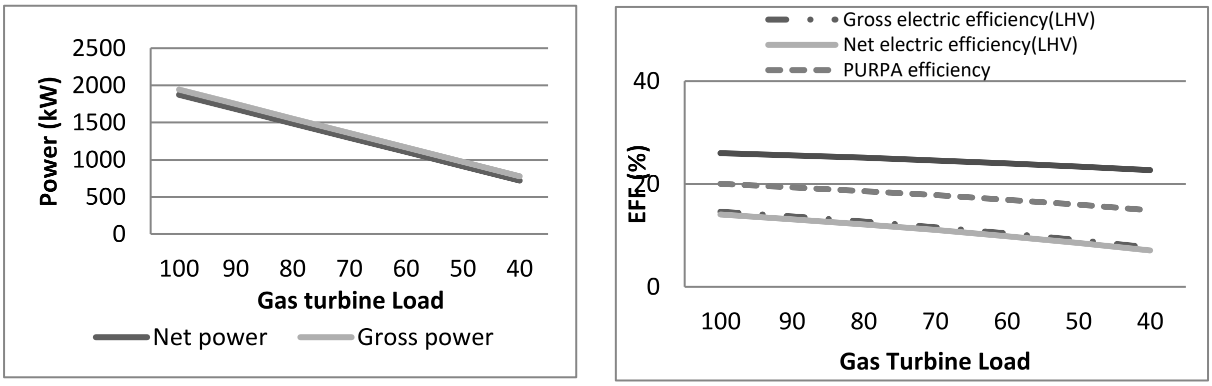

Fig. 11. Power and cycle efficiency at different gas turbine loads when using an auxiliary boiler

As can be seen in the figure above, in the case of not using the auxiliary boiler and at low loads of the gas turbine, the total production efficiency increases. These changes are due to the reduction of cycle fuel consumption while maintaining the production of heating and cooling energy, the result of which is an increase in efficiency.

1.3.4. Earth temperature

One of the parameters affecting the geothermal heat pump system is the ground temperature. This temperature changes according to Figure 3-8. The following table reports the effect of ground temperature change on various cycle parameters and geothermal heat pumps.

Table 5. Investigation of the effect of ambient temperature on the components of the cycle and geothermal heat pump.

Water/Steam SourceTemperature	**C**	**14**	**13**	**12**	**11**	**10**
CHP efficiency	%	24.01	24.05	24.08	24.11	24.14
Net power	kW	1550.3	1553.7	1556.6	1559.1	1561.1
Net electric efficiency(LHV)	%	19.25	19.29	19.33	19.36	19.38
Electric Heatpump (PCE)[13]: aux	kW	362.9	359.6	356.6	354.2	352.2
Electric Heatpump(PCE)[13] Total nameplate capacity at standard conditions	ton	749.8	765	782.4	802.3	825.1
Current COP		6.936	7.001	7.058	7.107	7.147
Electric Chiller(PCE	kW/ton	0.507	0.5023	0.4983	0.4948	0.4921
Water/Steam SourceTemperature	**C**	**9**	**8**	**7**	**6**	
CHP efficiency	%	24.16	24.17	24.18	24.18	
Net power	kW	1562.6	1563.6	1564.2	1564.3	
Net electric efficiency(LHV)	%	19.4	19.41	19.42	19.42	
Electric Heatpump (PCE)[13]: aux	kW	350.7	349.6	349.1	349	
Electric Heatpump(PCE)[13] Total nameplate capacity at standard conditions	ton	851.1	880.7	914.8	953.9	
Current COP		7.178	7.199	7.211	7.213	
Electric Chiller(PCE	kW/ton	0.49	0.4885	0.4877	0.4875	

1.4.1. Use and non-use of auxiliary boiler

The auxiliary boiler in the cycle is responsible for supplying the peak heat and cold load and can not be used in normal operation. Since the fuel used in this boiler is natural gas, whether or not to use it will greatly affect the efficiency of the cycle. In the table below, the modeling results are compared in two modes: boiler on and boiler off.

Table 6. Check the use or non-use of auxiliary boiler in the cycle

	Use of boiler	unit	Do not use the boiler
Fuel consumption rate	0/28	Kg/s	0/17
Fuel energy consumption	13312	kW	8055
Electric power (gross)	1943	kW	1974
Electric power (net)	1873	kW	1889
Electrical efficiency (gross)	14/6	%	24/18
Electrical efficiency (net)	14/07	%	45/23
PURPA efficiency	20/03	%	33/3
Cogeneration efficiency	25/99	%	43/15

1.4.2. Sharing heat and cold consumption

Based on the above table, it can be said that the use of auxiliary boilers in the cycle reduces the cycle efficiency. Therefore, the system should be designed and operated in such a way that the working time of this equipment is minimized. In one analysis, the ratio of thermal energy to refrigeration has changed and its effect on cycle parameters has been investigated. The results of this study are reported in the table below.

Table 7. Investigation of the effect of changing heat and cold demand on cycle performance

Cooling /Heating Load		0.187	0.210	0.452	0.810	1.396	2.528	5.780
Gross power	kW	1943.1	1943.1	1943.1	1943.1	1943.1	1943.1	1943.1
Net power	kW	1873.3	1869.6	1837.5	1805.5	1773.6	1741.6	1709.7
Gross electric efficiency(LHV)	%	14.6	14.6	14.66	14.73	14.79	14.85	14.91
Net electric efficiency(LHV)	%	14.07	14.05	13.87	13.68	13.5	13.31	13.12
CHP efficiency	%	25.99	27.19	37.63	48.15	58.75	69.45	80.24
PURPA efficiency	%	20.03	20.62	25.75	30.92	36.13	41.38	46.68
Energy chargeable to power	kW	11606	11425	9865	8305	6745	5186	3627
Electric efficiency on chargeable energy	%	16.14	16.36	18.63	21.74	26.29	33.58	47.14

As can be seen, the coefficient of cogeneration increases with increasing the ratio of cooling to heating load.

1.4.3. Using the cycle in real applications

One of the best applications of the hybrid system studied in this dissertation is the use of this system in meeting the thermal and refrigeration needs of a building complex. The thermal and refrigeration loads of the building mainly require low operating temperatures, and the use of cogeneration systems and geothermal heat pumps in this type of use can be economical.

1.5.1. Introducing the building and energy consumption diagrams

In this project, a hospital complex has been selected to evaluate the efficiency of the hybrid cogeneration system and geothermal heat pump. In this hospital, thermal energy consumption is always required with the aim of supplying sterilization steam, sanitary hot water, heating and supplying the energy of the absorption chiller. There is no limit in the field of electrical energy because it is possible to sell surplus electricity to the grid. The thermal energy consumption of this complex is shown separately in the figure below. Approximately 50% of the hospital's refrigeration load is supplied by absorption chillers and the rest by compression chillers.

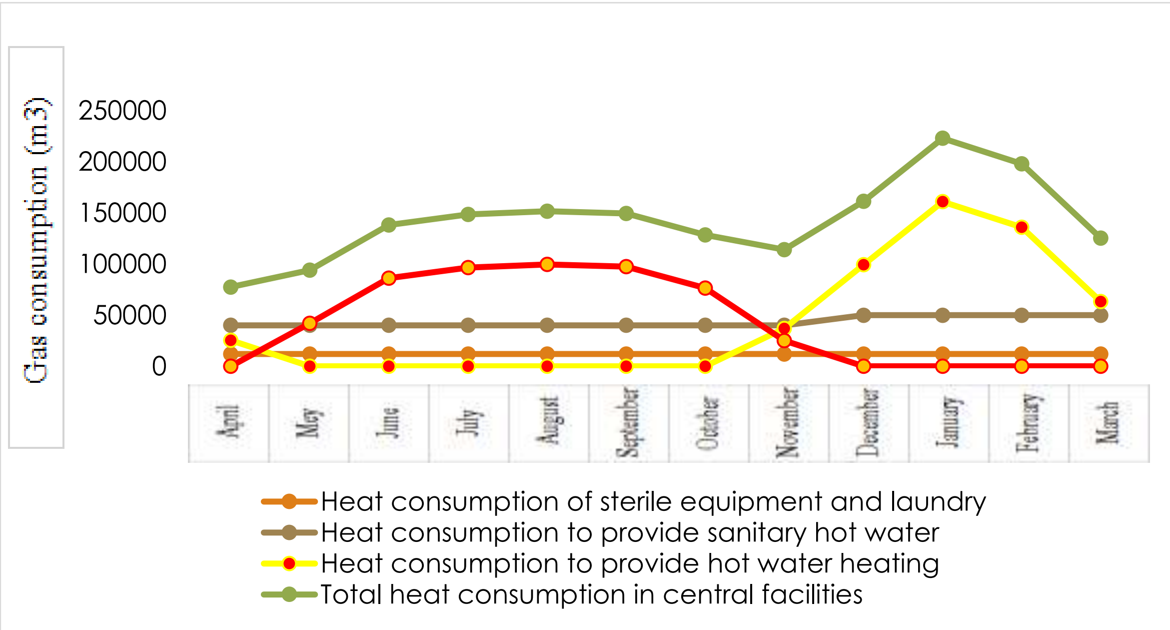

Fig. 12. Monthly changes in hospital heat consumption based on consumption terminals

In the above figure, the separation of consumption is based on natural gas consumption. By making appropriate assumptions, this diagram can be converted into heat and refrigeration load demand (Figure below).

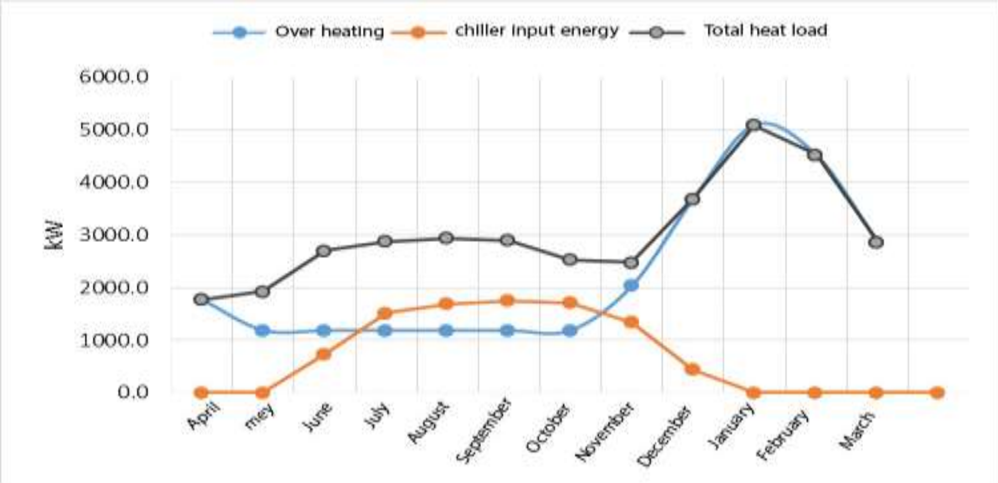

Fig.13. Average heat load and energy input to the chiller at different times of the year

100%	Fuel energy conversion efficiency	35000	kJ/m3	Fuel calorific value

1.5.2. Investigating energy demand supply scenarios

In order to apply the system studied in this study, three scenarios have been considered for this hospital. In the first scenario, all heat loads are supplied by the cogeneration system and the heat pump system is not used. In the second scenario, the geothermal heat pump system will replace the existing compression chiller and will be used only in cooling mode. In the third scenario, by changing the gas turbine to a smaller capacity, only the average heat demand will be met by the cogeneration system and other times by the heat pump system. In summer, the heat pump is used in cooling mode and in winter in heating mode.

2.1.1 Scenario 1: Providing heat load without using geothermal pump

In this scenario, the use of gas turbines is used as the basis method. In the case of full load gas turbine, the separation of production energies is as follows.

Table 8. Specifications of the gas turbine used in scenarios 1 and 2

Electric power	kW	1873
Fuel consumption	kg/s	0/174
Fuel energy consumption	kW	8092
Turbine exhaust gas energy	kW	5783
Recycled energy (steam generated)	kW	5105

In this case, the difference between the thermal and refrigeration load required by the system is less than the recyclable energy and therefore the heat loss of the cycle will be high. The table below shows how the generated thermal load changes at different gas turbine loads**.**

Table 9. Performance of gas turbines at different loads

Percentage of gas turbine load	%	100	80	75	70	65	60	55	50	45	40
Electric power	kW	1874	1490	1394	1298	1202	1105	1009	913	817	721
Fuel consumption	kg/s	174	151	146	140	134	129	123	117	112	106
Turbine output thermal power	kW	5783	5137	4978	4815	4651	4494	4332	4175	4014	3858
Recyclable heat	kW	5105	4407	4233	4058	3884	3712	3538	3366	3195	3026

Due to the supply of part of the refrigeration load using a compression chiller, in this project, due to the lack of use of this system and the presence of excess recycled heat in the cogeneration system, the system should be modified to provide all the refrigeration load using Supplied from the chiller. Under these conditions, the heat load demand increases compared to the past and the electric load demand decreases. In the table below, the operating conditions of the cogeneration system in this scenario are examined.

Table 10. How to supply thermal and refrigeration loads in scenario 1

Month	Total heat load demand	Percentage of gas turbine load	Gas turbine power	Energy recovery rate	Auxiliary boiler production energy	Fuel consumption
April	1946	40	721	3026	0	0.106
Mey	2930	40	721	3026	0	0.106
June	4632	90	817	4750	0	0.112
July	5030	100	817	5105	0	0.112
August	5150	100	913	5105	50	0.117
September	5067	100	817	5105	0	0.112
October	4264	80	817	4407	0	0.112
November	3203	50	817	3366	0	0.112
December	4052	75	1298	4233	0	0.14
January	1946	40	721	3026	450	0.106
February	2930	40	721	3026	0	0.106
March	4632	90	817	4750	0	0.112

2.1.2. Scenario 2: Using a geothermal heat pump system to replace the compression chiller system.

As mentioned, the refrigeration load of the hospital in summer is supplied by compression and absorption chillers. If the geothermal pump system is used, compression chillers can be taken out of operation. In this scenario, the remaining part of the heat demand will be met by a cogeneration system. Auxiliary boiler can also be used at peak heat demand loads. The table below describes how to meet the demand for heat and refrigeration.

Table 11. Supply of thermal and refrigeration loads in scenario 2

Month	Total heat load demand *	Demand for refrigeration	Percentage of gas turbine load	Electric power of geothermal heat pump	Fuel consumption
April	1946	0	40	0	0.106
Mey	2118	304	40	39	0.106
June	2969	623	45	81	0.112
July	3168	697	45	91	0.112
August	3228	720	50	94	0.117
September	3186	704	45	91	0.112
October	2785	554	45	72	0.112
November	2720	181	45	24	0.112
December	4052	0	70	0	0.14
January	5595	0	100	0	0.174
February	4967	0	100	0	0.174
March	3149	0	50	0	0.117

• In this case, 10% has been added to the total heat demand in order to waste the distribution system.

2.1.3. Scenario 3: Medium heat load supply with cogeneration system

In this case, in order to reduce the cost of gas turbine in the system is selected in such a way that heat recovery from it only responds to the average heat load and the rest of the heat demand is met by using a geothermal heat pump. For this purpose, a gas turbine with a rated power of 1080 made by Solar Company has been used. The efficiency of this turbine is 23% and its exhaust gas temperature is 500 ° C. Accordingly, the specifications of the power cycle section in full load mode will be as follows:

Table 12. Co-production system specifications used in scenario 3

Electric power	kW	1050
Fuel consumption	kg/s	0.099
Fuel energy consumption	kW	4604
Turbine exhaust gas energy	kW	3329
Recycled energy (steam generated)	kW	2900

Similar to the previous section, the geothermal heat pump system is responsible for providing cooling to half of the refrigeration needs from May to November. In these months, due to the reduction of the capacity of the heat recovery system, the simultaneous production of the cooling load of this system will be increased. In the cold months of the year, by changing the geothermal heat pump system, this system will directly provide part of the system's heating needs. Assuming uniform operation of the cogeneration system, the capacity of the geothermal heat pump system in different months of the year will be as follows.

Table 13. Supply of thermal and refrigeration loads in scenario 3

Month	Total heat load demand *	Gas turbine heat load	Refrigeration load of geothermal heat pump	Heat pump operation mode	Electric power of geothermal heat pump	Fuel consumption
April	1946	2900	0	3026	0	0/099
Mey	2118	2900	304	3026	39	0/099
June	2969	2900	651	4750	85	0/099
July	3168	2900	808	5105	105	0/099
August	3228	2900	855	5105	111	0/099
September	3186	2900	822	5105	107	0/099
October	2785	2900	554	4407	72	0/099
November	2720	2900	181	3366	24	0/099
December	4052	2900	1152	4233	136	0/099
January	5595	2900	2695	3026	317	0/099
February	4967	2900	2067	3026	243	0/099
March	3149	2900	249	4750	29	0/099

• In this case, 10% has been added to the total heat demand in order to waste the distribution system.

4. RESEARCH SUMMARY

In this research, the combination of two systems of simultaneous generation of cold heat and geothermal heat pump was investigated. This system has the advantages of the above two systems simultaneously and can be a good option to be used to meet energy demands. Based on the obtained results, scenario 3 has more appropriate energy parameters than other heat and refrigeration supply schemes of the hospital.

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