



## Performance and Exergy Analyses of GT-MHR Nuclear Cycles

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### Abstract:

In this study, the performance and exergy analyses of the GT-MHR type nuclear power plants with high thermal energy were analysed. The GT-MHR (Gas Turbine-Modular Helium Reactor), produces electricity by using helium gas in the Brayton cycle, and utilizes the waste heat of the system with the Rankine cycle. The Brayton cycle allows helium gas to be heated to high temperatures in the reactor core, then converted into mechanical energy by using a gas turbine. This hybrid structures in the designs of GT-MHRs allow the combined use of nuclear and renewable energy sources. It was found that exergy efficiency reached its maximum level between approximately 14,000 - 16,000 kPa. At this point, the exergy losses of this system approach its lowest level and the components reach the most ideal operating conditions. If the system is operated below or above of this value, the total exergy efficiency will decrease. The exergy efficiency, which is initially approximately 74.5% at 500 kPa pump pressure, increases to 75.1% at 1000 kPa pressure level. While the pump pressure at 500 kPa produces a work of approximately 532,000 kW, at 1000 kPa it increases to 560,000 kW. Thus, GT-MHR pioneers the energy technologies of the future by combining both high efficiency and environmental sustainability.

## 1. Introduction

In the 1970s, the steam generator and heat recovery steam generator were integrated and emerged. The waste heat generated at the end of the cycle in gas turbines is used to produce superheated steam and electrical energy through the heat recovery steam generator [1]. Natural gas is one of the most widely used sources in energy production today. It constitutes approximately 23.6% of electricity production in the world. Gas turbine combined cycle power plants are of great importance in the most efficient conversion of chemical energy generated by the combustion of natural gas into electrical energy. The working fluid of these systems generally consists of Brayton and Rankine cycles and provides approximately 63% electrical efficiency. Some efficiency improvement methods are also fundamentally used in combined cycle power plants. One of the most well-known of these is increasing the operating flow temperature at the gas turbine

inlet. However, high temperatures are required for the application of this method. Another method is to use heat sources with low potential in useful work [2, 3].

The modular helium reactor with gas turbine combines a single vessel and a high-efficiency Brayton cycle gas turbine energy conversion system in a single compartment. The reactor vessel and the energy conversion compartment are connected by a pipe. When we look at the design features, it has helium as a coolant, graphite as a moderator, and an envelope material that contains the fuel, also called a refractory material resistant to high temperatures. Since helium is not chemically active throughout the system, it always remains in the same phase. The coolant is heated by flowing down the coolant channels in the graphite fuel elements in the reactor core and is transmitted to the container where the gas turbine is located. As seen in Figure 1, the heated helium gas coming from the reactor core expands through the turbine and drives the generator and gas

compressor. In the second stage, the helium gas coming out of the turbine outlet is heated in the cold, high-pressure side of the heat recovery unit and sent back to the reactor [4].

In this case, the efficiency can be increased by adding the Rankine cycle to the plant, using the exhaust gas heat more thoroughly or reducing the cold source temperature. When this structure is considered, it also brings certain disadvantages. The fluid that carries the heat used may have a lower evaporation temperature compared to water, and it may also bring problems such as chemical stability and high toxicity. In these systems, the steam produced in the waste heat boiler is partially or completely directed to a steam turbine in order to increase electricity production, depending on the heat and power demands. This application is widely used in gas turbine systems. Combined cycles have an energy conversion rate of 40% and above of fuel energy. In thermodynamic terms, combined cycles consist of Brayton and Rankine cycles [5, 6, 7].

The heat produced here is used in the Brayton cycle, which is the main generator of work, and then in the Rankine cycle, which is the secondary generator of work. The combined cycle provides high-efficiency energy production by combining the high inlet temperature of the gas turbine with the low outlet temperature of the steam turbine. Thermal demands can be met by steam drawn directly from the waste heat boiler or steam turbine, depending on the thermodynamic properties of the desired steam [8]. Mahmoudi and Yari (2011), in their study, investigated the performance of GT-MHR (Gas turbine modular helium reactor) type reactor simple organic Rankine cycle, Rankine cycle with internal heat exchanger and regenerative Rankine cycles in terms of some parameters such as the first and second laws of thermodynamics, compressor pressure ratio, turbine inlet temperature, evaporator and ambient temperatures. They used EES software in the application of these combined cycles. As a result, it was seen that the GT-MHR simple organic Rankine cycle provided 5-10% improvement when the first and second law efficiencies were compared to the GT-MHR cycle itself. This showed that its performance increased both in terms of thermodynamics and economy. It was shown that the GT-MHR regenerative organic Rankine cycle can be used for power generation with heat [7].

Karaali and Keven (2023) mentioned in their study the integrated use of the organic Rankine cycle and the Brayton cycle, which allows us to obtain energy at low temperature levels. While this process was carried out, the compressor compression ratios, performance characteristics and changes according to the excess air coefficient were examined, and the data obtained showed that the total power and exergy

efficiency increased with compression expansion in gas turbine Rankine cycle combined cycle compressors, while the specific fuel consumption decreased. In addition, they stated that the excess air coefficient should be between 2.5 and 2.8 and mentioned its importance [9, 10].

Rashidi and Habibzadeh (2016) investigated the performance of organic Rankine cycle with 13 different fluids to recover waste heat using a modular helium reactor. These fluids were divided into 3 separate sections as dry, isentropic and wet fluids. As a result, they found that the increase in pump temperature reduces the total thermal efficiency and increases the total exergy loss. It was also concluded that R141b, R123 and R717 have the highest thermal efficiency values among isentropic, dry and wet fluids, respectively [11].

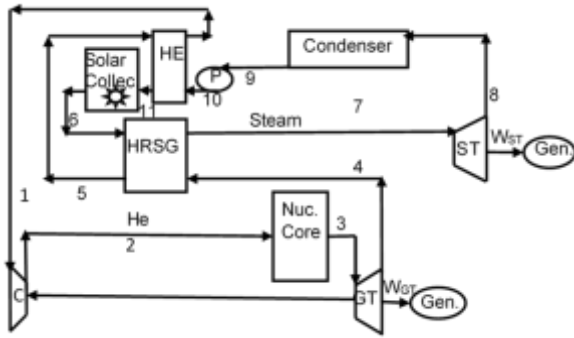
Wang et al. (2023) analysed the electrical efficiency, power, heat regenerator and thermal energy of a GT-MHR reactor with 250 MW thermal power according to a complex Brayton, a vertical gas turbine and a helium-cooled reactor. It was shown that the electrical efficiency had a great effect on the temperature coefficient of the helium intercooler. The analyses showed that the highest performance in the electricity production mode was 46.3%, while the decrease in the helium intercooler between the compressor stages directly affected the electrical efficiency [12].

Tao et al. (2023) conducted a study to evaluate the waste heat generated in the modified helium reactor (GT-MHR) using a gas-cooled closed Brayton cycle with an absorption cooler to cool the compressor inlet gas through the organic Rankine cycle. For the study, GT-MHR/ORC/ARC were combined and thermodynamic analyses were performed. It is shown that the combined system offers 12.4% higher efficiency and 9.7% lower LCOE (Unit electricity cost) than GT-MHR. In addition, the total levelized investment cost of the combined cycle is lower than the GT-MHR system. This is due to the cost reduction provided by the higher optimum pressure ratio in the compressor and the lower helium flow rate [13].

## 2. Material and Methods

As a result of the nuclear reaction that occurs in the reactor heart in sections 2-3 of the GT-MHR type nuclear power plant in Figure 1, a high amount of heat is released. This released heat is transferred to the helium fluid in the Brayton cycle in a controlled manner. This helium fluid is converted into work in the turbine in sections 3-4. The helium fluid, which loses some of its energy, transfers some of its remaining heat to the heat recovery steam generator in the cycle in sections 4-5. There are two separate

closed cycles here and only heat transfer occurs between the cycles without the fluids mixing. The helium fluid coming out of the flow line number 5 gives the other part of its heat to the heat exchanger in the Rankine cycle. A compressor is used to increase the pressure of the helium fluid, whose pressure has decreased between sections 1-2 compared to sections 2-3. The helium fluid, whose pressure has increased, is sent back to the reactor core [14].



**Figure 1.** GT-MHR schematic representation of Brayton and Rankine combined cycle.

The secondary cycle, the Rankine cycle, has a higher potential to produce energy at lower temperatures than the Brayton cycle. The energy it receives from the heat exchanger in the 10-11th branch and the fluid heated by the condenser solar energy also takes the energy of the water fluid entering the heat recovery steam generator in the 5th section and exits the 7th section as high-pressure water vapor. This water vapor that comes out is converted to work in the steam turbine in the 7th-8th range. After this stage, where more energy than desired can remain, it is reduced to the temperature levels required for the efficient operation of the cycle in the 8th and 9th stages, the condenser, and it passes from the water/vapor phase to the water phase again. As a result of these processes, the fluid, which has a very low pressure, is re-pressurized with the help of the pump in the 9th-10th range and enters the heat exchanger. In this way, higher efficiency is achieved in energy recovery by using the combined cycle [15, 16, 17].

Neutrons colliding with certain nuclei can cause the nucleus to split. This is called fission. The usable part of the heat energy formed in the nucleus as a result of fission or fusion is important. The usable energy amount per fission of a uranium atom is around 198-207 MeV. A nuclear power reactor is a structure that produces energy in its core and transfers this produced energy to the moving fluid. The heat energy coming from the reactor core only increases the temperature of the fluid [12].

Reactor is the part that contains the reactor core, turbine, recuperator, intercooler, compressor, precooler, reactor vessel, control rods, hot channel, transition vessel, and cooling systems. Helium is used as the fluid in the reactor seen in Figure 1. The mass, energy, exergy and entropy equations formed in the GT-MHR are given below.

**Table 1.** Mass, energy balance and entropy production equations [18, 19, 20]. Exergy, entropy and exergy efficiency equations

| Device                    | Mass Eq.  | Energy Eq.  | Entropy Eq.  |
|---------------------------|---|---|--|
| GT-MHR (Reactor)          | $\dot{m}_2 = \dot{m}_3$                               | $\dot{m}_2 h_2 + \dot{Q}_R = \dot{m}_3 h_3$                                 | $\dot{m}_2 s_2 + \frac{\dot{Q}_R}{T_R} = \dot{m}_3 s_3 + S_{Gen,R} = 0$                      |
| Gas Turbine               | $\dot{m}_3 = \dot{m}_4$                               | $\dot{m}_3 h_3 = \dot{W}_{GT} + \dot{m}_4 h_4$                              | $\dot{m}_3 s_3 - \dot{m}_4 s_4 + S_{Gen,GT} = 0$   |
| Pump                      | $\dot{m}_9 = \dot{m}_{10}$                            | $\dot{m}_9 h_{10} + \dot{W}_P = \dot{m}_{10} h_{10}$                        | $\dot{m}_9 s_9 - \dot{m}_{10} s_{10} + S_{Gen,P} = 0$  |
| Steam Turbine             | $\dot{m}_7 = \dot{m}_8$                               | $\dot{m}_7 h_7 = \dot{W}_{ST} + \dot{m}_8 h_8$                              | $\dot{m}_7 s_7 - \dot{m}_8 s_8 + S_{Gen,ST} = 0$   |
| Condenser                 | $\dot{m}_8 = \dot{m}_9$                               | $\dot{m}_8 h_8 = \dot{m}_9 h_9 + \dot{Q}_C$                                 | $\dot{m}_8 s_8 - \dot{m}_9 s_9 + \frac{\dot{Q}_C}{T_C} + S_{Gen,C} = 0$                      |
| HRSG                      | $\dot{m}_4 = \dot{m}_5 = \dot{m}_6 = \dot{m}_7$       | $\dot{m}_4 h_4 + \dot{m}_6 h_6 = \dot{m}_5 h_5 + \dot{m}_7 h_7$             | $\dot{m}_4 s_4 - \dot{m}_5 s_5 + \dot{m}_6 s_6 - \dot{m}_7 s_7 + S_{Gen,HRSG} = 0$           |
| Compressor                | $\dot{m}_1 = \dot{m}_2$                               | $\dot{m}_1 h_1 + \dot{W}_C = \dot{m}_2 h_2$                                 | $\dot{m}_1 s_1 - \dot{m}_2 s_2 + S_{Gen,C} = 0$  |
| Parabolic Solar Collector | $\dot{m}_{11} = \dot{m}_6$                            | $\dot{m}_{11} h_{11} + \dot{Q}_{Collector} = \dot{m}_6 h_6$                 | $\dot{m}_{11} s_{11} - \dot{m}_6 s_6 + S_{Gen,Collector} = 0$                                |
| Heat Exchanger            | $\dot{m}_{10} = \dot{m}_{11} = \dot{m}_1 = \dot{m}_5$ | $\dot{m}_{11} h_{11} - \dot{m}_{10} h_{10} = \dot{m}_1 h_1 - \dot{m}_5 h_5$ | $\dot{m}_{10} s_{10} - \dot{m}_{11} s_{11} + \dot{m}_5 s_5 - \dot{m}_1 s_1 + S_{Gen,HE} = 0$ |

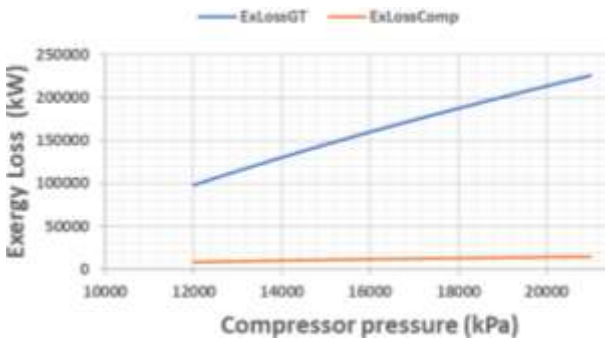
In Table 1 and Table 2, the mass balance, energy balance, entropy production, exergy balance and exergy efficiency formulas of the system elements used in the Rankine and Brayton combined cycle are given. The analyses are done by using these equations and the results obtained are compared with literature. By combining the solar energy to the system, high efficiencies and better performance are obtained. The solar energy used in the system improved the exergy efficiency of the system analysed in this study. Also, using waste heat can improve the efficiencies and the performance.

### 3. Results and Discussions

By using the equations in table 1 and 2 the results are figured as given here.

**Table 2.** Exergy, entropy and exergy efficiency equations [21, 22, 23]

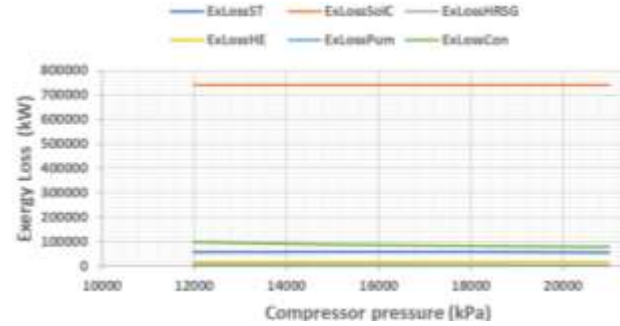
| Device                               | Exergy Eq.  | Exergy Efficiency Eq.  |
|--------------------------------------|---|--|
| GT-MHR (Reactor)                     | $\dot{E}x_2 + \dot{Q}_R \left(1 - \frac{T_0}{T_R}\right) = Ex_3 + E_{Loss,R}$                 | $\eta_{ex,R} = \frac{E_{ex,3} - E_{ex,2}}{\dot{W}_R}$  |
| Gas Turbine                          | $\dot{E}x_3 = \dot{W}_{GT} + \dot{E}x_4 + \dot{E}_{Loss,GT}$                                  | $\eta_{ex,GT} = \frac{E_{ex,3} - E_{ex,4}}{\dot{W}_{GT}}$  |
| Pump                                 | $\dot{E}x_9 + \dot{W}_P = \dot{E}x_{10} + \dot{E}_{Loss,P}$                                   | $\eta_{ex,P} = \frac{E_{ex,10} - E_{ex,9}}{\dot{W}_P}$   |
| Steam Turbine                        | $\dot{E}x_7 = \dot{W}_{ST} + \dot{E}x_8 + \dot{E}_{Loss,ST}$                                  | $\eta_{ex,ST} = \frac{\dot{W}_{ST}}{E_{ex,7} - E_{ex,8}}$  |
| Condenser                            | $\dot{E}x_8 = E_{x_9} + E_{Loss,C}$   | $\eta_{ex,C} = \frac{E_{ex,9} - E_{ex,8}}{\dot{W}_C}$  |
| HRSG (Heat Recovery Steam Generator) | $\dot{E}x_4 = \dot{E}x_5 + \dot{E}x_7 - \dot{E}x_6 + \dot{E}_{Loss,HRSG}$                     | $\eta_{ex,HRSG} = \frac{E_{ex,7} - E_{ex,6}}{E_{ex,5} - E_{ex,4}}$                                   |
| Compressor                           | $\dot{E}x_1 + \dot{W}_C = \dot{E}x_2 + E_{Loss,C}$  | $\eta_{ex,C} = \frac{E_{ex,2} - E_{ex,1}}{\dot{W}_C}$  |
| Parabolic Solar Collector            | $\dot{E}x_{11} + \dot{Q}_{Coll} \left(1 - \frac{T_0}{T_{coll}}\right) = Ex_6 + E_{Loss,Coll}$ | $\eta_{ex,Coll} = \frac{E_{ex,11} - E_{ex,6}}{\dot{Q}_{Coll} \left(1 - \frac{T_0}{T_{coll}}\right)}$ |
| Heat Exchanger                       | $\dot{E}x_{10} = E_{x_{11}} + E_{Loss,HE}$  | $\eta_{ex,HE} = \frac{E_{ex,11} - E_{ex,10}}{E_{ex,10}}$   |



**Figure 2.** Effect of increase in compressor pressure on exergy loss of the gas turbine and compressor of the Brayton cycle.

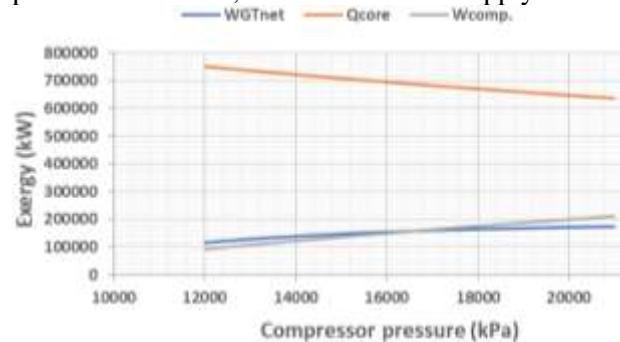
The most striking feature of Figure 2 is that the exergy loss in the gas turbine increases significantly as the compressor pressure increases. In particular, it is seen that the exergy loss at the 20,000 kPa pressure level is 120% higher than at 12,000 kPa. This situation shows that the increase in compressor pressure leads to greater exergy losses in the gas turbine. The main reason for this is the increase in

turbine inlet temperatures with the increase in pressure and the increase in irreversibility accordingly. If it is planned to increase the compressor pressure in order to increase the cycle efficiency, it is evaluated that the exergy losses occurring in this process should be analysed carefully.



**Figure 3.** Effect of increase in compressor pressure on exergy loss of some devices of the Rankine cycle.

Figure 3 shows how the exergy losses in the Rankine cycle work with the compressor pressure. The graph represents the excess pressure compressor pressure, which is divided between 10,000 kPa and 22,000 kPa. The vertical axis represents the exergy loss and reflects the energy losses that occur in different formations in the system. It is seen that the shape changes and the largest exergy loss is seen in the parabolic solar collector. The exergy loss in these solar components is quite high, remaining constant at around 750,000. Since the exergy loss in the collector is much larger compared to other payments, it is understood that this content should be focused on in order to increase the system efficiencies. An increase in compressor pressure leads to a significant decrease in exergy losses in other components. For example, exergy losses in components such as the steam turbine, pump, and condenser show a slight decrease as compressor pressure increases. This suggests that some energy losses can be reduced by increasing compressor pressure. However, this trend does not apply to the

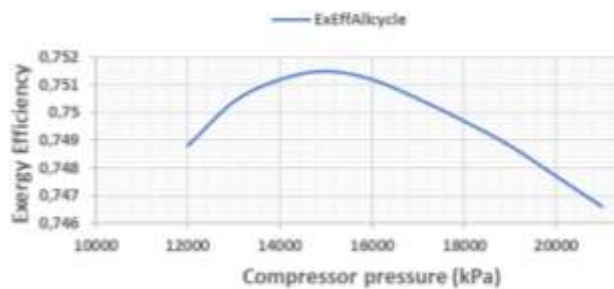


**Figure 4.** Effect of compressor pressure on gas turbine, reactor and compressor powers.

collector; the exergy loss in this component remains almost constant. Exergy loss in the gas-steam generator (GGBJ) is at lower levels compared to other components. Similarly, exergy loss in the heat exchanger is also significant, but not as high as losses in components such as the compressor or turbine. When Figure 4 is examined, it is observed that all three parameters examined remain almost constant with the increase in pump pressure. In particular,  $Q_{GTMHR}$  (Reactor) is not directly affected by changes in pump pressure since it represents the thermal input of the system. This shows that the amount of heat provided by the reactor remains constant regardless of the pump pressure.

Similarly, the compressor work and the net work of the gas turbine do not show a significant change as the pump pressure changes. The main reason for this is that the role of the pump pressure in the Brayton cycle is quite limited. The main energy input in the Brayton cycle is the heat provided by the GT-MHR reactor and the pump pressure does not have a direct determining effect on this process.

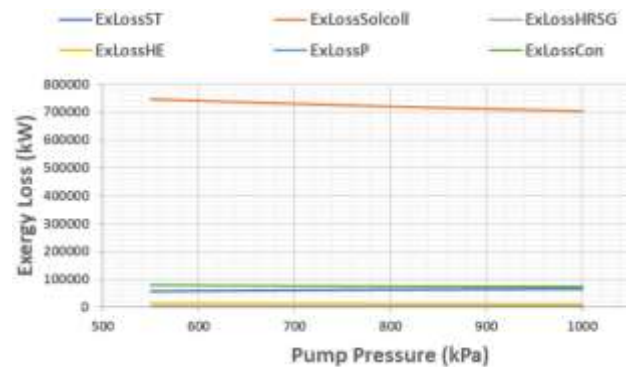
In particular, the small difference between the compressor work and the work of the gas turbine shows that the work consumed by the compressor takes a value very close to the work produced by the turbine. This indicates that the work transformations in the cycle occur in a balanced manner. However, increasing or decreasing the pump pressure does not directly affect this work transformation.



**Figure 5.** Effect of compressor pressure on exergy efficiency of the entire system.

When Figure 5 is examined, it is seen that the exergy efficiency of the system increases with the increase in compressor pressure at the beginning. It has been determined that the exergy efficiency reaches the maximum level in the range of approximately 14,000 - 16,000 kPa. At this point, the exergy losses of the system approach the lowest level and the components reach the most ideal operating conditions. However, when this pressure range is exceeded, further increase in compressor pressure causes the exergy efficiency of the system to decrease. At high pressure levels, irreversibility increases in the turbine and other components, leading to increased exergy losses. As seen in the

previous graphs, the exergy efficiency of the gas turbine in particular decreases at high pressures and exergy losses increase significantly. This causes the overall exergy efficiency of the system to decrease. This analysis reveals that there is an ideal compressor pressure range for the system. It is seen that the system reaches its highest exergy efficiency in the pressure range of approximately 14,000 - 16,000 kPa. If the system is operated below or above this range, the total exergy efficiency will decrease and more irreversibility will occur. Therefore, the compressor pressure must be carefully determined to ensure optimum operating conditions.



**Figure 6.** Effect of pump pressure on exergy losses in some devices of the system.

First of all, it is observed that the heat rejected in the condenser tends to decrease slightly as the pump pressure increases as seen in Figure 6. The main reason for this is that higher pump pressures provide a more efficient energy conversion in the Rankine cycle and therefore less waste heat is generated.

This is a positive sign that the system has become more thermodynamically efficient and exergy losses have decreased. High pump pressure allows the steam to enter the turbine at higher pressure and temperature, helping it to produce more work. As a result, the amount of waste heat is reduced and the overall energy conversion performance of the system is improved as seen in Figure 7.

However, excessive pump pressure increases irreversibility and can cause negative effects on cycle efficiency after a certain point. Therefore, determining an optimum pump pressure range is a critical factor in both minimizing the heat rejected in the condenser and increasing the overall exergy efficiency of the system. Similarly, the heat input provided by the parabolic collector also shows a slight decrease in strength. The main reason for this is that at higher pressure levels, the more efficient use of the Rankine cycle requires less heat input from the collector. When the thermodynamic efficiency in the system increases with the increase in pump pressure, the need for additional heat input decreases. On the other hand, the work consumed by

the pump shows a slight increase in pump pressure. This is expected, because the higher the pump pressure, the more work must be done to raise the fluids to higher pressure levels. However, as can be seen in the graph, the pump system remains quite low compared to other energy inputs. This shows that the energy consumption of the pump does not have a significant effect on the overall system balance.

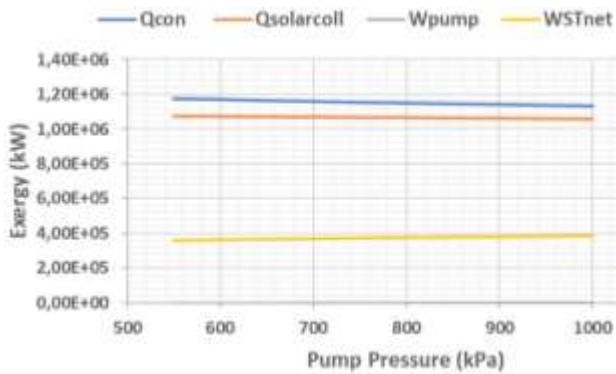


Figure 7. Effect of pump pressure on power of different devices in the entire cycle.

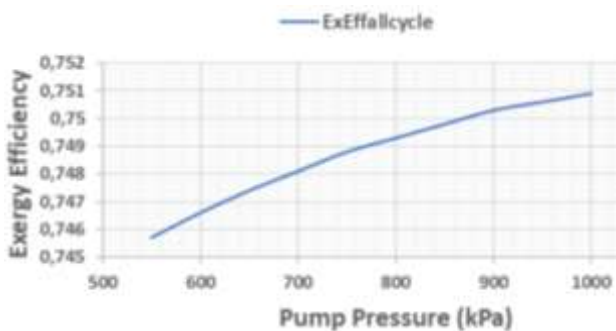


Figure 8. Effect of pump pressure on exergy efficiency of the entire system.

Finally, the net work produced by the steam turbine tends to increase slightly as the pressure increases. The main reason for this is that the exergy value of the steam entering the turbine increases at higher pressure levels, allowing the turbine to produce more work. However, this increase does not show a large change, so the system appears to exhibit a certain stability against pump pressure changes. This analysis shows that pump pressure positively affects system performance up to a certain point, but extremely high-pressure levels provide limited benefits. In terms of system optimization, careful determination of pump pressure is a critical factor in both balancing energy inputs and increasing net work production. When Figure 8 is examined, it is seen that the exergy efficiency of the system increases as the pump pressure increases from 500 kPa to 1000 kPa. The exergy efficiency, which is approximately 0.745 (74.5%) at the beginning, increases to 75.1% at the 1000 kPa pressure level. This situation shows that higher pump pressures

improve the overall exergy performance of the system. The main reason for this trend is that higher pump pressure supports higher pressure steam production in the Rankine cycle and as a result, the steam entering the turbine has a higher exergy content. Since high-pressure steam has the potential to produce more work in the turbine, the exergy efficiency of the cycle is positively affected. These findings show that determining the ideal pump pressure is an important factor in increasing system efficiency on the Rankine cycle side. However, it should also be taken into account that very high pump pressures can increase irreversibility losses and limit exergy gains after a certain point. Therefore, in system design, the pump pressure should be carefully optimized to increase the exergy potential of the steam and keep irreversibilities to a minimum.

Higher pump pressure can reduce thermal losses and enable the system to operate more efficiently. However, an important point to note is that the increase in exergy efficiency is not linear. While the increase in pump pressure initially increases exergy efficiency rapidly, it is observed that the increase in efficiency gradually slows down as higher pressure levels are reached. This situation shows that the positive effect of very high pump pressures on exergy efficiency may decrease after a certain point. Although the initially increased pump pressure improves the performance of the system by increasing the exergy potential of the steam, irreversibility increases after a certain level, limiting this improvement.

This graph shows that there is an ideal pump pressure range for the entire system and that efficiency gains will be limited beyond this range. If the pump pressure is increased to very high levels, the amount of work consumed by the pump can increase and this can have a compensating negative effect on the overall performance of the system. Therefore, determining the ideal pump pressure in system design is a critical factor in both increasing exergy efficiency and keeping pump work consumption under control. In order to maximize efficiency gains, the pump pressure must be carefully selected and the system must be operated in the most efficient range.

Pump and compressor pressure are two important parameters in hybrid energy systems and directly affect the overall performance of the system. In this system where Rankine and Brayton cycles work together, the effects of pressure changes occur in various ways due to the different dynamics of both cycles. Increasing the pump pressure positively affects the total net work production and exergy efficiency of the system. In the Rankine cycle, higher pump pressure increases the exergy content

of the steam entering the turbine, increasing the amount of work produced by the turbine, while reducing the heat lost in the condenser and collector. This contributes to the system becoming more efficient. However, since the work consumed by the pump will also increase at very high pump pressures, the efficiency increase slows down after a certain point. On the Brayton cycle side, it was observed that pump pressure changes did not have a significant effect on the gas turbine and compressor work balances.

The net work produced by the gas turbine, the work consumed by the compressor and the heat input provided by the GT-MHR remain almost constant regardless of the pump pressure. This shows that changing the pump pressure is not an effective method to increase the performance in the Brayton cycle. On the other hand, increasing the compressor pressure has an effect that reduces the net work production of the system. This is because the compressor consumes more work with the increase of the compressor pressure and the work produced by the turbine is insufficient to compensate for these losses. Therefore, the compressor pressure must be carefully optimized for the efficient operation of the system. In terms of the Rankine cycle, although the compressor pressure is not a direct determining factor, it can cause a decrease in the exergy efficiencies of the turbine and pump. However, no significant effect was observed on components such as the heat exchanger and the Gas-Steam Generator. It has been determined that the exergy efficiency reaches its maximum level in the range of approximately 14,000 - 16,000 kPa. At this point, the exergy losses of the system approach the lowest level and the components reach the most ideal operating conditions. If the system is operated below or above this range, the total exergy efficiency will decrease and more irreversibility will occur. Therefore, the compressor pressure must be carefully determined to ensure optimum operating conditions. The net work production, which is around 570,000 kW at the compressor pressure level of 12,000 kPa, decreases to approximately 535,000 kW at the level of 22,000 kPa. This trend shows that increasing the compressor pressure has a negative effect on the total work production of the system. The exergy efficiency, which is approximately

**Table 3.** System energy balance and exergy losses in optimal conditions where compressor pressure is 15,000 kPa and pump pressure is 1000 kPa.

| DEVICE           |  |
|------------------|--|
| GT-MHR (Reactor) | $GTMHR_{In} = En_2 + Q_{GTMHR}$<br>$GTMHR_{Out} = En_3$<br>$GTMHR_{In} = 708.444 + 156.889$<br>$= 865333 \approx En_3$ |

|                                      |  |
|--------------------------------------|--|
| Gas Turbine                          | $GasTur_{In} = En_3$ $GasTur_{Out} = En_4 + W_{Comp} + W_{GTnet}$<br>$865343 \approx 581.915 + 136.820 + 146.608$<br>$Ex_{Loss} = 145.542 \text{ kW}, Ex_{Eff} = 0,8643$<br>$W_{GT, Iz} = 171.254 \text{ kW}, W_{GT} = 146.608 \text{ kW}$   |
| HRSG (Heat Recovery Steam Generator) | $HRSG_{In} = En_4 + En_6$ $HRSG_{Out} = En_5 + En_7$<br>$581.915 + 1,245E+09 \approx 186.207 + 1,641E+06$<br>$Ex_{Eff} = 0,9946$ $Ex_{Loss} = 6242 \text{ kW}$   |
| Compressor                           | $Comp_{In} = En_1 + W_{Comp, is}$ $Comp_{Out} = m_{He}(h_{2s} - h_{0_{He}})$<br>$20.079 + 120402 \approx 200 (2256 - 1554) \approx 140.400$<br>$Ex_{Loss} = 11.331 \text{ kW}$ $Ex_{Eff} = 0,983$<br>$W_{Comp, is} = 120.402 \text{ kW}$ $W_{Comp} = 136.820 \text{ kW}$           |
| Steam Turbine                        | $ST_{In} = En_7$ $ST_{Out} = En_4 + W_{Pump, is} + W_{STnet} + En_8$<br>$1,641E+06 \approx 581915 + 463,9 + 437.933$<br>$1,202E+06$<br>$Ex_{Loss} = 69.550 \text{ kW}, Ex_{Eff} = 0,8817$  |
| Pump                                 | $Pump_{In} = En_9 + W_{Pump}$ $Pump_{Out} = En_{10}$<br>$19437 + 545,8 \approx 19982$<br>$Ex_{Eff} = 0,9037$ $Ex_{Loss} = 79,2 \text{ kW}$   |
| Heat Exchanger                       | $HE_{In} = En_5 + En_{10}$ $HE_{Out} = En_1 + En_{11}$<br>$Ex_{Eff} = 0,9751$<br>$186.207 + 19.982 \approx 20.078 + 186.110$   |
| Parabolic Solar Collector            | $Coll_{In} = Q_{Coll} + En_{11}$ $Coll_{Out} = En_6$<br>$1,059E+06 + 186.110 \approx 1,245E + 06$<br>$Ex_{Loss} = 702.690 \text{ kW}$  |
| Condenser                            | $Con_{In} = En_8$ $Con_{Out} = En_9 + Q_{Con}$<br>$1,202E+06 \approx 19.437 + 1,183E+06$<br>$Ex_{Loss} = 80.152 \text{ kW}$  |
| All Cycle                            | $All\ Cycle_{In} = Q_{Coll} + Q_{GT-MHR}$<br>$All\ Cycle_{Out} = Q_{Cond} + W_{GTnet} + W_{STnet}$<br>$1,059E+06 + 708.444$ $1,183E+06 + 437.933 + 146.608$<br>$W_{Net, total} = 584541 \text{ kW}$ $Ex_{Eff, All\ Cycle} = 0,755$<br>$Ex_{Loss, All\ Cycle} = 185.548 \text{ kW}$ |

0.745 (74.5%) at the pump pressure of 500 kPa at the beginning, increases to 75.1% at the pressure level of 1000 kPa. At 500 kPa pump pressure the net work is produced, which is approximately 532,000 kW, while at 1000 kPa the net work is up to 560,000 kW.

#### 4. Conclusions

The findings show that both pump and compressor pressures should be carefully optimized to increase the efficiency of the system. While increasing the pump pressure improves the efficiency of the Rankine cycle, turbine and pump exergy efficiencies decrease after a certain point. This reveals that excessively high pump pressures can limit cycle efficiency by increasing irreversibility. On the Brayton cycle side, the effect of pump pressure is not significant, while increasing compressor pressure directly reduces net work production. This situation

reveals that compressor pressure should be carefully determined for the Brayton cycle to operate efficiently. In this context, the interactions between the Rankine and Brayton cycles in hybrid systems should be taken into account and the effect of pressure changes on the entire system should be evaluated in a holistic manner. Instead of optimizing a single cycle, the operating conditions of both cycles should be considered together and the most appropriate design parameters should be determined. The optimum net work production is obtained at the compressor pressure level of 12 MPa, and decreases at the level of 22 MPa. This trend shows that increasing the compressor pressure has a negative effect on the total work production of the system. The exergy efficiency, which is approximately 0.745 (74.5%) at the pump pressure of 500 kPa at the beginning and increases to 75.1% at the pressure level of 1000 kPa.

### Author Statements:

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