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Performance and Exergy Analysis of Cryogenic Cycles

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Abstract:

This study examines and demonstrates the concepts of thermodynamics, exergy, and exergy analysis, and shows how these principles are applied to cryogenic cycles using equations. Furthermore, an energy, exergy, and performance analyses of the two-stage vapor compression cascade cryogenic cycle for natural gas liquefaction is conducted, and optimum operating conditions are presented in tables. Based on the data obtained from these equations, exergy analysis of cryogenic cycles was performed, and performance analysis and curves were plotted.When the compressor outlet pressures of the first sub cycle is increased from 3300 kPa to 4590 kPa, the exergy loss of the devices decreases, the exergy efficiencies of the increases, and the COP values increase. When the cooling amount in heat exchangers 1 and 2 is increased from 9 °C to 17 °C, i.e., when cooling is increased, it leads to a slight improvement in the performance of the system. The optimum total compressor consumption according to inlet/outlet temperature difference in heat exchangers 1 and 2 are obtained at about 14-15 °C. When these two improvement methods are applied together, optimum performance values of the system are obtained. The COP value increases from 0.078 to 0.1164. The system is affected by the compressor outlet pressure of the first sub cycle. Increasing the cooling level from 9°C to 17°C in heat exchangers 1 and 2 has a minor impact on system performance.

1. Introduction

Today, advancements in technology and industrial applications are increasing the importance of cryogenic systems capable of reaching extremely low temperatures. The need for cryogenic cooling systems is particularly critical in areas such as biomedical applications, superconductivity, and the transportation and storage of liquefied gases. Analyses aimed at improving the performance of these systems, ensuring energy efficiency, and reducing losses are of great importance from an engineering perspective. The scientific problem addressed in this study is the systematic analysis of the energy and exergy-based performance evaluations of cryogenic cycles. Cryogenic cycles can be implemented in many different structures and configurations, and energy conversions and losses differ in each cycle [1, 2]. Therefore, for efficient operation of the systems, it is necessary to examine each component based on energy and exergy and to determine at which stage the greatest loss occurs. The main objective of this research is to comparatively analyse different cryogenic cycle structures and to reveal which systems operate more efficiently under which conditions. In this context, two-stage vapor compression cascade cycles, were analysed. System performance was determined by calculating energy and exergy losses for each system component (compressor, condenser, evaporator, expansion valve) [3, 4]. The conceptual framework of the research is based on classical thermodynamics and exergy analysis. Exergy analysis is a very powerful tool for determining the actual losses and inefficiencies in a system [5, 6]. In this study, system performances were evaluated using parameters such as exergy efficiency and total exergy destruction.

Cryogenic systems play a critical role in many advanced engineering fields such as the storage and transport of liquefied gases, superconductivity, biomedical applications, and space technologies. The efficiency of the cycles used in these systems is of great importance in terms of both reducing energy costs and mitigating environmental impacts. Therefore, a detailed examination of the energy and exergy-based performance of cryogenic cycles is a crucial requirement from both an academic and industrial perspective. The rationale for this study is that most cryogenic cycle analyses in the literature are limited to specific systems, and comparative performance evaluations of different cycle structures are insufficient. Furthermore, most existing analyses are limited to energy balance and do not sufficiently focus on revealing the actual losses in the system through exergy analysis [7, 8]. This study aims to fill this gap in the literature by enabling the comparison of different cryogenic cycles based on energy and exergy analysis. In this way, it is aimed to contribute to engineering decisions for more efficient system design in cryogenic applications. The results of the research will provide important information, especially in terms of energy saving, system optimization, and sustainable engineering applications.

Cryogenics is the science that studies the formation of extremely low temperatures (below 100 K) and their effects on the liquefaction of stable gases such as oxygen, hydrogen, helium, nitrogen, and argon. Temperatures are lowered to approximately below -150 °C in technical processes such as the liquefaction of natural gas and hydrogen; biomedical applications such as maintaining the health of cells and tissues and freezing diseased tissues; and scientific research such as superconductivity research and the study of the properties of materials at low temperatures. Because gases like helium, hydrogen, oxygen, argon, and nitrogen have very low critical points, they exist only in gaseous form in the environment.

Cryogenics is the branch of science that deals with temperatures below 100 K (approximately -173.15 °C) and studies systems and materials that operate at these temperatures [11, 12]. Cryogenic cycles are special refrigeration cycles, that usually multi-stage, that work to liquefy gases or achieve low temperatures. Exergy is the maximum work a system can do until it reaches equilibrium with its environment. It expresses the quality and potential use of energy [13, 14]. Exergy efficiency is an efficiency indicator that expresses the portion of the system that can be used usefully in relation to the total exergy it possesses. Coefficient of Performance (COP) is the ratio of the cooling effect obtained in a refrigeration cycle to the energy given to the system. Cascade cycle is a multi-stage system in which two or more refrigeration cycles are integrated to achieve lower temperatures [15, 16].

Cooling is the transfer of heat from a low-temperature environment to a high-temperature environment. The equipment designed to perform this process is called refrigeration machines, and their operating cycle is called the refrigeration cycle. The most commonly used refrigeration system is the vapor compression refrigeration system. The fluids that carry heat and operate in refrigeration cycles are called refrigerants. The refrigerant evaporates, condenses, and is compressed again in the vapor phase during the cycle. The refrigerant compressed in the compressor first passes into the superheated vapor phase and is sent to the condenser. The refrigerant condenses and releases heat to the environment. The pressure of the refrigerant exiting the condenser is reduced in the expansion valve, and it enters the evaporator as a liquid-vapor mixture. Since the temperature of the refrigerant is lower than that of the environment, it evaporates and absorbs heat from the environment, thus cooling it. The refrigerant exiting the evaporator as saturated vapor is drawn in by the compressor, and the cycle ends. The refrigeration cycle consists of four main elements: compressor, condenser, accelerator, and evaporator [17, 18].

Cryogenic applications generally in liquefaction and gas separation, gas storage-transport, medical - biological applications, superconductivity, changing the properties of materials or fluids at very low temperature are used.

In all the above cases, cooling is the method that provides the desired use and performance. It is also the main way to approach absolute zero. Only liquids below the triple point below 100 K are considered cold. This means that they are still in liquid or gaseous state below this temperature [19, 20].

The effect of the thermodynamic properties of the gases used in the cryogenic process is important. Below are the characteristic temperatures (°C) of some cryogenic fluids. Hydrogen is not used commonly as a cryogenic refrigerant because of the instability of liquid hydrogen, while oxygen is almost not used because of its harmful effects. Although nitrogen is an inert gas, its high cost makes its use difficult. Consequently, the main refrigerants are helium and nitrogen. However, since the current nitrogen market is still limited to the triple point, the most marketed cryogenic refrigerant is mostly helium [18].

The most suitable operating conditions were obtained by applying exergy analysis and local optimization methods in two-stage vapor compression cascade cryogenic cycle. The 1. and 2. laws of thermodynamics, exergy analyses, and local optimization method were used to two-phase cascade cryogenic cycles. Results for optimum operating condition about the 2-stage vapor compression cryogenic cycles were obtained [18, 21, 22].

2. Material and Methods

Entropy is a thermodynamic property that expresses the degree of disorder or energy dissipation in a system. 1. law of thermodynamics (conservation of energy) states that the difference between the energy input and energy output of a system is equal to the energy stored in the system. 2. law of thermodynamics (Law of entropy) states that in energy transformations, some of the energy can't be converted into work and that entropy tends to increase in the system [15, 17, 18, 19].The cycle examined in the figure 1, occurs in two stages. At the 2. stage, the refrigerant enters the compressor as the gas of the substance being studied. In compressor, the refrigerant is compressed to condenser pressure. The fluid passes through the condenser tubes, condenses, and begins to cool by releasing heat to the environment. After the condenser, the liquid enters the capillary tube, and its pressure and temperature decrease under the influence of the gas valve. The low-temperature fluid enter into the evaporator, that evaporation occurs and heat is removed from the cooled environment. This cycle ends when refrigerant exits the evaporator and returns to the compressor.At the 1. partial cycle, the refrigerant enters in to the compressor as the gas of the substance being studied. The high-temperature refrigerant exits here, passes through the condenser tubes, condenses, and begins to cool by releasing heat to the environment. After the condenser, the high-temperature refrigerant enters the capillary tube, and its pressure and temperature decrease importantly under the influence of the gas valve. In the 2. part, after condensation, the refrigerant enters the evaporator and evaporates with the heat of the cooled environment. When the refrigerant exits the evaporator, this completes its cycle as a liquid natural gas mixture [15, 17, 18, 19].In the thermodynamic analysis of the cycles, the values of entropy, temperature, enthalpy and other thermodynamic properties were taken from web site https://webbook.nist.gov/chemistry/fluid/3 [9]. In table 3, formulas used in this study for exergy loss and exergy efficiency for the two-stage vapor compression cascade cryogenic cycle devices are given. In table 2, Formulas used in this study for mass balance, energy balance and entropy generations in the two-phase vapor compression multistage cryogenic cycles are given.

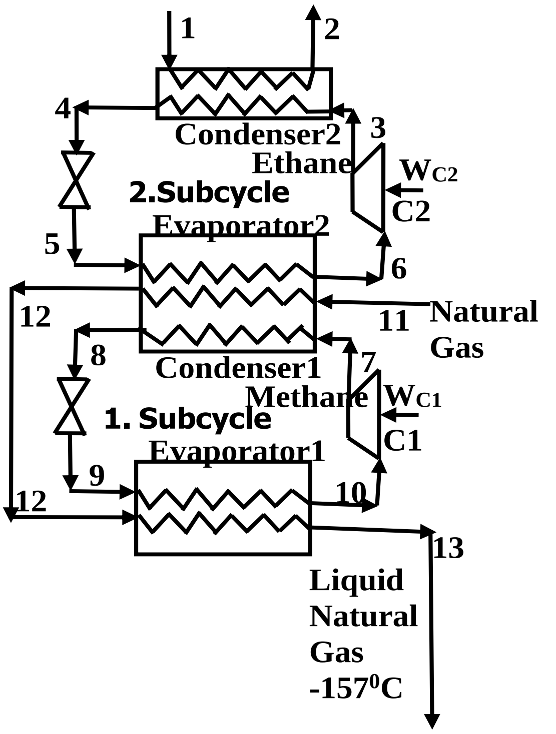

Figure 1. *Two-Stage Vapor Compression Cascade Cryogenic Cycle Diagram*

Table 2: *Formulas used in this study for mass balance, energy balance and entropy generations in two-phase vapor compression multistage cryogenic cycles [15, 17, 18, 19].*

Device	Mass Balance	Energy Balance	Entropy *Generation*
Compressor1	$\dot{m}_7 = \dot{m}_{10}$	$\dot{m}_{10}h_{10} + \dot{W}_{C1} = \dot{m}_7 h_7$	$\dot{m}_{10}s_{10} - \dot{m}_7 s_7 + \dot{S}_{gen,C1} = 0$
Condanser1	$\dot{m}_7 = \dot{m}_8$	$\dot{m}_7 h_7 = \dot{Q}_{1Cn} + \dot{m}_8 h_8$	$\dot{m}_7 s_7 - \dot{m}_8 s_8 + \dot{S}_{gen,Cn1} = 0$
Throttle Valve1	$\dot{m}_8 = \dot{m}_9$	$\dot{m}_8 h_8 = \dot{m}_9 h_9$	$\dot{m}_8 s_8 - \dot{m}_9 s_9 + \dot{S}_{gen,TV1} = 0$
Evaporator1	$\dot{m}_9 = \dot{m}_{10}$	$\dot{m}_9 h_9 = \dot{m}_{10}h_{10} + \dot{Q}_{1E}$	$\dot{m}_9 s_9 - \dot{m}_{10}s_{10} + \dot{S}_{gen,E1} = 0$
Compressor2	$\dot{m}_3 = \dot{m}_6$	$\dot{m}_6 h_6 + \dot{W}_{C2} = \dot{m}_3 h_3$	$\dot{m}_6 s_6 - \dot{m}_3 s_3 + \dot{S}_{gen,C2} = 0$
Condenser2	$\dot{m}_3 = \dot{m}_4$	$\dot{m}_3 h_3 = \dot{Q}_{2Cn} + \dot{m}_4 h_4$	$\dot{m}_3 s_3 - \dot{m}_4 s_4 + \dot{S}_{gen,Cn2} = 0$
Throttle Valve2	$\dot{m}_4 = \dot{m}_5$	$\dot{m}_4 h_4 = \dot{m}_5 h_5$	$\dot{m}_4 s_4 - \dot{m}_5 s_5 + \dot{S}_{gen,KV2} = 0$
Evaporator2	$\dot{m}_5 = \dot{m}_6$	$\dot{m}_5 h_5 = \dot{m}_6 h_6 + \dot{Q}_{2E}$	$\dot{m}_5 s_5 - \dot{m}_6 s_6 + \dot{S}_{gen,E2} = 0$

Table 3: Formulas used in this study for exergy loss and exergy efficiency for two-stage vapor compression cascade cryogenic cycle devices [15, 17, 18, 19].

Device	Exergy Loss	Exergy Efficiency
Compressor	$E_{c,loss} = E_i - E_o = W_c + E_{x1} - E_{x2}$ $E_{c,loss} = mT_0(s_{2} - s_1)$ $E_{c,loss} = m(h_2 - h_1 - T_0(s_{2} - s_1))$	$\eta_{C,ex} = \frac{W_{rov}}{W_{in}} = 1 - \frac{E_{xloss}}{W_{in}}$ $\eta_{C,is} = \frac{W_{is}}{W_g} = \frac{\mathrm{m}(h_{2s} - h_1)}{\mathrm{m}(h_2 - h_1)}$
Evaporator	Taking from ambient $E_{ex,loss} = m[(h_1 - h_2 - T_0(s_1 - s_2)] - [-Q_L\left(1 - \frac{T_0}{T_L}\right)]$	$\eta_{ex,evap} = \frac{E_{XQL}}{E_{x1} - E_{x2}} = \frac{-Q_L\left(1 - \frac{T_0}{T_L}\right)}{m[(h_1 - h_2 - T_0(s_1 - s_2))]}$
Condenser	$E_{ex,loss} = [\dot{m}(h_1 - h_2 - T_0(s_1 - s_2)] - [-Q_H\left(1 - \frac{T_0}{T_H}\right)]$ $E_{ex,loss} = T_0(m_{Co}(s_2 - s_1) - m_w(s_3 - s_4))$	$\eta_{ex,Con} = \frac{Q_H\left(1 - \frac{T_0}{T_H}\right)}{m[(h_1 - h_2 - T_0(s_1 - s_2))]}$ $\eta_{ex,Con} = \frac{E_{XQH}}{E_{x1} - E_{x2}} = 1 - \frac{E_{xy\imath k}}{E_{x1} - E_{x2}}$
Throttle Valve	$E_{ex,y\imath k} = T_0 S_{\ddot{u}ret} = \mathrm{m}T_0(s_2 - s_1)$ $E_{ex,y\imath k} = E_g - E_{ç}$	$\eta_{ex,kv} = 1 - \frac{E_{xy\imath k}}{E_{x1} - E_{x2}} = 1 - \frac{E_{x1} - E_{x2}}{E_{x1} - E_{x2}}$

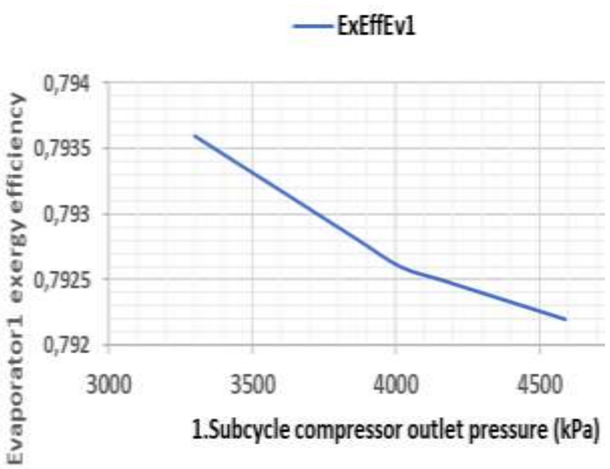

***Figure 2.** Effect on evaporator1 exergy efficiency with respect to compressor outlet pressure of subcycle1.*

3. Results and Discussions

Figure 2, shows the effect on exergy loss in the condensers with respect to the compressor outlet pressure of subcycle1. When the compressor outlet pressure of subcycle1 is increased from 3300 kPa to 4590 kPa, the exergy loss in heat exchangers1 and 2 remains almost constant about 0.79, up to 4300 kPa.

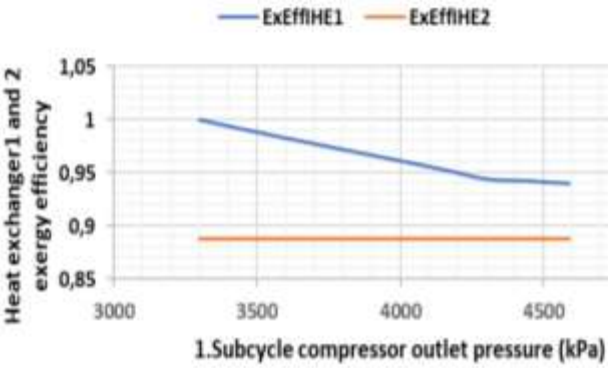

***Figure 3.** Effect on exergy efficiency in heat exchangers 1 and 2 with respect to compressor outlet pressure of subcycle1.*

Figure 3, shows the tffect on exergy efficiency in heat exchangers 1 and 2 with respect to the compressor outlet pressure of subcycle1. When the compressor outlet pressure of subcycle1 is increased from 3300 kPa to 4590 kPa, the exergy efficiency in heat exchanger 2 remains constant, but the exergy efficiency in heat exchanger 1 decreases by approximately 5%.

Figure 4, shows the effect on exergy loss in heat exchangers 1 and 2 according to the inlet and outlet temperature difference. When cooling is removed from 9°C to 17°C in heat exchangers 1 and 2, the exergy loss in heat exchangers 1 and 2 increases by more than 100%.

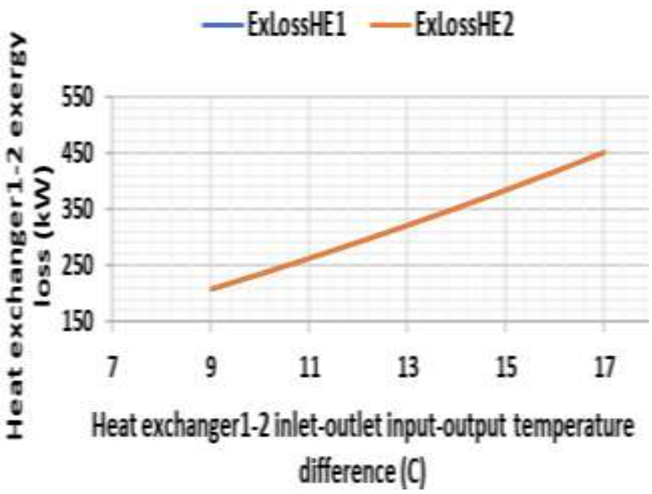

***Figure 4.** Effect on exergy loss in heat exchangers 1 and 2 according to inlet/outlet temperature difference.*

Figure 5, shows the change in exergy loss in evaporators 1 and 2 according to the inlet and outlet temperature difference in heat exchangers 1 and 2. When cooling is increased from 9°C to 17°C in heat exchangers 1 and 2, there is only a negligible change in exergy loss in evaporator 1 and condenser 2. Figure 6 shows the effect on exergy loss in heat exchangers 1 and 2 according to the inlet/outlet temperature difference. When cooling is removed from 9°C to 17°C in heat exchangers 1 and 2, there is only a very slight decrease in exergy loss in heat exchangers 1 and 2.

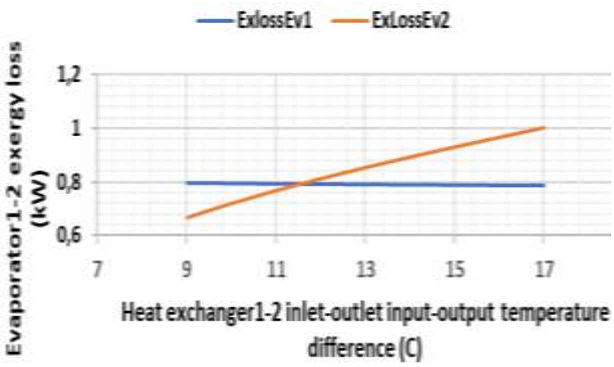

Figure 5. *Effect on exergy loss in evaporators 1 and 2 according to inlet/outlet temperature difference in heat exchangers 1 and 2.*

Figure 7 shows the change in exergy efficiency in compressors 1 and 2 according to the inlet and outlet temperature difference in heat exchangers 1 and 2. When the cooling in heat exchangers 1 and 2 is increased from 9°C to 17°C, there is only a negligible change in exergy efficiency in compressors 1 and 2.
Figure 8 shows the effect on total compressor consumption according to the inlet/outlet temperature difference in heat exchangers 1 and 2. When cooling is increased from 9°C to 17°C in heat exchangers 1 and 2, the total compressor consumption decreases by approximately 0.2%. Figure 8 also shows that the optimum total compressor consumption according to inlet/outlet temperature difference in heat exchangers 1 and 2 are obtained at about 14-15 °C.

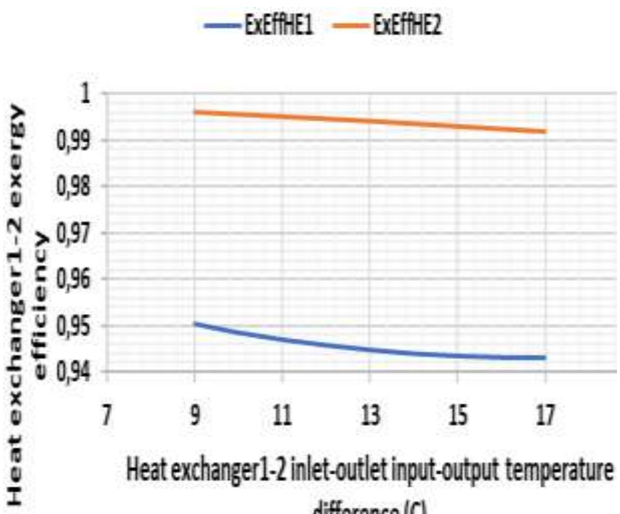

Figure 6. *Effect on exergy efficiency in heat exchangers 1 and 2 according to inlet/outlet temperature difference*

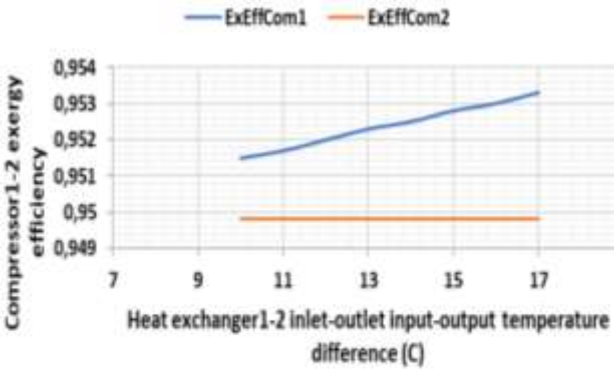

Figure 7. *Effect on exergy efficiency in compressors 1 and 2 according to inlet/outlet temperature difference in heat exchangers 1 and 2.*

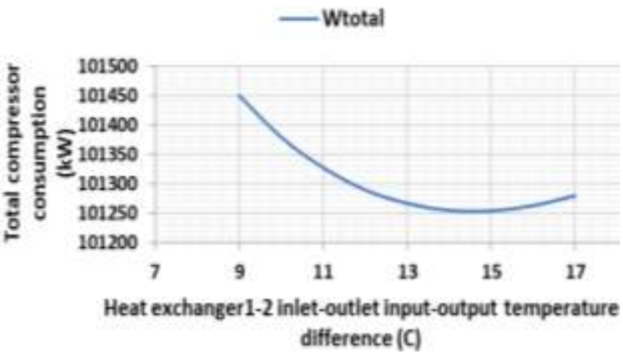

Figure 8. *Effect on total compressor consumption according to inlet/outlet temperature difference in heat exchangers 1 and 2.*

Table 4 provides a comprehensive representation of the optimum operating conditions of the system's working fluid temperatures, mass, entropies, energy and exergy flows based on compressor pressure and heat exchanger temperature difference. Pressure, mass, temperature, enthalpy, energy, entropy and exergy, are given for the optimum operating conditions.
Table 5 provides a comprehensive representation of the optimum operating conditions of the system's devices energy, energy balance, exergy, exergy losses, exergy efficiencies, COP value, and other properties based on compressor pressure and heat exchanger temperature difference. The given values are for the optimum operating conditions.
At the optimum operating conditions COP obtained as 0,1164 and the compressor consumption was at the minimum values. Also, mass flow for ethane and methane were found. At the optimum operating conditions minimum exergy loss were obtained.

4. Conclusions

In conclusion, this research revealed the necessary information and equations for exergy analysis in cryogenic cycles. Based on the data obtained from

Table 4: *A comprehensive representation of the optimum operating conditions of the system's working fluid temperatures, mass, entropies, energy and exergy flows based on compressor pressure and heat exchanger temperature difference.*

m_{NG}=2 kg/s	m_{eta}=23,09 kg/s	m_{met}=4,52 kg/s
T_0=0 ^{0}C	h_{0eta}=-44,64kJ/kg	s_{14}=-1,69 kJ/kgK
T_{0K}=273,2 K	h_{0met}=-56,1kJ/kg	s_{15}=-6,54 kJ/kgK
T_1=84,86 ^{0}C	h_1=110,8 kJ/kg	En_1=3589 kW
T_2=306,6 ^{0}C	h_3=-93,96 kJ/kg	En_2=15735 kW
T_3=30 ^{0}C	h_4=-375,7 kJ/kg	En_3=-1139 kW
T_4=13 ^{0}C	h_5=-375,7 kJ/kg	En_4=-7645 kW
T_5=-92,85 ^{0}C	h_6=-171 kJ/kg	En_5=-7645 kW
T_6=-82,85 ^{0}C	h_7=-562,4 kJ/kg	En_6=-2918 kW
T_7=-84 ^{0}C	h_8=-271,5 kJ/kg	En_7=-2288 kW
T_8=-101,6 ^{0}C	h_9=385,3 kJ/kg	En_8=-973,4 kW
T_9=188,5 ^{0}C	h_{9s}=332,8 kJ/kg	En_9=1995 kW
T_{10}=-101 ^{0}C	h_{10}=-674 kJ/kg	En_{10}=-2793 kW
T_{11}=-164,3 ^{0}C	h_{11}=-674 kJ/kg	En_{11}=-2793 kW
T_{12}=-154,3 ^{0}C	h_{12}=-383,1 kJ/kg	En_{12}=-1478 kW
T_{13}=20 ^{0}C	h_{13}=-15,13 kJ/kg	En_{13}=81,95 kW
T_{14}=-82 ^{0}C	h_{14}=-237,3 kJ/kg	En_{14}=-362,4 kW
T_{15}=-157 ^{0}C	h_{15}=-894,8 kJ/kg	En_{15}=-1677 kW
Tr=17 ^{0}C	s_{0eta}=-0,1503 kJ/kgK	Ex_1=89,65 kW
P_0=100 kPa	s_{0met}=-0,1887 kJ/kgK	Ex_2=11772 kW
P_1=80 kPa	s_1=0,4045 kJ/kgK	Ex_3=5654 kW
P_2=3800 kPa	s_2=0,4779 kJ/kgK	Ex_4=5197 kW
P_3=3800 kPa	s_{2s}=0,4045 kJ/kgK	Ex_5=2581 kW
P_4=3800 kPa	s_3=-1,227 kJ/kgK	Ex6=159,2 kW
P_5=80 kPa	s_4=-2,186 kJ/kgK	Ex_7=2993 kW
P_6=80 kPa	s_5=-1,772 kJ/kgK	Ex_8=99,88 kW
P_7=4590 kPa	s_6=-0,6381 kJ/kgK	Ex_9=2925 kW
P_8=80 kPa	s_7=-4,467 kJ/kgK	Ex_{10}=3246 kW
P_9=4590 kPa	s_8=-1,058 kJ/kgK	Ex_{11}=2529 kW
P_{10}=4590 kPa	s_9=-0,9419 kJ/kgK	Ex_{12}=555,5 kW
P_{11}=80 kPa	s_{9s}=-1,058 kJ/kgK	Ex_{13}=393,6 kW
P_{12}=80 kPa	s_{10}=-5,081 kJ/kgK	Ex_{14}=457,5 kW
P_{13}=400 kPa	s_{11}=-4,5 kJ/kgK	Ex_{15}=1793 kW
P_{14}=400 kPa	s_{12}=-1,836 kJ/kgK	
P_{15}=400 kPa	s_{13}=-0,7591 kJ/kgK	

Table 5: *A comprehensive representation of the optimum operating conditions of the system's devices energy, energy balance, exergy, exergy losses, exergy efficiencies, COP value, and other properties based on compressor pressure and heat exchanger temperature difference.*

Q_{1Ev}=1315 kW	$ExEff_{Co1Ev2}$=0,6119	COP=0,1164
Q_{2Ev}=4728 kW	$Scyc_{1out}$=4284 kW	$ExLoss_{Ev1}$=637,8kW
Q_{2met}=4284 kW	$Scyc_{1in}$=4284 kW	$ExLoss_{HE1}$=202,3kW
Q_{Co2}=16874 kW	$Scyc_{2out}$=16874 kW	$ExLoss_{HE2}$=526,2kW
W_{C1}=2969 kW	$Scyc_{2in}$=16874 kW	$ExLoss_{C1}$=143,4 kW
W_{C2}=12146 kW	$AllScyc_{out}$=16874kW	$ExLoss_{C2}$=463,5 kW
W_{Cs1}=2731 kW	$AllScyc_{in}$=16874 kW	$ExLoss_{TV1}$=717,3kW
W_{Ctot}=15114	$ExEff_{HE2}$=0,91	$ExLoss_{TV2}$=2616 kW
$ExEff_{Ev1}$=0,786	$ExEff_{C1}$=0,953	$ExLoss_{Co1Ev2}$=2289kW
$ExEff_{HE1}$=0,943	$ExEff_{C2}$=0,962	$ExLoss_{Co2}$=6118 kW

these equations, exergy analysis of cryogenic cycles was performed, and performance analysis and curves were plotted.

When the compressor outlet pressure of the first sub cycle is increased from 3300 kPa to 4590 kPa, the exergy loss of the system decreases, the exergy efficiency of the devices increases, and the COP values increase. Increasing the cooling amount from 9°C to 17°C in heat exchangers 1 and 2 leads to a slight improvement in the system's performance. The optimum total compressor consumption according to inlet/outlet temperature difference in heat exchangers 1 and 2 are obtained at about 14-15 °C.

When these two improvement methods are applied together, optimum performance values of the system are obtained. The COP value increases from 0.078 to 0.1164. The system is affected by the compressor outlet pressure of the first sub cycle. Increasing the cooling amount from 9°C to 17°C in heat exchangers 1 and 2 has a small effect on the system performance.

Author Statements:

- **Ethical approval:** The conducted research is not related to either human or animal use.
- **Conflict of interest:** The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper

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- **Use of AI Tools:** The author(s) declare that no generative AI or AI-assisted technologies were used in the writing process of this manuscript.

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