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ENERGY RECOVERY DURING LIQUEFIED NATURAL GAS REGASIFICATION

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ABSTRACT

In the liquefied natural gas (LNG) regasification process, the cold exergy can be recovered as electric power. For this purpose, in the present study, a simple open Brayton cycle was considered as a base cycle and its efficiency was improved by adding different equipment step by step. This procedure resulted in the investigation of four different cycles. The comparison of the results led to the proposal of a combined cycle that could recover the LNG cold energy most efficiently. This cycle consists of three parts; an LNG directly expanding cycle, a middle-pressure close Brayton cycle with nitrogen as working fluid, and finally an open gas turbine power cycle. All the cycles were analyzed based on the first and second laws of thermodynamics. Finally, the effects of different parameters upon the thermal and exergetic efficiencies were examined.

INTRODUCTION

The global LNG (liquefied natural gas) trade has been increased rapidly during recent years. A large amount of energy is consumed to produce low-temperature (about –160°C) LNG, which has plenty of cryogenic exergy/energy. Therefore, the effective utilization of the cryogenic energy associated with LNG vaporization is quite important. The use of fossil fuels as energy resources results in serious air pollution, also the emission of CO_2 gas as a combustion exhaust is known to play a major role in global greenhouse effect. Therefore, the use of energy resources with less or no influence on air pollution and environment is required. Liquefied natural gas (LNG) is known as a clean energy source which is commonly used as domestic and industrial fuel for combustion. LNG is composed of 85–99% methane by mole fraction and a few percent ethane and propane depending on its production site. Since moisture and sulphur are contained in crude natural gases, they should be removed during the liquefying preprocess.

NOMENCLATURE

m	[kg/s]	mass flow rate
h	[kJ/kg]	enthalpy
LHV	[kJ/kg]	lower heating value
T	[K]	Temperature
e	[kJ/kg]	exergy
E	[kW]	exergy
LA	[-]	LNG mass flow rate to air mass flow rate
NA	[-]	nitrogen mass flow rate to air mass flow rate
R	[-]	Pressure ratio
P	[Mpa]	Pressure
s	[kJ/kg-K]	Entropy
I	[kW]	irreversibility
W	[kW]	Work

Greek symbols

η	[%]	efficiency

Subscripts

CMB	Combustion
inlet	Inlet of device
outlet	outlet of device
c	Cold stream
h	hot stream
loss	loss
0	Ambient
fuel	fuel
in	input
out	output
th	thermal
ex	exergy
s	stack

Natural gas (NG) is often found in remote regions far from consumer. For the transportation of natural gas from producing wells to utilization sites, two approaches are commonly applied [1].

1-pipeline transport, which suits the areas where gas well and consumer are located in the same mainland or gas well sites are near offshore.

2-shipping transport, when oceans separate the gas source and the user. In shipping way, NG shall be first converted to liquefied natural gas and then shipped to the consumer country using insulated LNG tanks, because the volume of LNG could be reduced about 600 times than that of natural gas, at atmospheric pressure and at a temperature of about -162°C. At receiving terminals, LNG is off-loaded into storage tanks and then pumped from storage tanks to the required pressure and vaporized for final transmission to the consumers. In a real application, LNG should be vaporized in order to be supplied as natural gas. During the vaporization process, latent heat of vaporization and any sensible heat required to superheat the vapor should be supplied to the LNG. During the liquefying process, a large amount of mechanical energy is consumed in refrigeration process, so LNG contains much cold energy (cryogenic exergy).

From thermodynamic viewpoint, reheating represents a net loss of available energy, which causes degradation of overall energy efficiency of the conversion chain. Accordingly, utilization of LNG cold energy proves to be an interesting area of study. The potential of cold energy utilization includes power generation, refrigeration, air liquefaction and separation, reduction of CO_2 emission, cryogenic thermoelectric generator, and similar applications to cold usage. LNG cold energy could be utilized in two major approaches: cooling power supply and electric power generation. For cooling power supply, LNG is commonly used in freezing foods, making dry ice, air conditioning, low-temperature crushing, etc [1]. However, its performance is usually not good enough because the users only need relatively high temperatures (about –50°C) [2].

For electric power generation, there are two kinds of LNG utilization: (1) independent thermal cycle with natural gas direct expansion (natural gas as working fluid) and closed-loop Rankine cycle (organic working fluid). For natural gas direct expansion type, LNG is firstly pumped to higher pressure, then vaporized to superheating natural gas, and finally expands in a turbo-expender to a certain pressure for supplying to users. For closed-loop Rankine cycle type, the surrounding is a heat source and LNG is a cold sink. However, both of them have relatively low efficiencies. (2) Improvement of the performance of thermal cycles. The thermal cycles can be improved by LNG utilization in several ways such as pre-cooling the intake air of GT compressor, providing the required low pressure of condensers of Rankine cycles, etc [3].

One of the methods used to improve cycle performance is utilization of a CHP binary cycle composed of two Brayton cycles [4].

A gas cascading cycle for logical usage from LNG cool energy was suggested by Lu and Wang [1]. They analyzed the influences of key parameters on thermal efficiency and exergy efficiency of the cascading power cycle. Kaneko et al. [5] proposed a new kind of combined cycle consisting of a conventional gas turbine worked as a topping cycle and TG (inverted Brayton cycle) as a bottoming cycle used to recover the cold energy consumed in liquefaction of natural gas. The feasibility of using inlet air cooling by virtue of the cold energy of LNG to increase power output of gas/steam combined cycle power plants during warm seasons was studied by Kim and Ro [6]. Dispenza et al. [4] used LNG as the heat source in a gas turbine combined cycle with carbon dioxide, hydrogen and helium as working fluid. They showed that the specific net work delivered by an ideal simple Brayton cycle using helium as working fluid is much more than that of cycles using other working fluids [4] and compared thermal efficiency and exergy efficiency of helium and nitrogen cycles [7]. A novel combined power cycle utilizing low-temperature waste heat and LNG cold energy was proposed by Shi and Che [8]. The proposed combined system consisted of a Rankine cycle with ammonia–water mixture as the working fluid to recover low-temperature waste heat.

In the present study, a simple Brayton cycle is considered as the base case and its thermal efficiency and exergy efficiency are improved by adding different equipment which results in four different cycles. Analyzing all these cycles, the best one is proposed to use the cold energy released during LNG regasification.

DESCRIPTION OF THE POWER CYCLE WITH LNG COLD ENERGY RECOVERY

As discussed above, we consider an open Brayton cycle as the base cycle and add different equipment to enhance its performance. The common assumptions of all four cycles are presented in Table1. These values are considered in different analyses unless specified otherwise.

Table 1.The general assumptions for four cycles

Items	Unit	Value
compressor inlet temperature	°C	5
Isentropic efficiency of compressor	100%	0.8
Isentropic efficiency of turbine	100%	0.8
LHV of natural gas	kJ/kg	49192
Combustion efficiency	100%	0.99
Mass flow rate of fuel	kg/s	2.792
Mass flow rate of air	kg/s	145.737
Pinch Point	°C	30
Methane mass fraction	100%	0.8534
Propane mass fraction	100%	0.0676
Ethane mass fraction	100%	0.079
Ambient temperature	°C	30
Ambient pressure	kPa	100

Open Single Brayton Cycle

Figure1 illustrates an open single Brayton cycle. At this cycle the inlet air is at ambient temperature and pressure while the inlet temperature and pressure of fuel are 110°C and 2400kPa, respectively. To have better criteria for comparison, we refer to the efficiency of this cycle as the base efficiency.

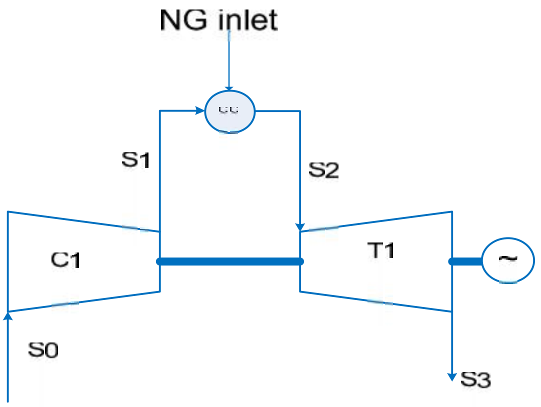

Fig1. Flow sheet of the Open Single Brayton Cycle

Open Brayton Cycle with LNG Cold Energy Recovery

The construction of LNG terminals and the need to vaporize LNG offers a thermal sink at a very much lower temperature than the usual thermal sinks like seawater. By using LNG as thermal sink in a combined plant that produces both power and gas, it is possible to recover power from the vaporization of LNG.

A combined cycle consisting of a Brayton power cycle of combustion gas (S0-HX1-S1-C1-S2-C.C-S4-T1-S5-S6), and an open LNG cycle (L0-P1-L1-HX1-L2-HX2-L3-T3-L4-L5), is proposed in this stage, Fig.2. In the open LNG cycle, LNG with low-temperature of -162°C at atmospheric pressure enters LNG pump, P1, to be pressurized and next passes heat exchanger HX1, in which it cools the inlet air of compressor, C1. After that, LNG is gasified and superheated by absorbing heat from heat exchanger HX2. Finally, the superheated natural gas enters turbine T3 to be expanded to the required conditions. The major part of the outlet NG is taken to supply to consumers from line L5 while the required fuel for the combustion chamber is taken from line L6. The extra assumptions related to this case are given in Table2.

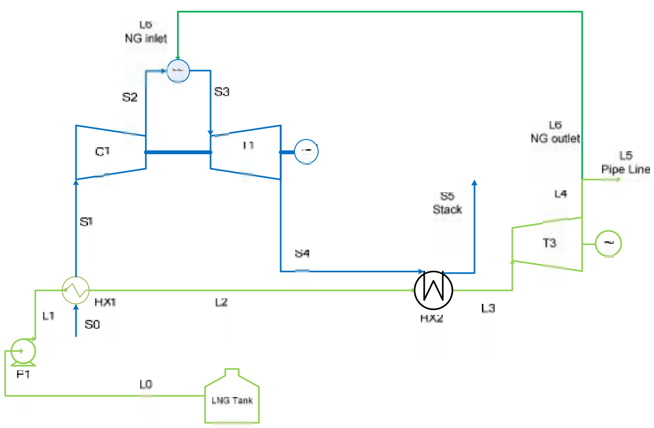

Fig.2. Flow sheet of open Brayton cycle with LNG cold energy recovery

Table 2.The Extra assumptions related to Open Brayton cycle with LNG cold energy recovery		
Items	Unit	Value
Pressure ratio of compressor C1	100%	15
Mass flow rate of LNG	kg/s	100.262
Pump inlet temperature	°C	-162
Pump inlet pressure	kPa	100
Pump outlet pressure	kPa	6000
Stack pressure	kPa	100
Stack temperature	°C	30

Open Brayton Cycle with LNG Cold Energy Recovery and Middle Cycle

In the cycle proposed in Fig.2, the temperature difference in HX2 is high which leads to high irreversibility proposing a good potential to improve the performance of the cycle. Therefore, in this stage we add a middle Brayton cycle which uses of nitrogen as working fluid, Fig.3. Actually, invoking this cycle, the lost work of previous section is recovered, now. In the middle Brayton cycle, nitrogen absorbs heat from the outlet gases of turbine T1 through HX3 and rejects heat to the LNG via HX2. The extra assumptions related to Fig 3 are shown in Table3.

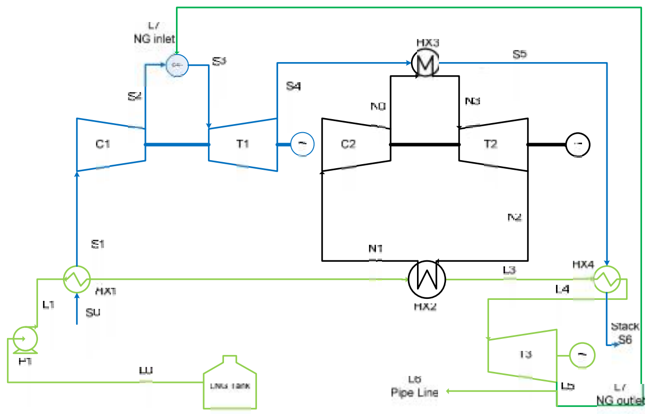

Fig.3. Flow sheet of open Brayton cycle with LNG cold energy recovery and middle cycle

Table 3.The Extra assumptions related to Open Brayton cycle with LNG cold energy recovery and middle cycle		
Items	Unit	Value
Pressure ratio of compressor C2	100%	15
Mass flow rate of nitrogen	kg/s	50.187
Pinch Point of HX3	°C	30
inlet pressure of compressor C2	kPa	500

Final Combined Cycle

The final amendment is performed by adding a regenerator between open Brayton cycle turbine outlet and middle cycle compressor outlet. This change decreases irreversibility and fuel consumption while increases nitrogen turbine inlet temperature, natural gas turbine inlet temperature and as a result, increases first and second laws efficiencies.

The resulted cycle consists of a middle cycle using nitrogen as working fluid, power cycle of combustion gas, and open LNG cycle. Figure4 shows this cycle which consists of three cycles: open LNG cycle (L0-P1-L1-HX1-L2-HX2-L3-HX4-L4-T3-L5-L6), middle cycle (R0-T2-R1-R-R2-HX2-R3-C2-R4-HX3) and open Brayton cycle (S0-HX1-S1-C1-S2-R-S3-C.C-S4-T1-S5-HX3-S6-HX4-S7). In the open LNG cycle, LNG with low-temperature of -162°C at atmospheric pressure enters LNG pump P1 to be pressurized, and next goes to heat exchanger HX1 to cool the compressor inlet air, after that by absorbing heat from outlet gas of turbine T2 to be gasified, and then comes into heat exchanger HX4 to be superheated by exhaust gas of Brayton cycle, and finally it enters turbine T3 to expand. Hence, natural gas is produced with required pressure according to variable usages of natural gas.

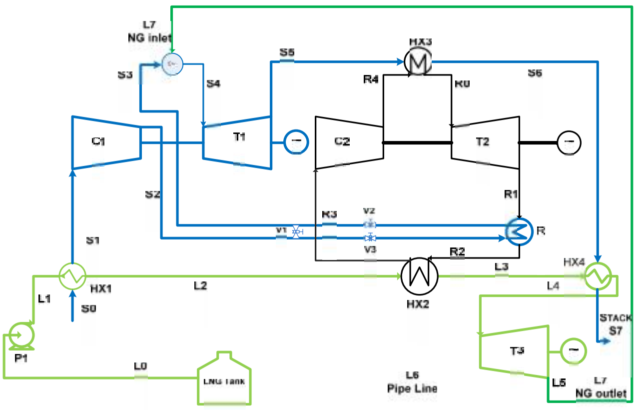

Fig.4. Flow sheet of the final combined power cycle

In the top Brayton cycle, inlet air, S0, is passed through HX1 to be cooled prior to be compressed by compressor C1. For preheating, air comes into regenerator R and then goes with NG of line L7 to combustion chamber. The high temperature combustion products enter turbine T1 via line S4 and after expansion they pass heat exchangers EX3 and EX4 to recuperate their energy.

In the middle Brayton cycle, nitrogen absorbs heat from the outlet gases of turbine T1 through HX3 and rejects heat to the LNG via HX2. The Pinch point of the regenerative R is assumed as 10°C.

In order to analyze the performance of the systems, the influence of several key parameters on thermal efficiency, exergy efficiency and net power delivered by the cycles was evaluated by simulation the proposed cycles in the Engineering Equation Solver Commercial Version 6.883 [9].

MATHEMATIC MODEL FOR PERFORMANCE ANALYSIS

Energy and exergy equilibrium equations of each unit of the cycles are established with neglect of pressure drop in fired heater, heat exchangers, and pipelines.

Energy Balance Equations, Thermal Efficiency and Base Thermal Efficiency

The energy equation for pumps and compressors is

$$W_i = -m_{i,inlet}\left(h_{i,inlet} - h_{i,outlet}\right)\eta_i = -m_{i,inlet}\left(h_{i,inlet} - h'_{i,outlet}\right) \quad (1)$$

while, for turbines, the energy equation is written as

$$W_j = m_{j,inlet}\left(h_{j,inlet} - h_{j,outlet}\right) = \eta_j m_{j,inlet}\left(h_{j,inlet} - h'_{j,outlet}\right) \quad (2)$$

For heat exchangers, the energy equation is

$$m_c\left(h_{c,inlet} - h_{c,outlet}\right) = m_h\left(h_{h,inlet} - h_{h,outlet}\right) \quad (3)$$

Also, for the fired heater C.C., the energy equation is

$$\eta_{CMB} m_{fuel} LHV + m_{s_{in}} h_{s_{in}} = m_{s_{out}} h_{s_{out}} \quad (4)$$

The total thermal efficiency of power cycles is written as

$$\eta_{th} = \frac{\left(\sum W_j - \sum W_i\right)}{\eta_{CMB} m_{fuel} LHV} \quad (5)$$

The base thermal efficiency of power cycles is defined as

$$\eta_{th,Base} = \frac{W_{T_1} - W_{C_1}}{\eta_{CMB} m_{fuel} LHV} \quad (6)$$

while, the base thermal efficiency of final combined cycle is

$$\eta_{th,Base} = \frac{W_{T_1} - W_{C_1}}{\eta_{CMB} m_{l7} LHV + m_{s0}\left(h_{N2} - h_{N3}\right)} \quad (7)$$

Exergy Balance Equation, Exergy Efficiency and Base Exergy Efficiency

The specific flow exergy is defined as

$$e = \left(h - h_0\right) - T_0\left(s - s_0\right) \quad (8)$$

The output exergy Ej for turbines is

$$E_j = m_{j,inlet}\left(e_{j,inlet} - e_{j,outlet}\right) \quad (9)$$

while, the output exergy Ej for pumps and compressors is

$$E_i = -m_{i,inlet}\left(e_{i,inlet} - e_{i,outlet}\right) \quad (10)$$

Overall exergy balance equation is written as

$$E_{in} = E_{out} + E_{loss} \quad (11)$$

where the overall input exergy of the system, E_{in}, is

$$E_{in} = m_{l_6} LHV + m_{l_0} e_{l_0} \quad (12)$$

The output exergy of the system E_{out} is

$$E_{out} = \sum E_j - \sum E_i \quad (13)$$

Also, total exergy efficiency is defined as

$$\eta_{ex} = \frac{E_{out}}{E_{in}} \quad (14)$$

Base thermal efficiency is

$$\eta_{ex,Base} = \frac{E_{T_1} - E_{C_1}}{E_{in} - m_{l_0} e_{l_0}} \quad (15)$$

while, base exergy efficiency for final combined cycle is

$$\eta_{ex,Base} = \frac{\dot{W}_{T_1} - \dot{W}_{C_1}}{\eta_{CMB}\,\dot{m}_{l_7}\,LHV + \dot{m}_{N_0}\left(h_{N2} - h_{N3}\right)\times\left(1 - \frac{T_{l0}}{T}\right)} \quad (16)$$

RESULTS AND DISCUSSION

The presented cycles in Figs. 4, 1, 3 & 2 are referred as B1-B4, respectively. Figure5 represents the effect of pressure ratio of compressor C1 on the base thermal efficiency of cycles B1-B4. It is notable that as the input energy to open Brayton cycle of cases B2-B4 is the same and invariant, the trends of their base thermal efficiency variations with pressure ratio are similar. But, the absorbed heat by the open Brayton cycle of case B1 varies with pressure changes which causes its different trend. From Fig.5 it is also seen that at some pressures, regenerative heat exchanger 'R' is no longer profitable, because after the pressure ratio 13, its duty is reversed, i.e. instead of pre-heating, it cools the air entering combustion chamber. This is the reason of the different trend of the curve B1 at high pressures. Thus, as a practical solution, it is recommended to use appropriate valves to by-pass heat exchanger 'R' at compressor pressure ratios more than 13. Doing so, curve B3 substitutes curve B1 at these pressure ratios. Therefore, we can say the final combined cycle has the best performance from thermal efficiency standpoint.

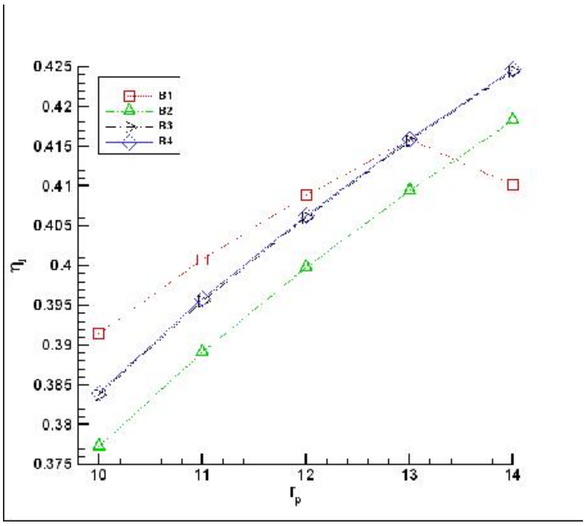

Fig.5. Effect of the pressure ratio of compressor C_1 on the base thermal efficiency

Figure6 contains the net base power delivered by all the mentioned cycles for different pressure ratios of compressor C1. The interesting notion of this plot is the reduction of net base power produced by cycle B1 which is justified as in the higher pressure ratios of C1, the consumed power of C1 is enhanced while the absorbed heat rate by open Brayton cycle is reduced. Therefore, despite the reduction of delivered power of the cycle, the base thermal efficiency of this cycle is increased. However, it is clear that case B1 delivers more power than other cases.

The variations of base exergy efficiency of the aforementioned cycles versus the pressure ratio of C1 are illustrated in Fig.7. From this figure, it is obvious that the second-law efficiency increases with the pressure ratio of compressor C1. The only exception is again for case B1 at pressure ratios more than 13. From this figure again, we can conclude that case B1 which is equipped with by-pass valves has the most exergy efficiency among the proposed cycles.

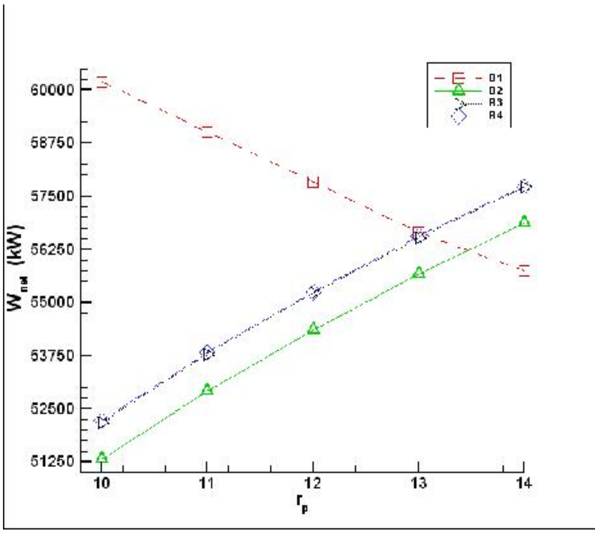

Fig.6. Effect of the pressure ratio of compressor C_1 on the base net power

The effect of the pressure ratio of compressor C1 upon the total thermal efficiency of different cycles is shown in Fig.8. We observe in this graph when the pressure ratio of compressor C1 is increased, the total thermal efficiency of cases B2-B4 is also enhanced; but, the total thermal efficiency of case B1 which is our final combined cycle is decreased. This matter is due to the degradation of regenerative heat exchanger 'R'. Again, the superiority of case B1 is obvious.

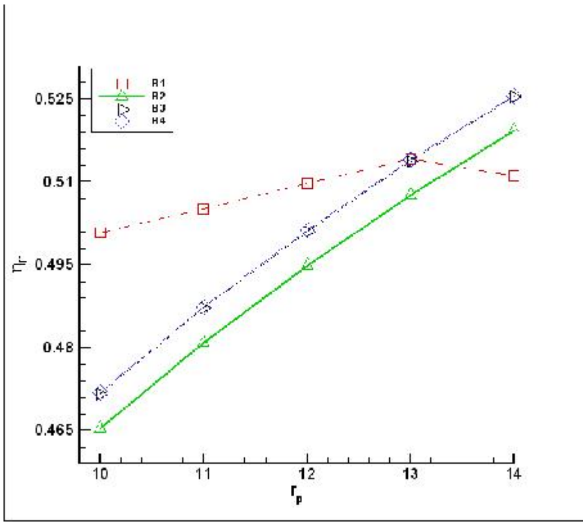

Fig.7. Influence of the pressure ratio of compressor C_1 on the base exergy efficiency

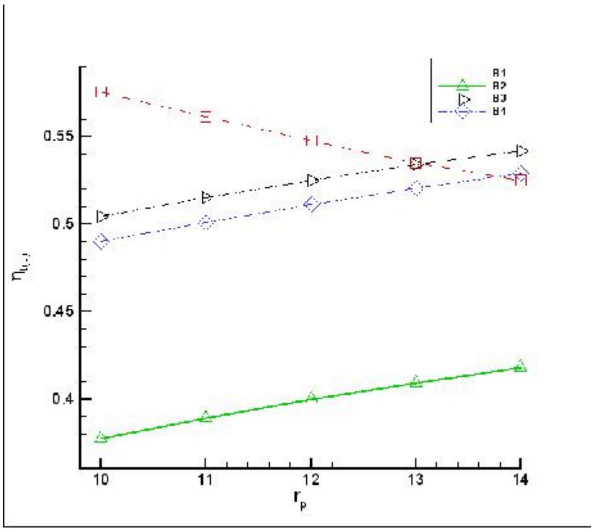

Fig.8. Effect of the pressure ratio of compressor C1 on the total thermal efficiency

A similar trend of Fig.8 is also observable in Fig.9 which shows the curves of total net power versus pressure ratio of compressor C1. This graph appreciates the results of Fig.8 well.

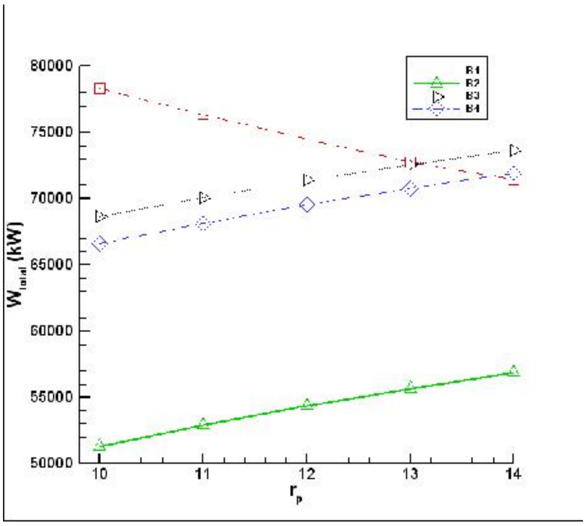

Fig.9. Variation of total net power vs. the pressure ratio of compressor C1

The variations of exergy efficiency of different cycles vs. the pressure ratio of compressor C1 is illustrated in Fig.10. As presented in Fig.9, the total net power delivered by cycle B1 is decreased when the pressure ratio of compressor C1 is increased; but, the input energy rate to the cycle is enhanced which leads to lower second-law efficiencies.

A significant point of Fig.10 is that curve B2 is plotted for its optimal conditions while curve B1 is drawn at initial conditions which differ greatly from optimum situations. Therefore, the illusion of better performance of case B2 should be avoided. Actually, when all the cycles are in optimum conditions, cycle B1 has the best performance from both first and second laws standpoints.

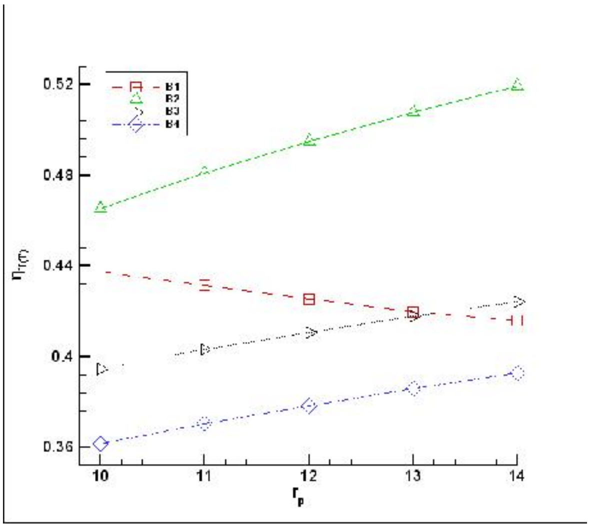

Fig.10. Influence of pressure ratio of compressor C1 on total exergy efficiency

Conclusions

In the present study, an open Brayton cycle was considered as the base case and was further developed by adding different equipment including liquefied natural gas to enhance its performance which resulted in four cycles. Using EES, the proposed cycles were analyzed based on the first and second laws of thermodynamics and it was found out that the final combined cycle performs the best among the proposed cycles; but, further measures should be taken into consideration at high pressure ratios of open Brayton cycle compressor.

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