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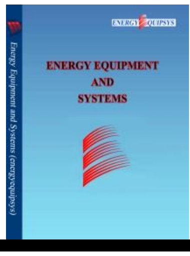

Modeling, analyses, and assessment of a liquid air energy storage (LAES) system

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ABSTRACT

In this research paper, a comprehensive thermodynamic model of a LAES system is carried out. Exergy analysis as a potential tool to determine the location and magnitude of losses is applied. The proposed system is comprised of two sub-systems, one is used for air liquefaction which is considered as energy storage. This step is called the charging process. The liquid air is then stored in a cryo-tank. The liquid air is then directed to a series of turbo-expanders for the discharge process. For the system simulation, a code is developed which links the Refprop software to Matlab software where thermophysical properties are obtained. In order to have a better insight of the analyses, a parametric study is performed. The effect of several main parameters (i.e., the main pressure entering the cold box, compressor inlet temperature and the first compressor pressure) of the storage cycle on efficiency and cost are investigated. The results indicate that the outlet temperature of the final turbine is limited to the temperature that can enter the evaporator. This constrains limits the inlet temperature to the turbine, which reduces efficiency. In addition, the storage cycle pressure is low. An increase in the main pressure to the cold box can be useful in increasing efficiency. Besides, reducing the inlet temperature to the compressors can increase performance.

Keywords: Liquid Air Energy Storage, Exergy Analysis, Cost Assessment, Storage.

1. Introduction

Currently, energy resources are declining, and pollution is increasing, leading to changes in climate conditions. Fossil fuels are expected to run out in the next few decades; therefore, in addition to the proper use of them, humans are looking for suitable and guaranteed alternatives. Advanced technologies offer substitutes such as wind energy, solar energy, and biomass power plants, but these technologies produce much less electricity than traditional power plants, while also having more problems generating electricity. Global electricity demand is considerably high and growing by about 3% every year [1]. For technical reasons, the amount of electricity injected into the grid must always be at the level of consumer demand to prevent power outages or severe damage to the grid, however not solar nor wind energies are available all day. One of the primary and constant concerns of Iran's electricity network is the balance between the supply and demand of electric charge. In most cases, production exceeds demand, and this balance is easy to control. Nevertheless, when demand exceeds supply, controlling this balance is complicated, and if there is no basic thought for it, blackouts might occur in some parts of the country. The most probable time of the power outage is at the

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peak of the network in the summer and is considered a significant problem of the electricity industry. This problem can be solved in different ways that require anticipation, planning, and management. Today, storage technologies are a vital element in balancing and addressing these shortcomings. Because of the growing importance of renewable energy sources, scientists and engineers are looking to increase efficiency and reduce the cost of these technologies. Having energy storage systems alongside renewables is also very important and can reduce the problems of generating electricity from renewable sources.

There is a different way of storing energy at low temperatures called cryogenics. The use of gas liquefaction to store energy was first used by Smith in 1977. This method is used for energy storage to supply sector management (SSM) and stores energy in non-peak times and cheap electricity of the network and returns it to the network in peak and expensive times of electricity [2]. The cryogenic process refers to a liquid or liquefied gas that boils at a low temperature of about -150 degrees Celsius.

1.1 Literature review

Cryogenic Engineering, a regular and organized activity involving the production, storage, and utilization, has been practiced on a large scale for air and helium since 1940. It was also used and developed in various sciences such as physics, chemistry, biology, and the manufacture of high-precision equipment [3]. However, the idea of using gas liquefaction to store energy on a large scale for use in the grid was first presented in an article by Smith (the University of Newcastle in the United Kingdom) in 1977 at the Mechanical Engineering Conference. The results of Smith's research for the processes of compression and adiabatic expansion with a temperature of 1048 K and a pressure of 85 bar, provided an efficiency of 72% [2]. In 1988, Mitsubishi, a major manufacturer of power plants, developed and tested a system for producing and storing liquefied air at low load times to use high-pressure air as inlet air for gas turbines [4]. Since 2006, Highview Power company in the United Kingdom, in collaboration with researchers at the University of Leeds, has begun extensive research on the liquid-air storage system, with studies on the liquid air production system, expansion, and power generation in the turbine, as well as its integration with heat losses of other systems and achieved technically and economically acceptable results [3]. In 2013, this company built the first prototype large-scale power storage and generation unit. The system was built as a pilot at the site of Slough, England, with a capacity of 300 kilowatts of electrical power output from the Cryogenic turbine and was successfully tested. In the 300 kW model, the liquid air storage tank can hold up to 100 tons of liquid air in isolation, and its pressure is less than 10 bars [5, 6]. The efficiency of the pilot test sample increases with the heat stored in the system, and an efficiency of about 60% (AC to AC) was recorded when using this energy to heat the compressed air out of the liquid air storage tank when passing through the turbine [6]. Morgan et al. [7] experimentally showed that increasing the charge pressure of the air tank, increasing the pressure of the liquid air pump, and increasing the temperature of the inlet air to the turbine increase the overall efficiency of the system and the system can deliver more electrical energy to the grid compared to the amount of received energy. Dincer and Rosen [8] presented an exergy analysis for cryogenic systems and gas liquefaction. They presented and calculated the energy-exergy relationships for a simple gas liquefaction system called Linde-Hampson. Lee et al. [9] coupled a cryogenic storage system (for air or nitrogen) with a nuclear power plant and presented the results of their research for load shifting. Nuclear power plants prefer to operate at maximum load, and for this purpose, their integration with desalination plants or storage systems can be beneficial to their performance. Extensive research has also been conducted on air separation and liquefaction units, and air components such as oxygen have been used for a variety of applications, such as gas turbines or coal-fired power plant combustion [10-14]. Air is composed of nitrogen (78%), oxygen (21%), and argon (1%), and these gases can be separated from the air; because their boiling temperature is different. Nitrogen boils at -196 ° C, oxygen at -183 ° C, and argon at -186 ° C [3]. Lee et al. [15]

investigated hybrid system integration of storage systems with cryogenic and solar energy. First, each of the cryogenic and solar power generation systems was modeled separately, and the output power of each was calculated. The results of this study indicate that by using this integration and using solar energy to heat the air entering the turbine in the liquid air storage system, the electrical energy output from the two systems increases by a total of 30%. Zhou et al. [16] performed the thermodynamic analysis of a liquid air storage system based on the Linde-Hampson cycle, in which a choke valve is used for liquefaction, and examined the crucial parameters on the efficiency of the storage set. They showed that the liquid air energy storage system has certain subtleties and complexities that require careful analysis and even optimization to obtain optimal efficiency. They also acknowledged in their discussion and conclusion that the liquid air storage system is complex and challenging and that optimization studies on this system need to be conducted in future research. In summary, the novelties of this research paper can be as:

- Thermodynamic modeling of a LEAS along with exergy analysis.
- A comprehensive parametric analysis of the effective design parameters on the system performance
- Techno-economic assessment of the LEAS for future consideration.

2. Liquid air energy storage cycle model description and simulation

The most critical challenge in the present study is the lack of access to cycle design and modeling methods. Except for the limited published information of the desired cycle, no specific data are available. The data published in reliable sources are available on the pilot cycle, and its industrial cycle information is severely restricted and protected because the plant is in the process of development, research, and commercialization. One of the disadvantages of the pilot cycle is that it is designed for research and therefore has many ambiguities and additional points embedded in it. Air separation for the regenerator, storage limit in the high-grade cold store, as well as the presence of a pressure controller at the inlet of the turbines and the main storage tank are some of the things that exist only because the cycle is pilot. However, due to limited resources, the pilot cycle must first be considered. In fact, in the first step, the design philosophy is extracted, and in the next step, changes are made on the cycle to be adapted to the conditions of the desired gas turbine cycle.

2.1. Design of a pilot cycle of energy storage with liquid air

In this research, the pilot cycle is designed, and its schematic is shown in Fig.1.

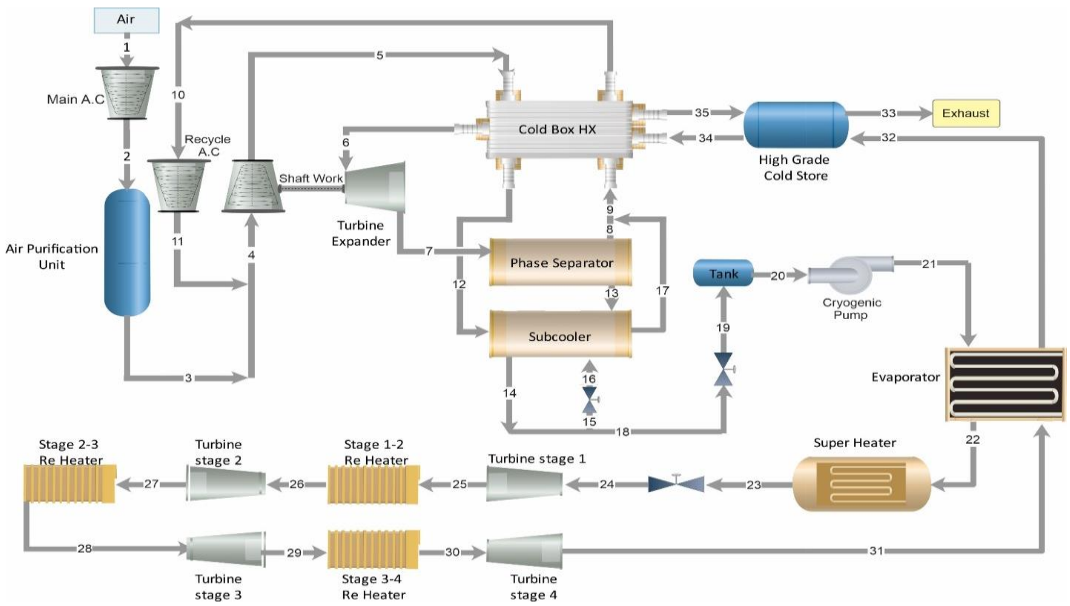

Fig.1: Schematic diagram of the proposed LAES system

The logic of this cycle is as follows:

- **Storage phase:**

Compressors compress ambient air. At the compressor outlet (shown in Fig.1), there is an air-cooled heat exchanger that lowers the inlet temperature to the next compressor to ambient temperature. The air temperature in the cold box drops significantly to enter two-phase mode after channeling to the expander. In two stages, first in the expander and then in the throttle valve TV_1, (line 15-16), the gas pressure decreases adiabatically to provide the required cooling. After separating the liquid air, this air is stored in a storage tank.

- **Power generation phase:**

In the power generation phase, liquefied air enters the pump to increase its pressure.

In the evaporator, the liquid air is converted to gas, and cold air is sent to the high-grade cold store (HG) heat exchanger to be used in the storage phase. After the evaporator, air enters the first superheater, and its temperature reaches the design temperature and then enters the turbine. The above process is repeated in several steps to bring the air pressure closer to atmospheric pressure. Cycle design is also divided into two stages. The first is the design of the power cycle, and the second is the storage cycle. The main input of the design is the amount of power required in a specified time. For example, 10 MW during peak hours from 12 pm to 4 pm. On the other hand, the specific parameter of the design of the storage cycle is only the amount of allowed operating hours, for example, from 12 am to 6 am (low load). In other words, the following parameters are the main design data:

$$\begin{cases} W_d & \text{Peak Demand} \\ T_d & \text{Peak Duration} \\ T_{ch} & \text{Charging Duration} \end{cases}$$

If the output power of the turbine i is W_i , then the total output power is:

$$W_d = \sum W_i - W_P = m_d \left(\sum w_i - w_P\right) \quad (1)$$

Therefore, the required flow during the power generation interval is:

$$\dot{m}_d = \frac{\dot{W}_d}{(\sum w_i - w_P)} \quad (2)$$

Then the mass flow rate obtained for the storage cycle is obtained with the following relation:

$$m_{ch} = \frac{T_{ch}}{T_d} m_d \quad (3)$$

The above equations represent that the cycle must be solved without knowing the masses. The design equations are developed as relative masses to eliminate at least one unknown and be independent of the capacity design.

- **Power cycle calculations**

In power cycle design, design variables are:

1- Storage tank temperature and pressure
2- Pump pressure ratio
3- Turbine pressure ratio
4- Pinch Point Evaporator
5- Outlet temperatures of superheaters

- **Pump calculations:**

The equations used in the pump are as follows:

$$P_{out} = r_p \times P_{in} \quad (4)$$

$$\eta_{is} = \frac{(h_{out} - h_{in})_{is}}{(h_{out} - h_{in})} \quad (5)$$

$$w = (h_{out} - h_{in}) \quad (6)$$

- **Turbines calculations:**

The following equations are also used for turbines:

$$P_{out} = r_T \times P_{in} \quad (7)$$

$$\eta_{is} = \frac{(h_{in} - h_{out})}{(h_{in} - h_{out})_{is}} \quad (8)$$

$$w = (h_{in} - h_{out}) \quad (9)$$

For the evaporator, it should be noted that the exhaust air enters the evaporator from the last stage of the turbine and then enters the HG storage by losing its heat and evaporating liquid air. The diagram of temperature changes of this heat exchanger is shown in Fig.2.

According to Fig.2:

$$T_{pinch} = T_{h_{out}} - T_{c_{in}} \quad (10)$$

The law of energy conservation considers the same mass flow rate for both hot and cold currents, giving the enthalpy of the evaporator output:

$$h_{c_{out}} = h_{h_{in}} - h_{h_{out}} + h_{c_{in}} \quad (11)$$

Furthermore, for pressure using the pressure drop coefficient, the output pressure for both hot and cold lines is calculated as follows (f is the drop percentage)**:**

$$P_{out} = (1 - f) \times P_{in} \quad (12)$$

- **Superheater calculations**

The variation of superheater temperature is shown in Fig.3. In superheaters, the inlet and outlet temperatures of the cold line are specified. Hotline conditions are unclear because different heat sources can be used. Since the hotline in this study is the exhaust gases from the turbine, so the inlet temperature of the hotline is known, but the outlet temperature is unknown. There are two possible scenarios. First, the gas thermal energy is sufficient to heat the air to the design temperature before the turbine**.** Second, thermal energy is not enough to reach this temperature. In this case, the hotline can only reach the minimum temperature (dew point)**.**

$$T_{h_{out}} - T_{c_{in}} = T_{pinch} \quad (13)$$

$$T_{h_{in}} = T_{ambient} \quad (14)$$

$$T_{c_{out}} = T_{h_{in}} - T_{pinch} \quad (15)$$

$$Q = m_c \times (h_{c_{out}} - h_{c_{in}}) \quad (16)$$

$$\dot{m}_h = \frac{\dot{Q}}{h_{h_{in}} - h_{h_{out}}} \quad (17)$$

If the outlet cold flow temperature is significant, it can be used as a cold source. One use of this source is storage for the input temperature to the compressors in storage mode. If the turbine outlet temperature is higher than the ambient temperature, these heat exchangers will not be considered in the design.

- **Cycle modeling in storage mode**

In storage, first, the air pressure increases, then the temperature decreases, and finally, in the expansion turbine, the drop-in air pressure causes the temperature to decrease, and the air becomes two-phase. The most important heat exchanger of this cycle is called COLD BOX, in which a large volume of heat transfer takes place. The cooling stored in HG absorbs a significant amount of air heat. According to the cycle, the HG flow rate is calculated from the following equation

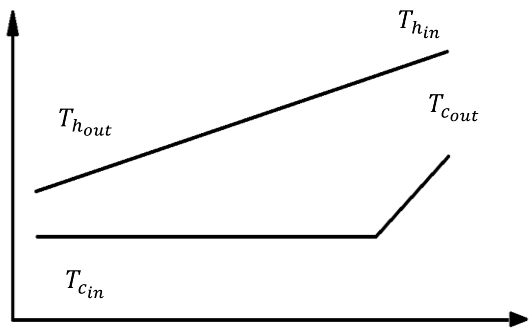

Fig.2: Temperature profile of the evaporator

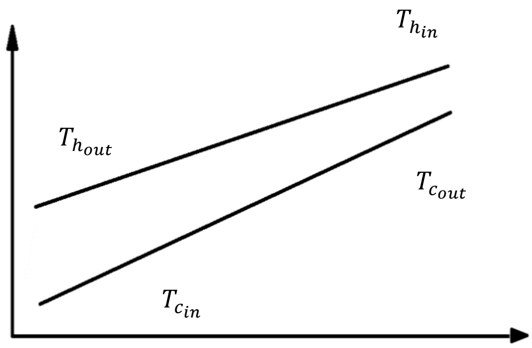

Fig.3: Temperature profile in the heat exchangers superheaters

$$m_{HG} = m_d \times \frac{T_d}{T_{ch}} \quad (18)$$

- **Compressor modeling**

The compressors in the cycle include the compressor package and the cooler. In modeling, first, the compressor is modeled, and then the amount of dissipated heat to the environment or coolant is calculated. Heat exchanger output temperature and compressor pressure ratio are design variables.

$$P_{out} = r_c \times P_{in} \quad (19)$$

$$\eta_{is} = \frac{(h_{out} - h_{in})_{is}}{(h_{out} - h_{in})} \quad (20)$$

$$w = (h_{out} - h_{in}) \quad (21)$$

The compressed exhaust air is then cooled to T_{ac}. The amount of dissipated heat is:

$$Q = m \times \left(h_{out} - h(T_{ac}, P_{out})\right) \quad (22)$$

These equations are used for all compressors.

- **Cold Box governing equations**

In a cold box by writing the mass and energy balance equations and assuming that the amount of heat transfer to the environment can be neglected:

$$\dot{m}_9 = \dot{m}_{10} \quad (23)$$
$$\dot{m}_5 = \dot{m}_6 + \dot{m}_{12}$$
$$\dot{m}_{34} = \dot{m}_{35}$$
$$\dot{m}_9 h_9 + \dot{m}_5 h_5 + \dot{m}_{34} h_{34} =$$
$$\dot{m}_{10} h_{10} + \dot{m}_{35} h_{35} + \dot{m}_6 h_6 + \dot{m}_{12} h_{12}$$

At the outlet of the expander, fluid is in two-phase mode. Expander equations are similar to turbines. For phase separators, the equation is:

$$h_8 = h(Q = 1, P = P_7, x = x_g) \quad (24)$$

In the above equation x_g is the composition of the fluid in the gaseous state at the separator pressure.

$$h_{13} = h(Q = 0, P = P_7, x = x_l) \quad (25)$$

$$m_g = Q \times m_7 \quad (26)$$

$$m_l = Q \times m_7 \quad (27)$$

The separated liquid then enters the heat exchanger, which lowers the temperature of the main fluid (line 12) to below the saturation temperature. Sub-saturation temperature is a design parameter. In order to ensure that the design temperature is reached, a branch line from the mainline enters the heat exchanger after passing through a throttle valve. The following equations are used to model this heat exchanger:

$$\dot{m}_{12} h_{12} + \dot{m}_{13} h_{13} + \dot{m}_{16} h_{16} = \quad (28)$$
$$\dot{m}_{14} h_{14} + \dot{m}_{17} h_{17}$$
$$h_{14} = h_{15} = h_{16}$$

- **Mass and energy balance of the storage cycle**

In the above equations, the mass flow of different points is not known and must be calculated by solving the system of equations. To solve the above equations, the relative mass flow rates are defined:

$$r_1 = \frac{m_{13}}{m_{12}} \quad (29)$$

$$r_2 = \frac{m_{16}}{m_{12}} \quad (30)$$

$$r_3 = \frac{m_6}{m_5} \quad (31)$$

$$r_p = \frac{m_{35}}{m_5} = \frac{m_{18}}{m_5} \quad (32)$$

Considering the above equations, the energy equations are transformed into the following form:

$$r_p = \frac{(r_3)(\Delta h_6) - h_5 + h_{10}}{h_{10} - h_{14} - \Delta h_{35}} \quad (33)$$

$$r_2 = 1 - \frac{r_p}{1 - r_3} \quad (34)$$

$$r_1 = \frac{(1 - Q) r_3}{1 - r_3} \quad (35)$$

$$\vec{x}_{17} = \frac{\overrightarrow{x_{13}} r_1 + \overrightarrow{x_{16}} r_2}{r_1 + r_2} \quad (36)$$

$$h_{17} = H(P_{17}, Q, \vec{x}_{17}) \quad (37)$$

$$h_{12} = h_{14} - (r_1 h_{13} + r_2 h_{16} - (r_1 + r_2) h_{17}) \quad (38)$$

$$r_3 = \frac{(1 - r_p)(-h_{10} + h_9) - r_p \Delta h_{35} + h_5 - h_{12}}{h_6 - h_{12}} \quad (39)$$

Using the above equations, we can calculate the mass ratios that ultimately give the mass flows. The sum of the above equations leads to cycle modeling. Pressures at different points are design parameters. The inlet temperature of the turbine is also a design variable.

3. Results and discussion

In this part, we try to assess the system from both technical and economical viewpoints.

3.1 Estimation of the cost of the cycle

The costs of doing a project are:

- Equipment purchase and installation cost
- Cost of electrical and mechanical installations
- Setup and operation costs
- Maintenance costs
- Land related costs
- Insurance and other expenses

Part of these costs is related to the design of the cycle, and part is related to the method of financing. Cycle-related costs associated with the design include:

- Cycle component costs
- Cost of electrical and mechanical work
- Commissioning and operation
- maintenance

Here only the above costs are considered and calculated. The cost estimation functions referred to as Table (1) are presented as follows [17]:

$$C_{comp} = \frac{c_c \dot{m}}{0.92 - \eta} r \ln(r) \quad (40)$$

$$C_{Turb} = c_T \frac{\dot{m}}{.92 - \eta} log(r)\big(1 + exp(0.036TIT - 26.4)\big) \quad (41)$$

$$C_{TExp} = c_{TE} \dot{W}^{0.7} \quad (42)$$

$$C_{Gen} = c_G \dot{W}^{0.95} \quad (43)$$

$$C_{Pump} = c_P \dot{W}^{0.71} \quad (44)$$

For heat exchangers, the following equations are used [7].

$$C_{Heat} = c_H((A(ln(c_H) - ln(c_0)) + B) \quad (45)$$

$$c_H = \frac{Q}{LMTD} \quad (46)$$

The constant values of the above equation are variable for different heat exchangers and different capacities. The cost of electrical-mechanical work is 15% of the total cost, and the cost of commissioning and operation is 1% of the total cost. Compressors, turbines, and cold boxes are the most expensive components of the cycle. The share of cycle costs by components is shown in Fig.4.

The important thermodynamic parameters of different points of the designed cycle of energy storage with liquid air, including temperature, pressure, enthalpy, entropy, exergy, and mass flow shown in Fig.1, is presented in Table (2).

Table 1: Cost of each component of the system

Total Compressor Work	8186 (kW)
Total Turbine Work	640 (kW)
All costs * 1000$	
Total Cost	3435
Total Compressors Cost	575
High-Grade Storage	113
Storage Tank	45
Evaporator	176
Cold Box	386
Subcooler	15
Pump	97
Turbines	829
Expander	627
Generator	22
Mechanical -Electrical Works	515
Commissioning	34

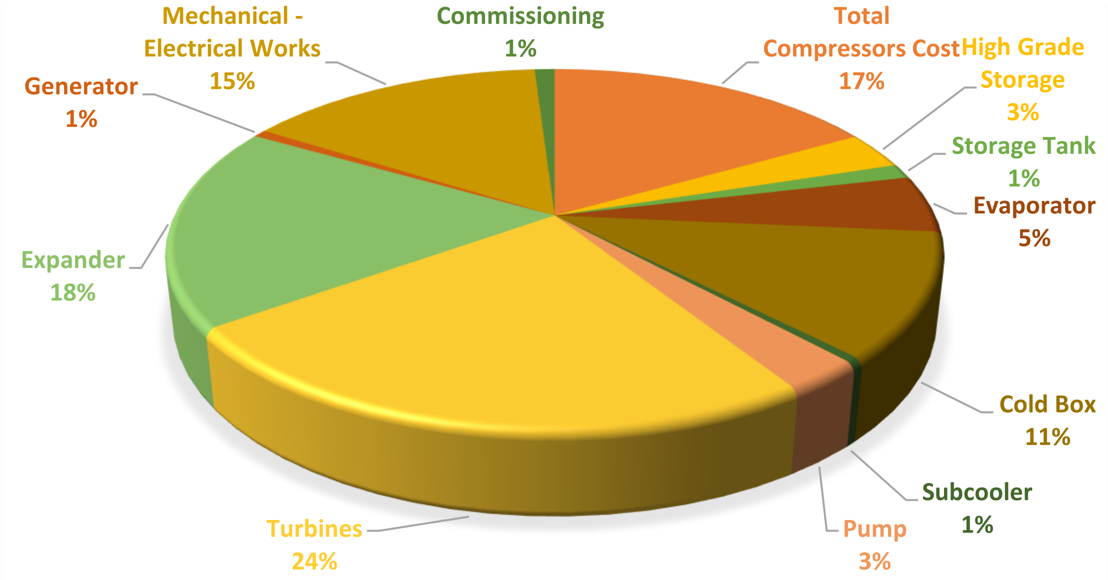

Fig.4: Cost breakdown of the LAES system

Table 2: Thermodynamic properties of each state point in the proposed LAES system

point	T (k)	P (kPa)	h (kJ/kg)	s (kJ/kg K)	Ex (kJ/kg)	Q (or *x*)	m (kg)
1	298.15	100.00	298.45	6.8641	0.00	1.00	0.40
2	297.95	1000.00	296.19	6.1962	196.89	1.00	0.40
3	297.95	1000.00	296.19	6.1962	196.89	1.00	0.40
4	297.95	1000.00	296.19	6.1962	196.89	1.00	2.36
5	297.95	1220.00	295.70	6.1376	213.86	1.00	2.36
6	130.45	1199.63	116.61	5.2496	299.54	1.00	1.83
7	83.79	130.00	74.35	5.4180	207.06	0.97	1.83
8	82.32	110.00	79.55	5.5242	180.59	1.00	1.78
9	82.45	110.00	79.60	5.5265	179.98	1.00	1.97
10	295.95	110.00	296.21	6.8292	8.16	1.00	1.97
11	297.95	1000.00	296.19	6.1962	196.89	1.00	1.97
12	109.55	1199.63	-26.29	3.9730	537.26	0.24	0.53
13	82.20	110.00	-129.45	3.0421	695.64	0.00	0.06
14	98.23	1199.63	-87.09	3.4051	645.77	0.00	0.53
15	98.23	1199.63	-87.09	3.4051	645.77	0.00	0.13
16	79.92	110.00	-87.09	3.4676	627.15	0.19	0.13
17	83.81	110.00	79.97	5.5413	173.99	1.00	0.19
18	98.23	1199.63	-87.09	3.4051	645.77	0.00	0.40
19	98.30	800.00	-87.09	3.4103	644.22	0.00	0.40
20	98.30	800.00	-87.09	3.4103	644.22	0.00	1.98

21	103.37	5600.00	-74.87	3.4703	638.55	0.00	1.98
22	150.04	5550.16	82.73	4.6829	434.62	1.00	1.98
23	337.60	5494.66	329.36	5.8126	344.43	1.00	1.98
24	337.05	5110.00	329.36	5.8335	338.20	1.00	1.98
25	298.05	2469.79	293.01	5.9270	273.95	1.00	1.98
26	333.55	2409.78	330.07	6.0516	273.88	1.00	1.98
27	299.85	1409.23	297.22	6.1015	226.16	1.00	1.98
28	329.25	1349.19	327.50	6.2103	224.00	1.00	1.98
29	272.55	518.92	271.54	6.2975	142.03	1.00	1.98
30	322.85	478.96	322.60	6.4924	134.98	1.00	1.98
31	262.15	169.66	262.03	6.5822	47.62	1.00	1.98
32	106.27	118.77	104.42	5.7697	132.27	1.00	1.98
33	283.24	98.58	283.45	6.8166	-0.84	1.00	1.98
34	114.47	120.48	112.85	5.8421	119.12	1.00	0.4
35	295.95	100.00	296.21	6.8292	8.16	1.00	0.4

3.2. Parametric analysis of storage cycle

In this section, the effect of several main parameters of the storage cycle on efficiency and cost are investigated. The cost here refers to the total cost of implementation. The most important parameter of the storage cycle is the pressure of the mainline of the fluid entering the cold box (line 5). Figure 5 shows the impact of this parameter on the performance and investment cost of the storage cycle**.**

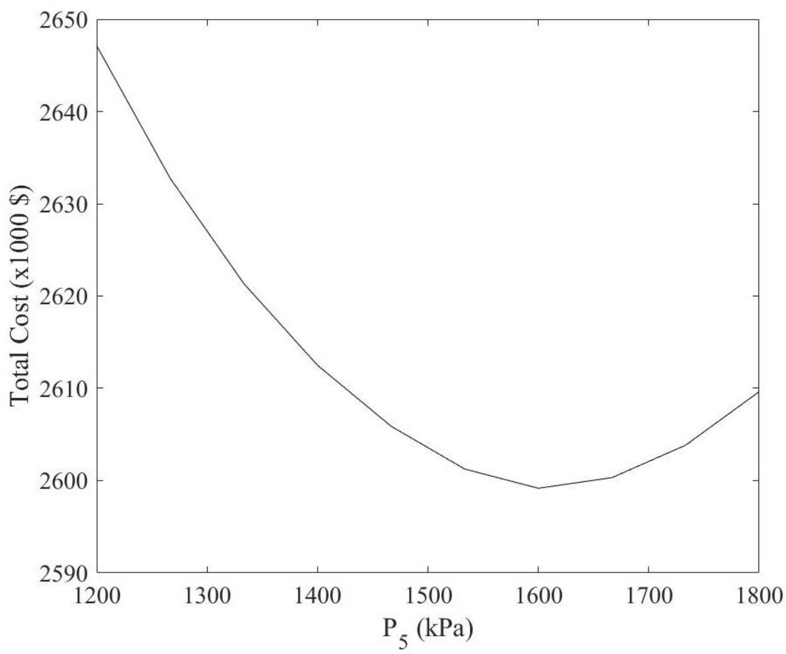

Fig.5: The effect of P_5 on the cost of the proposed system

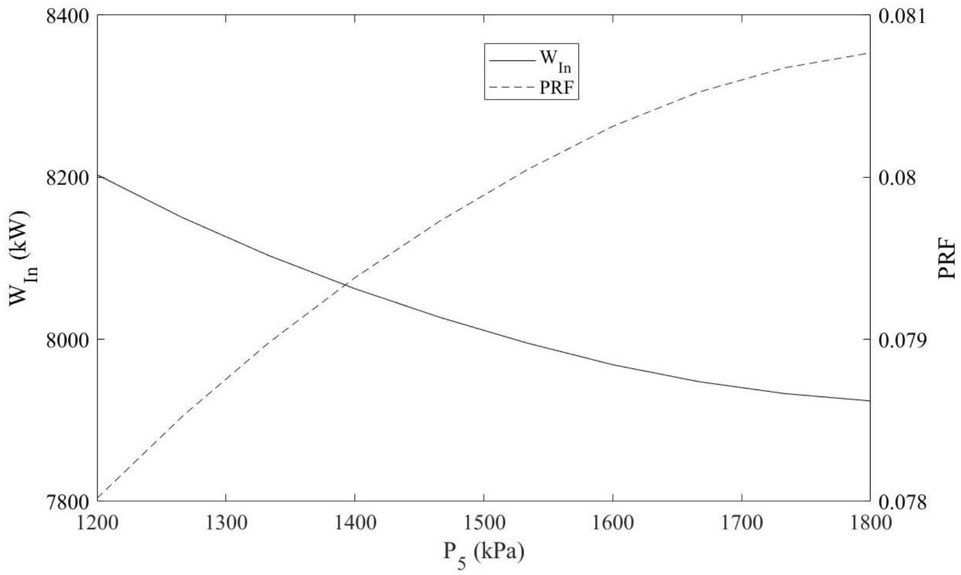

Fig.6: The effect of P_5 on the system efficiency

As shown in Fig.6, increasing the pressure increases the cycle efficiency. As the pressure increases, the outlet fluid in the expansion turbine experiences a more significant reduction in enthalpy, thus reducing the fluid flow rate required to produce a certain amount of liquid air. Therefore, the input work will be reduced. The cost also decreases first due to the shrinkage of the components but then due to the increase in the cost of the compressors as a result of the increased pressure. Compressor inlet temperature has a direct effect on cycle performance. Reducing the inlet temperature to compressors can significantly reduce the input work to the storage cycle. As shown in Fig.7, lowering the temperature leads to lowering the cost and increasing the performance. The reason for the increase in performance is the reduction in compressor power, while the cost is reduced due to the reduction of heat exchange in the Cold Box and its dimensions.

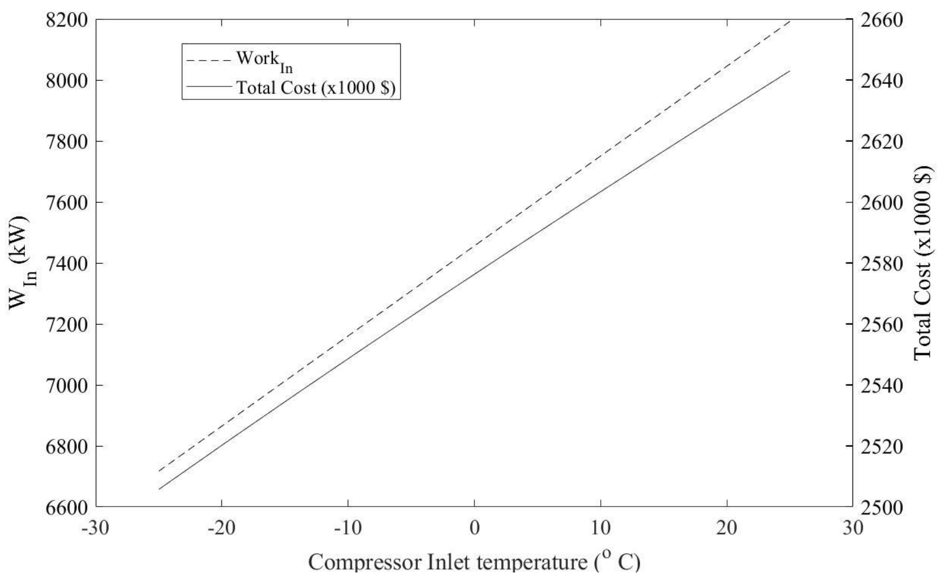

Fig.7: The effect of compressor inlet temperature on the system performance

Another parameter is the contribution of the primary to the secondary compressor in increasing the pressure from the inlet to the line 5 pressure. Increasing the pressure of the first compressor, reduces the cycle performance and increases the required input work (Fig.8). The main reason is that the return line compressor must also work in the desired pressure, which increases the pressure.

4.Conclusion

As mentioned in this paper, since this cycle is a pilot, its efficiency is low. According to the schematic of the cycle and parametric analysis, a few points in the cycle reduce the efficiency:

- The power cycle and storage cycle are related by HG. In the current design, the amount of cooling storage that can be stored during power generation depends on the turbine flow and is limited.
- The outlet temperature of the final turbine is limited to the temperature that can enter the evaporator. This limitation limits the inlet temperature to the turbine, which reduces efficiency.
- The storage cycle pressure is low. Increasing line 5 pressure can be useful in increasing efficiency.
- Reducing the inlet temperature to the compressors can increase performance.

Therefore, the separation of HG and the increase of the inlet temperature to the turbines can increase efficiency. This change, in addition to increasing the cycle efficiency, can lead to the integration of this cycle and the gas cycle.

References

[1] Patterson R., “World Energy 2016-2050: Annual Report”, Political Economist, Peak Oil Barrel, June 2016.

[2] Smith E.M., “Storage of Electrical Energy Using Supercritical Liquid Air”, Proceedings of the Institution of Mechanical Engineers, pages 289-298, 1977.

[3] Taylor P. et al, Pathways for energy storage in the UK, “Liquid Air in the energy and transport systems”, THE CENTRE FOR LOW CARBON FUTURES, 2013.

[4] Kishimoto K. et al., “Development of Generator of Liquid Air Storage Energy System”, Mitsubishi Heavy Industries Technical Review, October 1998.

[5] HIGHVIEW POWER STORAGE COMPANY, liquid air energy storage, 2014.

[6] “Results from performance tests carried out by Highview at the Slough site”, personal communication, February 2013.

[7] Morgan R., Nelmes S., Gibson E., Brett G., “Liquid air energy storage – Analysis and first results from a pilot scale demonstration plant”, Applied Energy, 137, pages 845–853, 2015.

[8] Dincer I. and Rosen M.A., “Exergy Analysis of Cryogenic and Liquefaction Systems”, Chap. 15, Published by Elsevier Ltd, 2013.

[9] Li Y., Hui H., Wang S., Jin Y., Li D., Wang X., Ding Y., “Load shifting of nuclear power plants using cryogenic energy storage Technology”, Applied Energy 113, pages 1710–1716, 2014.

[10] Pacheco K.A., Li Y., Wang M., “Study of Integration of Cryogenic Air Energy Storage and Coal Oxy-fuel Combustion through Modelling and Simulation”, Proceedings of the 24th European Symposium on Computer Aided Process Engineering – ESCAPE 24, Budapest, Hungary, June 15-18, 2014.

[11] Fu Q., Kansha Y., Song C., Liu Y., Ishizuka M., Tsutsumi A., “An elevated-pressure cryogenic air separation unit based on self-heat recuperation technology for integrated gasification combined cycle systems”, Energy 103, pages 440- 446, 2016.

[12] Aneke M., Wang M., “Process analysis of pressurized oxy-coal power cycle for carbon capture application integrated with liquid air power generation and binary cycle engines”, Applied Energy 154, pages 556–566, 2015.

[13] Sapali S.N., Raibhole V.N., “Exergy Analysis of Cryogenic Air Separation Unit Integrated with Biomass Gasifier”, Proceedings of the World Congress on Engineering and Computer Science, San Francisco, USA 23-25 October, 2013.

[14] Ding Y., Tong L., Zhang P., Li Y., Radcliffe J., Wang L., “Electrical Energy

Storage Techniques", Chap. 9, Liquid Air Energy Storage, Elsevier Inc., 2016.

[15] Li Y., Wang X., Jin Y., Ding Y., "An integrated solar-cryogen hybrid power system", Renewable Energy 37, pages 76-81, 2012.

[16] Xue X.D., Wang S.X., Zhanga X.L., Cui C., Chen L.B., Zhou Y., Wang J.J., "Thermodynamic analysis of a novel liquid air energy storage system", Physics Procedia 67, pages 733– 738, 2015.

[17] Yongliang Li, Xiang Wang, Yulong Ding, "A cryogen-based peak-shaving technology: systematic approach and techno-economic analysis", INTERNATIONAL JOURNAL OF ENERGY RESEARCH, John Wiley & Sons, Ltd, 2011.