Liquid air/nitrogen energy storage and power generation system for micro-grid applications

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Liquid air/nitrogen energy storage and power generation system for micro-grid applications

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Abstract

The large increase in population growth, energy demand, CO_2 emissions and the depletion of the fossil fuels pose a threat to the global energy security problem and present many challenges to the energy industry. This requires the development of efficient and cost-effective solutions like the development of micro-grid networks integrated with energy storage technologies to address the intermittency of renewable energy sources, provide localized electricity production, and smooth out power demand and supply curve. Among other energy storage systems, the cryogenic energy storage (CES) technology offers the advantages of relatively large volumetric energy density and ease of storage. This paper concerns the thermodynamic modeling and parametric analysis of a novel power cycle that integrates air liquefaction plant, cryogen storage systems and a combined direct expansion with closed Rankine power recovery system using two cryogens, liquid nitrogen, and liquid air. This cycle is part of a micro-grid system that provides electricity for a typical 50 unit residential building using either renewable energy sources or national grid off-peak electricity. This power cycle was modeled using MATLAB integrated with REFPROP software to investigate its performance at various operating conditions. Results showed that using liquid air as the working cryogen can significantly improve the cycle performance compared to that of liquid Nitrogen at all operating conditions, yielding maximum round trip efficiencies of 63.27% and 84.15% respectively. Also results showed that as the cryo-turbine efficiency and recovery expansion ratio are increasing the cycle round trip efficiency and network will increase, while as the compressor efficiency increases the round trip efficiency increases and the network decreases to reach the best value at 84% to produce round trip efficiency 80.62% and work 397KJ/kg for the liquid air condition.

1. Introduction

The continuous growth in population and urbanization levels have increased energy consumption, CO2 emissions, the demand-supply mismatch and present major challenges to the energy industry [1]. This requires the development of novel solutions that optimize energy use and minimize fuel consumption and emissions [2]. Residential buildings are responsible for a significant proportion (approximately 30%) of the global energy consumption and carbon emissions due to a large number of populations living in them [3]. Therefore, the ability to control the timing and levels of electricity consumption in buildings will have a significant impact on the electricity demand-production profile and plays a major role in developing sustainable energy strategies [4]. Recently, there has been considerable interest in microgrids (MGs) for various buildings like residential, commercial and industrial as an attractive solution to local efficient energy generation, reduce carbon emissions and national grid losses [5]. A microgrid is a local system to generate, store and provide energy for buildings as a standalone system or connected to the main utility grid, using wind turbines, fuel cells, photovoltaic panels, diesel generator, and microturbines for power generation [6]. MGs are flexible, smart and active power systems that able to improve the national grid efficiency and security, thus allowing more integration of renewable energy sources (RESs) [7]. With the increased integration of renewable energy sources in micro-grids networks, there is a need for effective solutions to manage the time shift between energy production and demands [8]. Although load shifting has been proposed as a mean to adjust power demands in buildings and reduce peak electricity load, like using water heaters, refrigerators and washing machines, however, not every type of power demand is suitable for load shifting, including lighting and cooking [9].

Energy storage (ES) offers the ability to manage the surplus energy production from intermittent renewable energy sources and national grid off-peak electricity with the fluctuation of electricity demand and provide the required flexibility for efficient and stable energy network [10]. The main storage technologies are mechanical, electrical, chemical and thermal energy storage technologies, detail description and comparison of these storage technologies in terms of system energy density, the efficiency of recovery, development level, capital cost, advantages, and disadvantages are presented in [11]. Currently, the large-scale energy storage plants with a storage capacity of 100MWh used worldwide are Pumped Storage Hydropower (PSH) and Compressed Air Energy Storage (CAES) [12]. The PSH is a mature storage technology which makes 95 GW of the worldwide storage capacity, while the CAES technology is growing, for example, the McIntosh site in Alabama generates 226 MW of electricity using CAES technology [13]. Both PSH and CAES suffer from drawbacks in terms of geographical restrictions, high capital costs, environmental impacts and limited potentials for future development [14]. Cryogenic energy storage (CES) technology offers the advantages of relatively large volumetric energy density, ease of storage and offers the potential to overcome the PSH and CAES drawbacks [15].Also, this system is economically doable with relatively low capital cost (3–30 \$/kW h) [16]. Cryogens normally refer to a liquid media (liquefied gasses) that boils at temperatures below -150^{0}C such as liquefied natural gas (LNG), liquid air

(Lair) and liquid nitrogen (LN2) [17]. The use of cryogen as an energy carrier in energy storage system is more efficient than other energy carriers since the energy is stored through decreasing the internal energy while increasing the exergy of the cryogen [16]. Despite the high energy density, safety, availability and very low environmental impacts, the use of liquid air/nitrogen as an energy carrier has not been extensively exploited [18]. Recently, increased interest in liquid air energy storage technology (LAES) for grid scale application has been reported and few pilot plants are developed such as [19] which used packed beds to improve efficiency of LAES by 50% and [20] worked with optimum pressure of the working fluid 20 MPa to achieve an overall efficiency of 50% using a waste heat source.

With the increased interest in cryogenic energy storage, there is a need to develop efficient energy recovery technologies that exploit the cryogen stored energy. Several studies were carried out investigating the use of combined cycles using a range of working fluids, as summarized in Table 1.

Table1
Reported literature for combined cryogen cycles

Author	**Working fluid**	**Cycle arrangement**	**Results**
Feifei and Zhang [21]	Liquid Natural Gas LNG, nitrogen, water ammonia	two schemes combining an open expansion with Brayton cycle and Rankine cycle	the thermal efficiency of scheme 1 is 60.94%, for scheme 2 is 60%
Guizzi et al. [22]	LAir, propane, methane,essotherm 650	stand-alone air liquefaction and power recovery plant	the thermal efficiency of around (50-60%)
Li et al. [23]	water, nitrogen, methane, Thermal-oil 66	integrated solar-cryogen hybrid power system	Integrated system can increase the power by 30% compared to the two (solar and cryogen) systems acting separately
Chino and Arakian [24]	LAir, flue gas	combines a gas turbine cycle with a liquid air storage system	Energy storage efficiency reaches 74%
Li et al. [25]	LAir, flue gas, nitrogen, oxygen, helium	combines a gas turbine cycle with a liquid nitrogen storage system and CO_2 captured as dry ice	the thermal efficiency reaches 70%
Kantharaj et al. [26]	LAir	Integrated Liquid Air Energy Store (LAES) with (CAES)	the thermal efficiency reaches 67%
Li et al. [27]	Steam, LAir	integration of nuclear power generation and a CES subsystem	the thermal efficiency reaches 71.2%
Li et al. [28]	nitrogen, LNG, LAir, flue gas	combining an open expansion, Brayton cycle	thermal efficiency reaches 64%
Morgan et al. [29]	LAir	stand-alone air liquefaction and power recovery plant	thermal efficiency reaches 60%
Smith [30	LAir, water, Freon	Cryo-storage power plant	the thermal efficiency reaches 72%
Ordonez CA.[31]	nitrogen	closed Brayton cycle,	specific energy reaches 482 kJ/kg
García RF et al. [32]	LNG, argon, methane	combining an open expansion, Rankine cycle	Exergy efficiency reaches to 85.60%
Ahmad A et al. [33]	nitrogen, xenon	combining an open expansion, Rankine cycle	the recovery efficiency of 78%

Generally, most of the work described in the reported literature is related to grid scale power cycles, the need to use external heat sources, and only using direct expansion for the cryogenic power recovery cycle, which in general limits the advantages of these cycles. To the best of the authors' knowledge, it is only Du et al. [34] who is investigated the feasibility of a small-scale (lab scale) cryogenic energy storage system with a power capacity of 5kW and total electricity storage capacity of approximately 10kWh. Their experimental results showed that the efficiency of the small-scale cryogenic energy storage system using the large engine for generation can reach up to 44%. Therefore, this work develops a thermodynamic modeling of a novel power cycle for a micro-grid application that integrates air liquefaction plant, heat and cold storage, cryogen storage and a power recovery system, which combines direct expansion with a closed Rankine cycles, for two cryogens Lair and LN2. In this proposed cycle, the heat rejected from the liquefaction process will be stored and then used as an input to the power recovery subsystem to improve the efficiency. Also, cold is released by the power recovery subsystem, stored, and then used as an input to the liquefaction process to improve its efficiency.

To address this issue two schemes for cryogenic energy storage power plant suitable for a micro-grid system in the large residential building are proposed. The first scheme upgrades the existing oxygen liquefaction plant by integrating it with a power recovery cycle which combines both open and closed Rankine cycles using the produced LN2. The current oxygen liquefaction plants produce surplus cryogenic fluids mainly LN2 without using it efficiently, which is about four times that of the main product (oxygen) [35]. The second scheme stores the surplus energy at the off-peak times in form of liquefied air which will be used to drive recovery system, this scheme considered as a standalone plant with no need for waste heat in recovery part. For these two schemes, the thermodynamic analysis was carried out using MATLAB code integrated with reference fluid thermodynamic and transport properties database (REFPROP) software [36, 37]. Fig.1 shows the daily power demand for a residential building with 50 accommodation units, used as case study for the proposed systems [38].

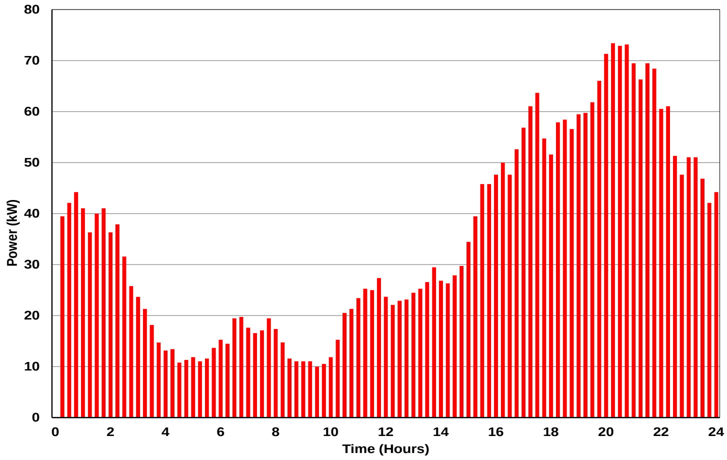

Fig1 Load profile for the residential 50 units building in a winter day [38]

2. Proposed schemes

The proposed schemes aim to use stored energy in LAir/LN2 to provide power for a residential building. The systems consists of two main cycles; the first one is a liquefaction cycle which produces the cryogen by compression and cooling process at off-peak times to store energy in LAir/LN2 then, in the recovery cycle in which the LAir/LN2 from liquefaction cycle is evaporated and superheated, the stored energy is extracted by the expansion process at peak times. The recovery cycle is a combined cycle which integrates open and closed Rankine cycles to recover the maximum amount of energy from the cryogen. By using storage systems the integration between liquefaction cycle and recovery cycle stores hot energy from compression in liquefaction mode to reuse it in recovery mode and stores coldness from LAir/LN2 in recovery mode to reuse it in liquefaction mode. These schemes were modeled using MATLAB integrated with REFPROP software to investigate the system performance of various design conditions.

2.1 Scheme 1

The layout of this scheme in which an oxygen liquefaction cycle is integrated with combined cycle recovery system as presented in Fig.2 and Fig.3. In the liquefaction cycle, the environmental air is compressed to high required pressure by two compressors (Comp1 and Comp2). The heat generated from compression process extracted by the heat exchangers (HX1 and HX2) in order to store it in the hot storage system. The pressure is divided between the two compressors to minimize compressor input work and achieve maximum storage capacity. Then the compressed air cools further through a heat exchanger (HX3) by cold air from the separator unit, cold storage system, and external cooling system. The cold air passes through a cryo-turbine (Turb1) to decrease its temperature until reaches to a two-phase liquid-vapor mixture at the same time generating power. The liquefied air separates to its main components liquid oxygen and LN2 which will store at temperature 77K at atmospheric pressure to be used in the recovery system. All the above process is running during the off-peak times to produce liquid oxygen and LN2 which is the main duty of the current oxygen liquefaction plants.

At the peak times, the stored LN2 is used to drive the recovery cycle where LN2 is pumped to a heat exchanger (HX4) to extract its coldness which stores in cold storage system to reuse in liquefaction plant mode while LN2 evaporates and superheats. The nitrogen then flows through the heat exchanger (HX5) where it is further heated by the hot storage system. The nitrogen then expands through three turbines (Turb2, Turb3, Turb4) with inter-heating heat exchangers (HX6, HX7), again this heating is done by the hot storage system. The nitrogen emerging from the final turbine in the recovery cycle and the output from inter-heating exchangers are still at a high temperature so they use to drive closed Rankine cycle via a heat exchanger (HX8) to recover waste heat by reaching to the room temperature. In the closed Rankine cycle, many working fluids have been tested such as (R125, R13, R115, R116, R142b, R143a) where the working fluid R143a gives the better performance due to it is boiling temperature and other properties. The liquid R143a is pressurized by (pump2) and then flows into a heat exchanger (HX8) where it will evaporate and superheat by waste heat from open Rankine cycle. After heating, the R143a gasses generate output work through expanding in expander (Turb5) to flows then through the condenser (HX9).

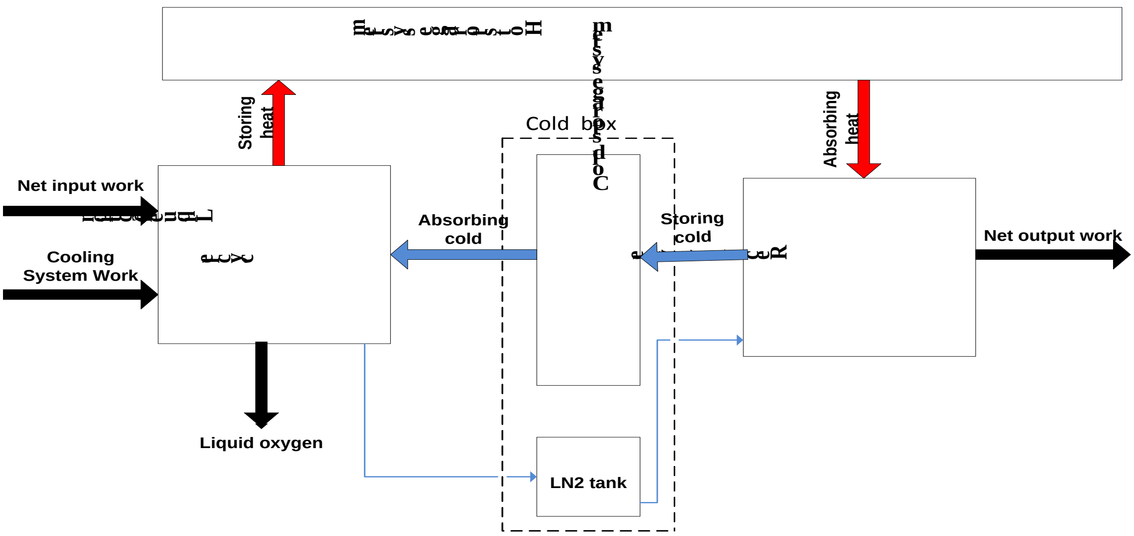

Fig2 Scheme1 energy and work flow block diagram

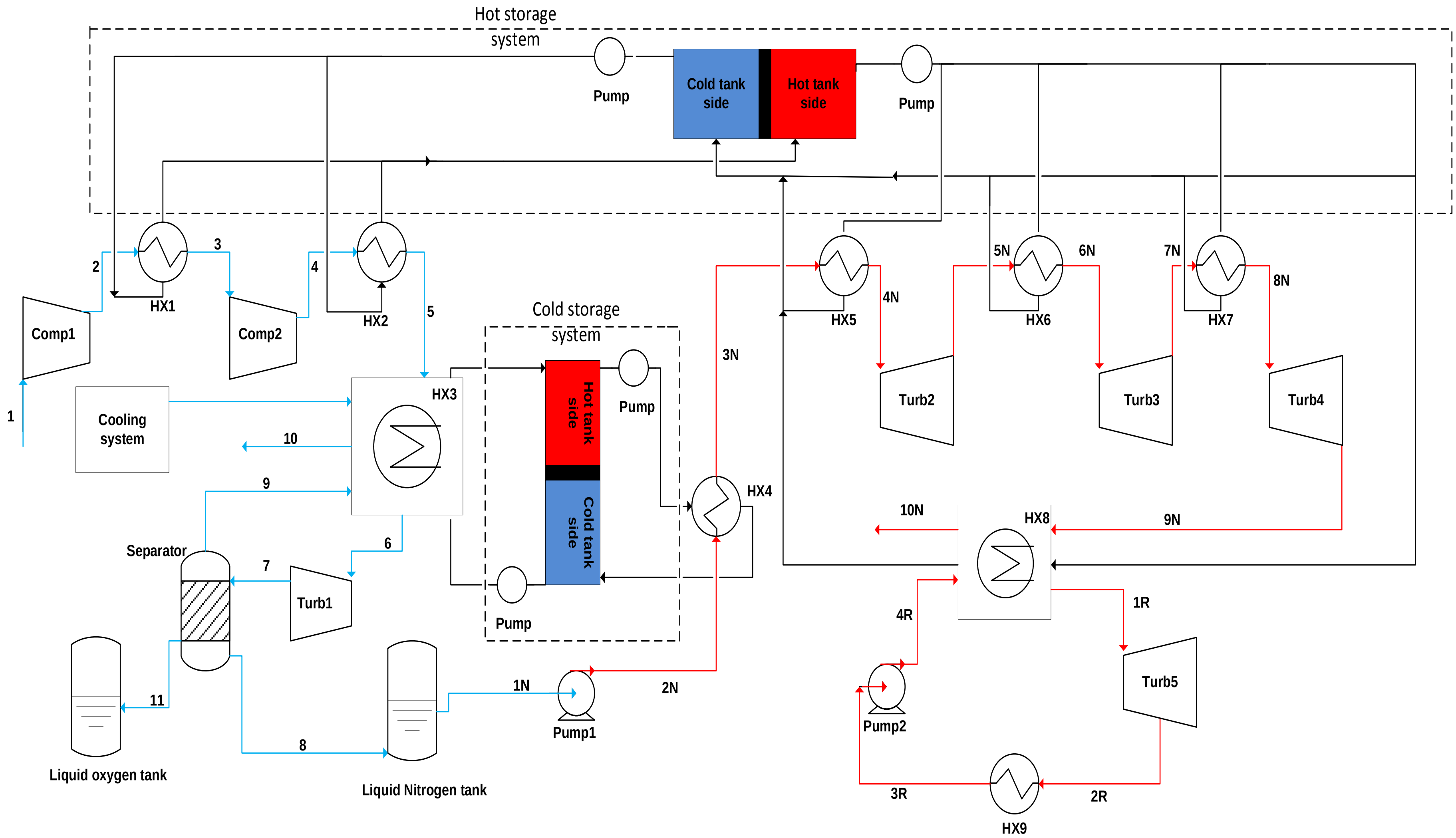

Fig3 Scheme1 liquid nitrogen energy storage plant layout

2.2 Scheme 2

The layout of this scheme which integrates the air liquefaction cycle with a combined recovery cycle is presented in Fig.4 and Fig.5. The liquefaction cycle in scheme 2 is the same in scheme1 where the air is compressed by two compressors (Comp1 and Comp2) with intercooling heat exchangers (HX1 and HX2). The compressed air is then cooled through a heat exchanger (HX3) by cold air from the separator and the cold storage system, then flows into cryo-turbine (Turb1) to expand to a two-phase liquid-vapor mixture which is separated to liquid air, and cold air. So the difference in liquefaction part between scheme1 and scheme 2 is, no need to use external cooling in heat exchanger (HX3) and also the liquid air will not separate into two parts (nitrogen and oxygen). The liquid air is stored in a tank at temperature 78K and atmospheric pressure for use in the recovery cycle. In recovery cycle, the liquid air is pumped from its tank to the required pressure and super-heated by the heat exchanger (HX4) to near room temperature. The air then flows through a heat exchanger (HX5) where it is further heated to expand through three turbines (Turb2, Turb3, Turb4) with inter-heating heat exchangers (HX6, HX7), where heating is done by the hot storage system. The air emerging from the final turbine in the recovery cycle and the inter-heating exchangers are still at a high temperature so they combine with Rankine cycle in heat exchanger (HX8) to recover waste heat. In the closed Rankine cycle, the liquid R143a is pressurized by (pump2) and then flows into a heat exchanger (HX8) for heating and vaporization by waste heat from open Rankine cycle. After heating, the R143a gasses generate output work through expanding in expander (Turb5) to flows then through the condenser (HX9). The difference in recovery part between scheme1 and scheme 2 is the air fraction flows in recovery cycle scheme2 are higher than that of the nitrogen in scheme 1 and that will reflect on cycle generation performance. The scheme 2 uses liquid air as energy storage media and generates power from it in recovery part without using any waste heat from an industrial plant or other sources so this scheme considers standalone storage power generation plant.

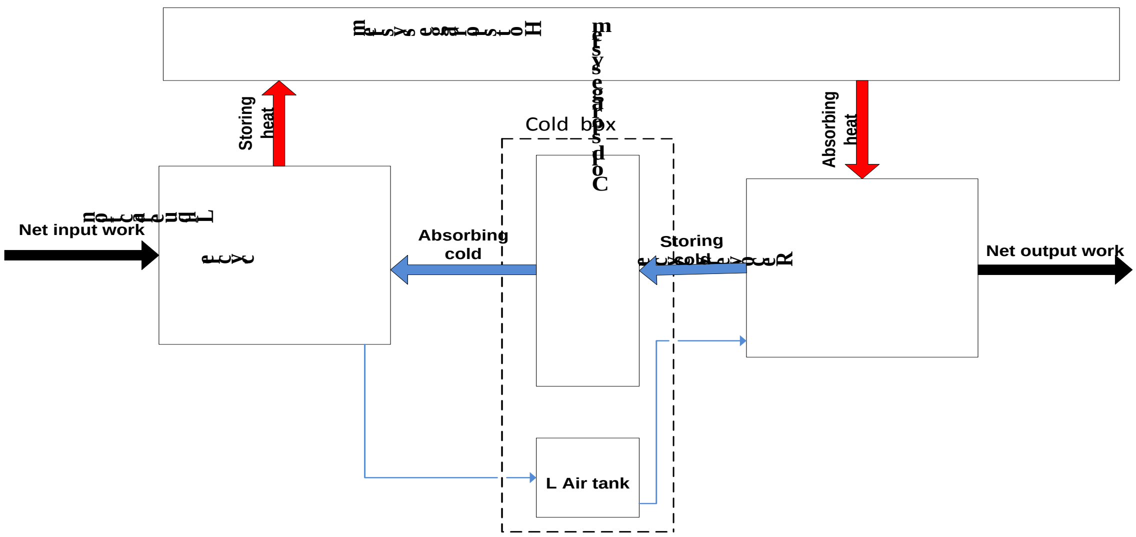

Fig4 Block diagram of scheme 2 energy and work flow

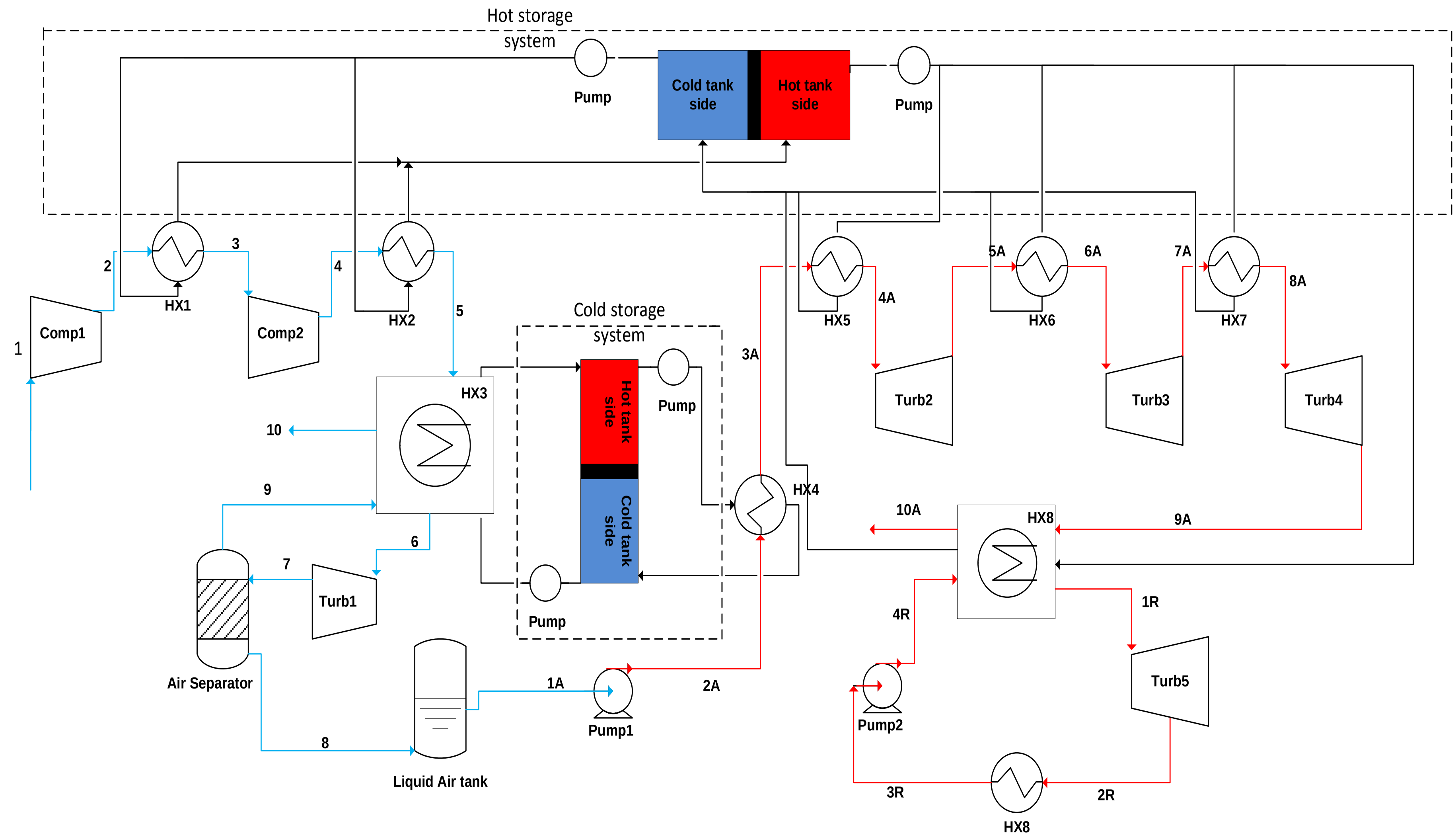

Fig5 Scheme 2 liquid air energy storage plant layout

3. Thermodynamic modeling of the storage power generation plant

To simplify the thermodynamic analysis general assumptions have been made as follows: (a) the LAir and LN2 stored at a saturated temperature and atmospheric pressure, (b) the environment temperature for the system is 298 K and the air and nitrogen leave the system at this temperature. (c) The pressure drops in the system pipes and heat exchanger are negligible, and heat exchange with surrounding is negligible. (d) The closed Rankine cycle condensation pressure is atmospheric pressure. It is worth to mention here that the cryogenic storage vessels heat loss depends on cryostat characteristics where different configurations have used such us complex super insulated or cooled radiation shields, which are in simple neck with polished copper screen configuration have ranged from 1.6 to 4.8 Watt/litter [39].

The governing equations for involved components in the two proposed schemes are listed below:

Compressor model:

$$h_{out} = h_{in} + \frac{h_{out,s} - h_{in}}{\eta_{s,comp}} \quad (1)$$

Turbine model:

$$h_{out} = h_{in} + \eta_{s,turb}\left(h_{out,s} - h_{in}\right) \quad (2)$$

Pump model:

$$h_{out} = h_{in} + \frac{h_{out,s} - h_{in}}{\eta_{s,pump}} \quad (3)$$

Heat exchanger model:

$$\dot{m}_{hot}\left(h_{hot,in} - h_{hot,out}\right) = \dot{m}_{cold}\left(h_{cold,out} - h_{cold,in}\right) \quad (4)$$

Separator model:

$$h_{in,gas} = Y * h_{out,liquid} + (1 - Y) * h_{out,gas} \quad (5)$$

$$Y = \frac{\dot{m}_{out,liquid}}{\dot{m}_{in,gas}} \quad (6)$$

The pressure ratio in liquefaction process is:

$$Pr = \frac{P4}{P1} \quad (7)$$

The specific work input for liquefaction cycle scheme1 is:

$$W_{net,input} = W_{Comp1} + W_{Comp2} + W_{CS} - W_{Turb1} \quad (8)$$

The specific work input for liquefaction cycle scheme2 is:

$$W_{net,input} = W_{Comp1} + W_{Comp2} - W_{Turb1} \quad (9)$$

The specific work output for combined recovery cycle scheme1 and scheme2 is:

$$W_{net,output} = W_{Turb2} + W_{Turb3} + W_{Turb4} + W_{Turb5} - W_{Pump1} - W_{Pump2} \quad (10)$$

The round trip efficiency of the overall plant is [22]:

$$\eta_{round\ trip} = Y * \frac{W_{net,output}}{W_{net,input}} \quad (11)$$

Where the (h_{in}) is inlet specific enthalpy, (h_{out}) is outlet specific actual enthalpy, ($h_{out,s}$) is outlet isentropic specific enthalpy after compressors, turbines, and pumps, ($\eta_{s,comp}$), ($\eta_{s,turb}$) and ($\eta_{s,pump}$) are isentropic compressors, turbines, and pumps efficiencies, ($h_{hot,in}$), ($h_{hot,out}$) are inlet and outlet specific enthalpies for heat exchanger hot stream, ($h_{cold,in}$), ($h_{cold,out}$) are inlet and outlet specific enthalpies for heat exchanger cold stream, ($\dot{m}_{hot}$), ($\dot{m}_{cold}$) are mass flow rate for hot and cold streams, ($h_{in,gas}$), is inlet specific enthalpy inter to the cryo-turbine, ($h_{out,gas}$) is outlet specific enthalpy for separator gas stream, ($h_{out,liquid}$) is outlet liquid enthalpy from separate, ($\dot{m}_{in,gas}$), ($\dot{m}_{out,liquid}$) are mass flow rate for gas and liquid in separator component, ($P1$), ($P4$) are the inlet and out let pressure to compressors train, (W_{Comp1}), (W_{Comp2}) are specific work input to the compressors, (W_{Pump1}), (W_{Pump2}) are specific work input to the pumps, (W_{CS}) is specific work input to the cooling system, (W_{Turb1}) to (W_{Turb5}) are specific work extract from the recovery cycle turbines, ($W_{net,output}$) is specific total network extract from the cycle, ($\eta_{round\ trip}$) cycle round trip efficiency of the overall plant. The system's performance is assessed in terms of their work output, work input, liquid yield, and round trip efficiency. In Eq.(6) the liquid yield Y has been presented, which is the ratio of mass of (L Air or LN2) created by liquefaction to the mass of air drawn by the two stages compressors; in the Eq.(11) the liquid yield becomes part of round trip efficiency which means that this parameter becomes one of the key performance parameters in these plants. The Eq. (8, 9) clarifies the difference between the input works to liquefaction cycle in two schemes wherein scheme1 the cooling system work will add to input work. Also the Eq. (10) shows how the output work could decrease when pumps work increase.

4. Results and discussion

The proposed plants aim to use LN2 or LAir to generate power for residential building application and to produce oxygen in scheme1, where the liquid nitrogen or liquid air will be the main energy source leading to zero environmental damage. The results of the simulations for the two schemes will be shown in this section as a parametric study of selected key performance parameters with the reference plant design parameters given in the Table2. The results of these parametric studies are shown in Figs. 6- 15. Liquefaction process depends mainly on two process compression and cooling respectively for the first process which is done by the two-stage intercooled compressor in order to reduce required work so the compression ratio has a significant effect on system performance. Fig.6 shows the effect of varying liquefaction compression ratio on plant round trip efficiency and liquid yield (Y) of two schemes. In both schemes, the round trip efficiency decreases when the compression ratio increases while the liquid yield increases as reported by [40]. The compression ratio 40 gives maximum round trip efficiency which is for scheme1 Fig.6 A is 61.32% while in scheme2 Fig.6 B is 81.02%; according to Eq.(11), the increase in input compressor work is higher than the increasing in liquid yield leading to a decrease in round trip efficiency.

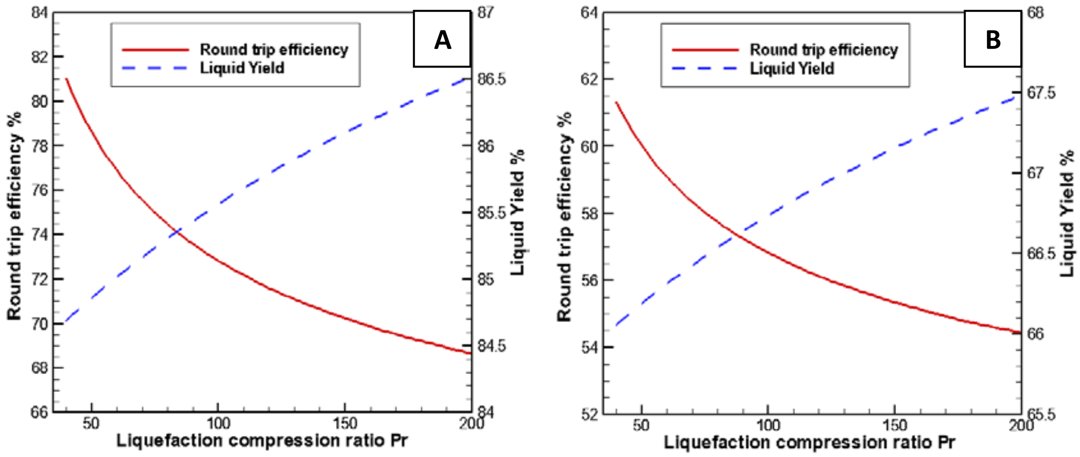

Fig 6 Effect of liquefaction compression ratio (Pr) on the round trip efficiency and liquid yield: A Scheme1, B Scheme2

Table 2
The investigated ranges and reference values of input parameters

Parameters	Input range	Default Value	Units
Environment temperature	298	298	K
Environment pressure	100	100	kPa
Liquid air storage pressure	100	100	kPa
HX pinch-point	5	5	K
Liquefaction compression ratio (Pr)	40-200	40	-
Inlet cryo-turbine temperature (T6)	91-120	96	K
Recovery Turbine expansion ratio		4	-
efficiency of air compressors	70%-100%	85%	-
efficiency of cryo-turbine	60%-100%	75%	-
efficiency of cryogenic pump	75%	75%	-
efficiency of recovery turbines	70%-100%	85%	
Rankine inlet turbine pressure	200-2000	1000	kPa

The cooling is the second step in a liquefication process where the compressed air is cooled in a cryogenic heat exchanger to the temperature T6, which is the inlet temperature to cryo-turbine, so this temperature considers as one of the key parameters in the current study because it affects significantly on liquid yield and overall efficiency as shown in Fig.7. It is clearly seen that in both schemes an increase in this temperature will decrease the round trip efficiency and liquid yield. According to Eq.(11), the decrease in liquid yield leads to decrease in round trip efficiency also less yield mean less working fluid in the recovery cycle which lead to less output work. The maximum round trip efficiency occurs at the minimum achievable temperature 91K where for scheme1 Fig.7A it is 63.27% while in scheme2 case Fig.7B it is 84.15 %. Figs. (8, 9) show the effect of the variation of the liquefaction compression ratio and inlet cryo-turbine temperature on net output power for two schemes. Fig.8 shows that increasing compression ratio will increase net output power, the pressure increase in points 2 and 4 in Figs 3 and 5 leads to increases the enthalpies at these points and according to Eq. (4) the heat extracted by heat exchangers HX1 and HX2 increase by consequent that will add more power for recovery cycle. Fig.9 shows increase inlet cryo-turbine temperature will decrease net output power, increase the temperature in point 6 in Figs 3 and 5 leads to increase the enthalpy at this point which is the output from heat exchanger HX3 so according to Eq. (4) the heat extract decrease by consequent that will lead to adding less power for recovery cycle.

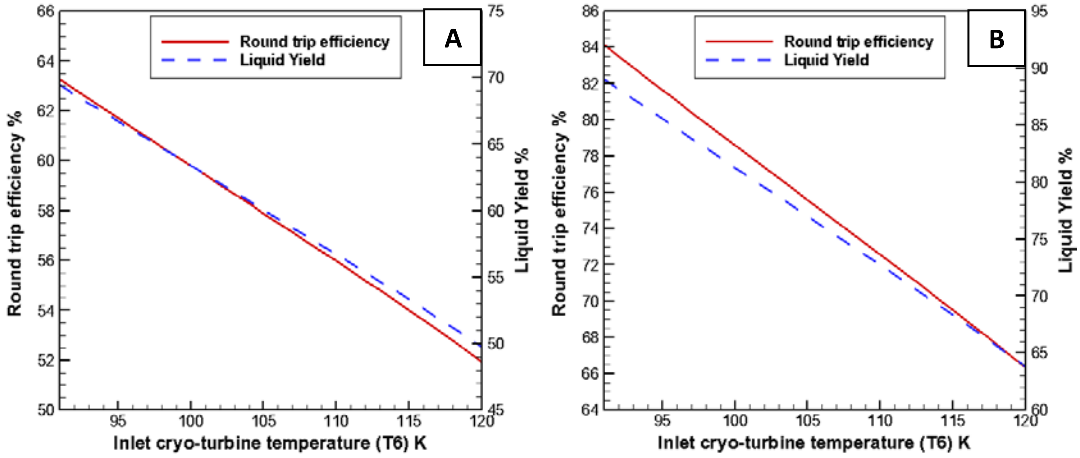

Fig7 Effect of the Inlet cryo-turbine temperature (T6) on the round trip efficiency and liquid yield: A Scheme1, B Scheme2

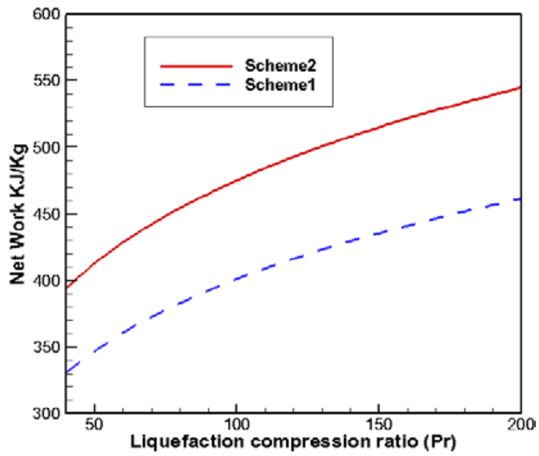

Fig8 Effect of liquefaction compression ratio (Pr) on net work output for both schemes

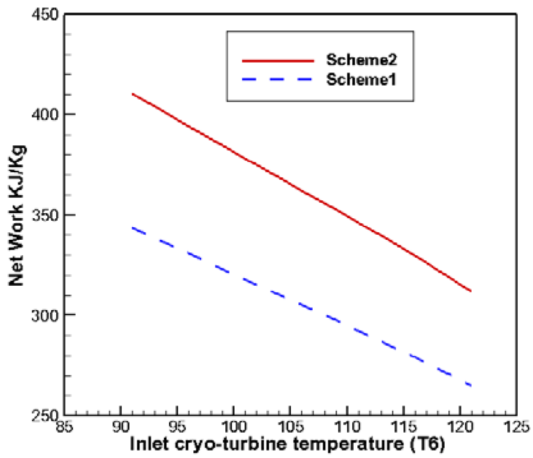

Fig9 Effect of Inlet cryo-turbine temperature (T6) on net work output for both schemes

The overall efficiency also affected by the compressors input power which depends on their pressure ratio and their isentropic efficiencies. The Fig.10 shows the effect of compressors efficiency on cycle's round trip efficiencies and net specific output work where increases compressors efficiency increases the round trip efficiency, however, decreases the net specific work output. For each 10% increase in compressor efficiency enhances the round trip efficiency is 2.2% and 4.1% for scheme1 and scheme2 respectively, to reach best values at 84% to produce round trip efficiency 80.62% and work 397KJ/kg for the liquid air condition. The increasing compressors efficiencies as shown in Eq.(1) will decrease the enthalpies deference (h2,h3) and (h4,h5) in heat exchangers HX1 and HX2 which will lead to less heat stored and in turn decreases the net output work from the recovery cycle. On the other hand, the round trip efficiency will increase because of the decreasing in compressor input work.

The other critical component in the liquefaction cycle is cryo-turbine (Turb1) which is commonly used in the liquefaction of natural gas with high isentropic efficiencies [41]; Fig.11 presents the effect of cryo-turbine efficiency on plant round trip efficiency and net specific output work of two schemes. The increase in cryo-turbine efficiency increases the liquid yield which according to Eq. (11) increases the round trip efficiency. It is also seen that the effect of cryo-turbine efficiency becomes more significant with the high-pressure ratio (200) due to more yield and power produces by this turbine according to Eqs. (2, 11).

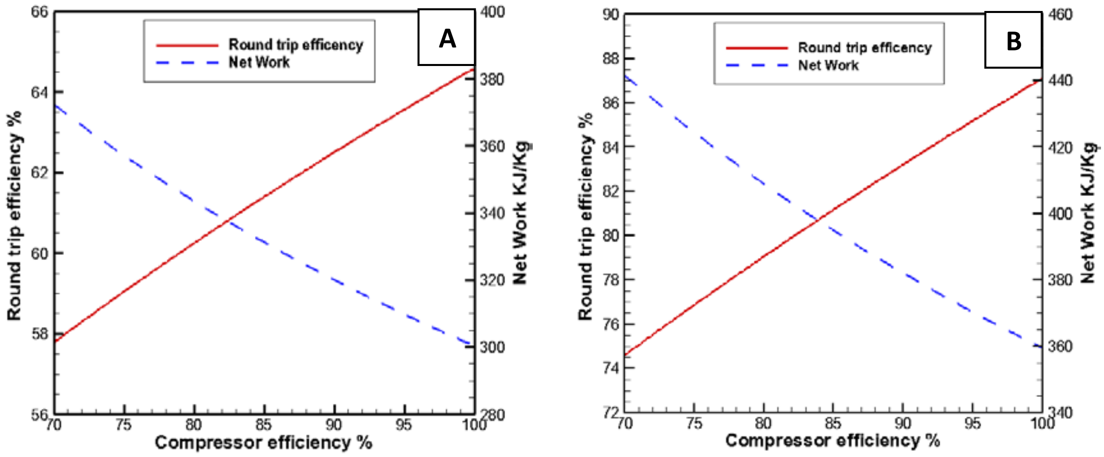

Fig10 Effect of the compressor efficiency on the round trip efficiency and net output power: A Scheme1, B Scheme2

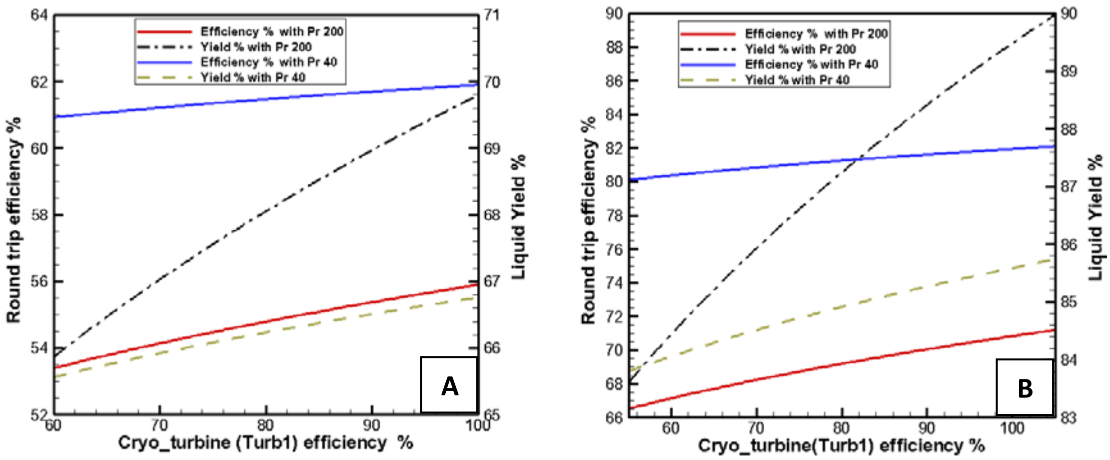

Fig11 Effect of the cryo-turbine efficiency on the round trip efficiency and liquid yield: A Scheme1, B Scheme2

Analysis of the energy recovery process is shown in Figs. (12-14) where Fig.12 shows the variation effect of recovery turbines expansion ratio on round trip efficiency and net specific output work of scheme1 and scheme2. Increasing the expansion ratio increases specific output power which in turn increases round trip efficiency and each 0.5 increasing in expansion ratio will increase round trip efficiency by 4.28% for scheme1 and 6.76% for scheme2. For the same reason, an increase in recovery turbines efficiencies will increase round trip efficiency as shown in Fig.13 and Eqs.(2, 10, 11). For the closed Rankine cycle increasing the inlet pressure turbine pressure (P1R) leads to increase the output power from Turb5 in Figs 3 and 5 which in turn increases Rankine cycle efficiency and round trip efficiency for the system where at the pressure 2MPa the Rankine efficiency reaches to 20.7% and round-trip efficiencies 62.5% and 81.9% for scheme1 and 2 respectively as shown in Fig14.

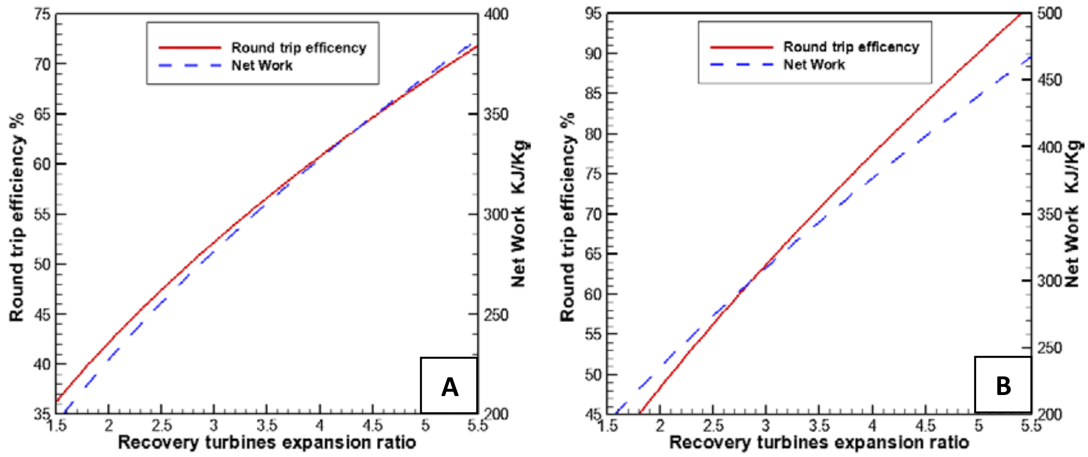

Fig12 Effect recovery turbine expansion ratio on round trip efficiency and network output: A Scheme1, B Scheme2

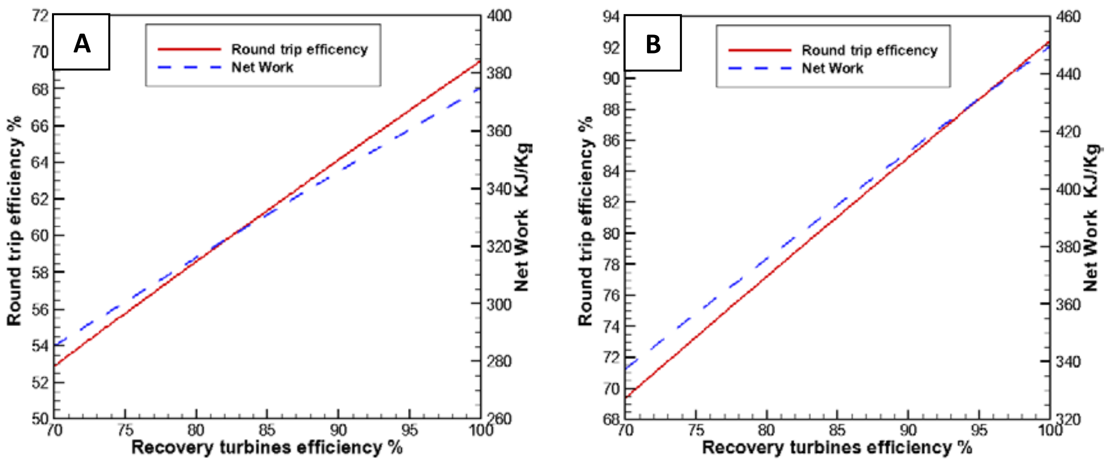

Fig13 Effect recovery turbine efficiency on round trip efficiency and network output: A Scheme1, B Scheme2

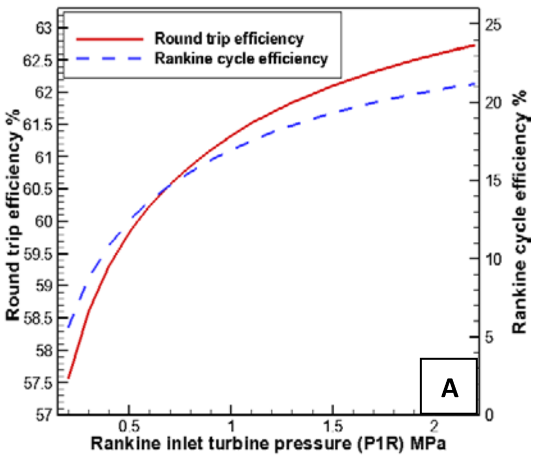

Fig14 Effect Rankine inlet pressure (P1R) on round trip efficiency and Rankine cycle efficiency: A Scheme1, B Scheme2

The typical residential building mentioned in the introduction was used as a case study to determine the required air mass of the system to achieve the maximum output specific work. Fig.15A shows the mass flow rate of air in the liquefaction stage, nitrogen mass flow rate which drives recovery part and oxygen production mass flow rate. Fig.15B shows the mass flow rate of air in the liquefaction stage and air mass flow rate which drives recovery part. The peak times maximum mass flow rate values in recovery part are scheme1 nitrogen 0.11 kg/s and scheme2 air 0.093 Kg/s, from designing turbines point of view these flow rates values indicate that these turbines classified as small-scale turbines.

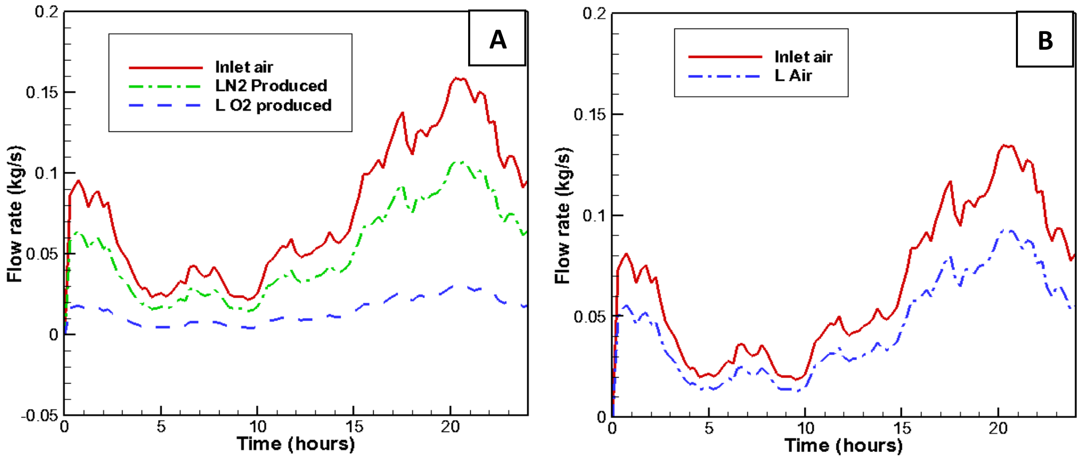

Fig15 The system mass flow rates for the residential 50 apartments building in a winter day

5. Conclusions

With the increased use of renewable energy sources and micro-grid networks, there is very limited work related to the development of small-scale cryogenic energy storage technology for local power generation applications. Cryogenic energy storage technology offers advantages of relatively large volumetric energy density and ease of storage. Thermodynamic modeling and parametric analysis of a novel power cycle that integrates air liquefaction plant, cryogen storage systems and a combined direct expansion with closed Rankine power recovery system using two cryogens, liquid nitrogen, and liquid air has been carried out. This cycle is a part of a micro-grid system that provides electricity for a typical 50 unit of a residential building at peak times while it uses either renewable energy sources or surplus electricity at the off-peak times. A major advantage of this cycle is that heat rejected from the liquefaction process will be stored and then reuse it as an input to the power recovery subsystem to improve the round trip efficiency. Also, cold is released by the power recovery subsystem, stored, and then reused as an input to the liquefaction process to improve its efficiency. In this work, two cycle configurations were investigated namely scheme1 and scheme2; the first one modifies the existing oxygen liquefaction plant with a combined recovery cycle using surpluses LN2, while the second one stores the energy in form of liquefied air to be used for power generation and the results showed that:

- Scheme1 is suitable for producing oxygen and generating power by using off the shelf liquefaction technology with acceptable round trip efficiency reaching 63.27%, while the scheme2 can be used for generating power with higher round trip efficiency of 84.15%.
- The recovery turbine efficiency has a significant effect on the plant's performance where each 10% increase in turbine efficiency will increase the round trip efficiency by 5.5% for scheme1 and 7.7% for scheme2.
- Combining closed Rankine cycle with the recovery process enhances the plant's overall performance where each 5.1% increase in Rankine cycle efficiency will increase the round trip efficiency by 1.8%
- Peak time's cryogen flow rates at high specific work output are 0.11 kg/s nitrogen in scheme1 and 0.093 kg/s air in scheme2 highlighting the potential of using small-scale turbines in residential building power generation and energy storage applications.

Nomenclature

Symbols		Subscript	
h	Inlet specific enthalpy (J/kg)	in	Inlet state to the component
$\eta_{s,comp}$	Isentropic compressors efficiency	out	Outlet state to the component
$\eta_{s,pump}$	Isentropic pump efficiency	s	Isentropic process
$\eta_{s,turb}$	Isentropic turbine efficiency	hot	Hot stream
$\dot{m}$	Mass flow rate (kg/s)	$cold$	Cold stream
Pr	Pressure ratio (-)	CS	Cooling system
Y	Liquid yield (-)	Acronyms	
W	Specific work (J/kg)	CAES	compressed air energy storage
$\eta_{round\ trip}$	Round trip efficiency	CES	cryogenic energy storage
		PSH	pumped storage hydropower
		LASE	liquid air storage energy
		LNG	Liquid natural gas
		LN2	Liquid nitrogen
		LAir	Liquid air

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