An evaluation of high-temperature energy storage with closed gas turbine discharge cycle

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***Abstract*—This paper presents research on improving the efficiency of air-based pumped heat energy storage. We used a simulation tool to model the charging and discharging cycles of the reference case. The discharge cycle of the baseline system is inefficient because it loses a lot of low-potential residual heat. Organic Rankine cycle is the technology used to convert this thermal energy into electricity. Adding it as a bottoming cycle increased the generated power by 6% and the round-trip efficiency by 5.6%.**

***Keywords*— energy storage, molten salts, air Bryton Cycle, organic Rankine cycle, round-trip efficiency**

I. Introduction

Pumped Heat Energy Storage (PHES) is a form of energy storage that uses heat pumps and heat engines to store and release energy. It operates by converting electricity into heat, storing that heat, and later converting it back into electricity when needed. This technology is being explored as a flexible, efficient, and scalable way to store renewable energy like solar and wind power [1]. As of now, PHES is still in the early stages of commercialization and has not yet been widely deployed on a commercial scale. However, several companies and research institutions are actively working to develop and bring PHES technologies to market [2]. Air is a commonly used working fluid in some PHES systems, particularly those employing a Brayton cycle. Inert gases like argon or nitrogen are also used in some configurations, especially for systems that operate at higher temperatures. These gases are stable, non-reactive, and readily available, making them suitable for heat pump and heat engine cycles. However, most PHES using inert gases are still at the proof-of-concept stage [3].

The innovative PHES that offers a promising, scalable, and sustainable solution for long-duration energy storage is presented in [4]. Its use of low-cost materials like air and molten salts, flexibility in siting, and expected long lifespan make it a strong candidate for future energy storage needs. However, the lower efficiency of the system and the early stage of commercialization present challenges that need to be addressed as it moves towards large-scale deployment. This paper presents the results of research efforts focused on improving the round-trip efficiency of this otherwise promising concept, hereafter called Baseline.

II. A comprehensive analysis of the Baseline PHES

A. General description

As with all PHESs, the one presented in reference [4] is comprised of three phases: Energy Conversion (Charging Phase), Energy Storage and Energy Recovery (Discharging Phase). During times of excess electricity (such as when renewable energy is abundant), a heat pump is used to convert electrical energy into thermal energy. The heat pump moves heat from a cold reservoir to a hot reservoir, storing energy in the form of temperature differences. The hot reservoir is a well-insulated container storing the heat at a high temperature using molten salts. The cold reservoir stores the other side of the temperature gradient, at very low temperatures, using chilled fluid. This consequence represents the phase of the process during which the charge is applied. The heat and cold can be stored for extended periods with minimal energy loss, depending on the insulation quality and the temperature difference maintained.

When electricity is needed, the process is reversed. A heat engine extracts the stored heat from the hot reservoir and cold from the cold reservoir, converting the temperature difference back into electricity. This is achieved through the utilization of a closed-loop Brayton cycle, which facilitates the conversion of thermal energy into mechanical work, thereby driving a generator. This consequence represents the discharging phase.

B. An evaluation of the charging phase

The charging phase involves converting electrical energy into thermal energy, which is stored as heat and cold in two set of separate tanks. This phase is essential for preparing the system to store energy until it's needed, when it will then be converted back into electricity during the discharge phase. Fig. 1 illustrates the configuration of the charge cycle.

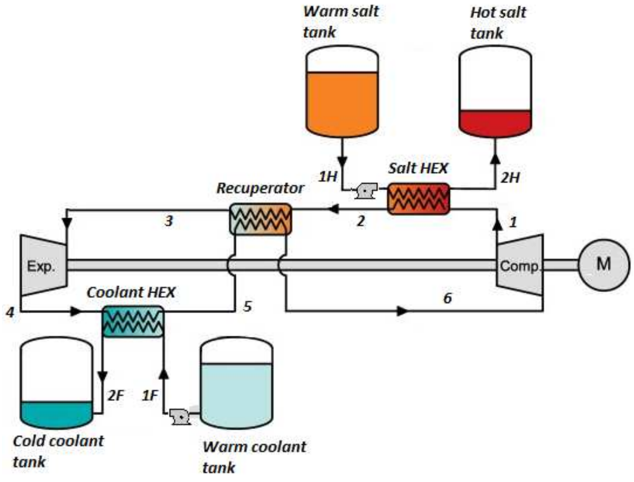

Fig. 1. Principal diagram of the charging cycle

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The incoming electricity is used to power the compressor of a heat pump. The compression causes the fluid to heat up to very high temperatures. The heat generated during this compression is transferred to the hot reservoir. The heat transfer fluid is molten salts. The molten salts, originating from the warm salt tank, are pumped into the heated Salt Heat Exchanger, where they are subsequently routed to the hot salt tank. The molten salt can store high amounts of heat and maintain its temperature for long durations with minimal heat loss. The temperature of the warm salt is approximately 270°C, while that of the hot salt is nominally 565°C.

Once the air has released heat to the hot reservoir, it expands and cools. Prior to this, it passes through recuperation, whereby heat transfer occurs between the fluid and the air entering the compressor. The cooling effect produced by the expansion in the expander is utilized to reduce the temperature of the air to sub-zero levels. After the expansion the air is routed to coolant heat exchanger. The coolant is not exactly specified in [4]. The cold and warm coolant tanks are filled with antifreeze, which is a solution of water and ethylene glycol that prevents the formation of ice in the system. In a coolant heat exchanger, the coolant serves to absorb heat from the air in order to facilitate the transfer of heat from one medium to another. The coolant, originating from the worm coolant tank, is pumped into the cooled Heat Exchanger, where it is subsequently routed to the cold coolant tank. The temperature of the warm coolant is approximately 25°C, while that of the cold coolant is nominally -60°C.

As stated above, the coolant is not specified in reference [4]. Our study suggests using a specific type of heat transfer fluid for this purpose, namely Duratherm XLT. The oil manufacturer's recommended temperature range for this application is between -67.78°C and 167°C. This is in good agreement with the range used in baseline study.

The baseline charge cycle configuration described above was modelled in detail using a thermodynamic performance model in order to simulate a steady-state condition. For this purpose, we used the Thermoflex® software product, commercially developed and offered by the software company Thermoflow Inc. [5]. The Thermoflex computer program is used in a wide range of research and engineering applications. The model calculates the mass flow and temperature of the working fluids, taking into account both the ambient conditions and those of the storage media..

Table 1 presents a comparison between the computational results obtained by our research team (column D) and those of the baseline case (column BL).

In both cases, the compressors and expanders exhibit identical pressure ratios and expansion ratios, with values of 4.592 and 4.331, respectively. The comparison demonstrates a favorable alignment between the two cases, with the exclusion of coolant flows. The discrepancy in observed flow rates can be attributed to the distinct physical properties of the materials utilized. Furthermore, the favorable correlation between the models is evident also with regard to the heat exchanged in the main cycle's heat exchangers, as illustrated in Table 2.

TABLE I. CHARGE CYCLE AIR AND HEAT TRANSFER FLUID PARAMETERS

№ of stream	Flow		Temperature	
	(kg/s)		(^{0}C)	
	BL	D	BL	D
1	766.1	766.1	576.70	756.6
2	766.1	766.1	279.20	272.9
3	766.1	766.1	36.14	31.15
4	766.1	766.1	-66.39	-71.30
5	766.1	766.1	17.72	17.22
6	766.1	766.1	267.50	267.50
1H	553.02	553.00	270	270.00
2H	553.02	553.00	564.99	565.00
1F	311.77	400.00	25.19	25.19
2F	311.77	400.00	-59.87	-62.18

TABLE II. HEAT EXCHANGE IN THE MAIN HEAT EXCHANGER

Heat exchanger	BL	D
Salt HEX, MW	246 855	247 892
Recuperator, kW	196 153	197 444
Coolant HEX, kW	66 509	68 593

C. C. An evaluation of the discharging phase

During the discharging phase, the system taps into the thermal energy that was stored in the hot tanks (high-temperature molten salt) and the cold tank. The energy stored as the temperature difference between these two reservoirs is the driving force that will generate electricity. Fig. 2 illustrates the configuration of the discharge cycle.

The core component in the discharging phase is a heat engine, which operates in reverse of the charging phase's heat pump. The heat engine extracts heat from the hot tank and cold from the cold tank to generate mechanical energy, which is then converted into electricity. This process is based on a closed loop Brayton thermodynamic cycle, where a working fluid is expanded in gas turbine to produce power.

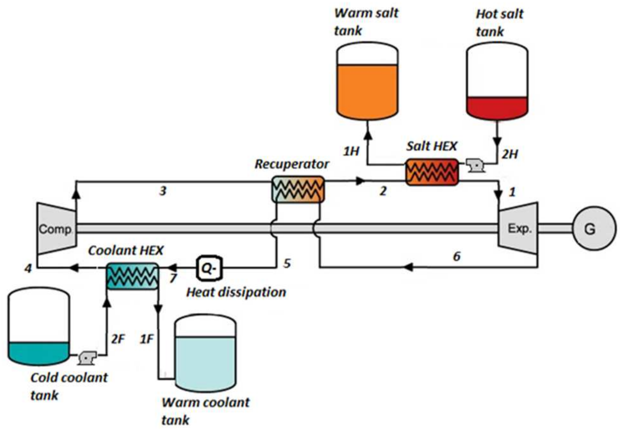

Fig. 2. Principal diagram of the discharging cycle

The Brayton cycle is a thermodynamic process that utilizes a working fluid to produce mechanical work, which is then used to drive a generator. The process involves compressing, heating, expanding and then cooling the working fluid in a series of steps. The air enters a compressor, where it is compressed to a higher pressure. This process naturally increases the temperature of the air, but it's still at a substantially lower temperature than the heat in the hot tank. The compressor is powered by the mechanical energy generated by the expander. Following the compression stage, the air is conveyed to the salt heat exchanger, passing through the recuperator. In the recuperator, the air is heated by the air that has exited the expander.

The molten salts, originating from the hot salt tank, are pumped into the Salt Heat Exchanger, where they are cooled and subsequently routed to the warm salt tank. The air absorbs heat in the salt HEX, increasing its temperature further. This stage replaces the fuel combustion in a conventional Brayton cycle and is what makes the system renewable and emission-free. The hot, high-pressure air is directed into a expander/turbine. As the fluid expands through the turbine, it generates mechanical work. The turbine drives a generator, converting the mechanical energy into electricity.

After passing through the turbine, the working fluid has released much of its energy and is at a lower pressure and temperature. However, the temperature remains sufficiently elevated to facilitate the heating of incoming air into the recuperator. Subsequently, the air is cooled in heat rejection heat exchanger where its heat is dissipating into the atmosphere. This heat dissipation is an avoidable one and it carries out a part of the heat dissipation to the cold reservoir that is necessary for a very thermodynamic cycle.

The coolant heat exchanger then cools the air further. For this purpose the coolant, originating from the cold coolant tank, is pumped into the coolant heat exchanger, where it is subsequently routed to the warm coolant tank.

The cooled, low-pressure working fluid is then reintroduced into the compressor, where it is compressed again to repeat the cycle. The compressor, turbine, heat exchangers, and reservoirs work together to create a closed-loop system.

The baseline discharge cycle configuration described above was modelled in detail using a thermodynamic performance model in order to simulate a steady-state condition. For this purpose, we used again the Thermoflex® software product. The model calculates the mass flow and temperature of the working fluids, taking into account both the ambient conditions and those of the storage media.

Table 3 presents a comparison between the computational results obtained by our research team (column D) and those of the baseline case (column BL).

In both cases, the compressors and expanders exhibit identical pressure ratios and expansion ratios, with values of 5.911 and 5.445, respectively. The comparison demonstrates a good alignment between the two cases, with the exclusion of coolant flows. As in the charging phase the discrepancy in observed flow rates can be attributed to the distinct physical properties of the materials utilized. As illustrated in Table 4, the correlation between the models is also evident in the heat exchange data for the primary cycle's heat exchangers.

TABLE III. DISCHARGE CYCLE AIR AND HEAT TRANSFER FLUID STREAM PARAMETERS

№ of stream	Flow (kg/s)		Temperature (K)	
	BL	CS	BL	CS
1	762.21	762.20	556.10	556.10
2	762.21	762.20	258.73	258.70
3	762.21	762.20	114.65	114.70
4	762.21	762.20	-52.76	-52.75
5	762.21	762.20	121.64	125.20
6	762.21	762.20	269.48	272.50
7	762.21	762.20	32.19	32.16
1H	553.02	553.00	563.15	563.10
2H	553.02	553.00	271.50	270.90
1F	311.77	400	25.19	22.56
2F	311.77	400	-59.75	-59.75

TABLE IV. HEAT EXCHANGE IN THE MAIN HEAT EXCHANGER

Heat exchanger	BL	D
Salt HEX, MW	245 003	245 557
Recuperator, kW	115 706	115 209
Coolant HEX, kW	66 441	65 391
Heat dissipation, kW	69 289	71 712

The analysis carried out and the results obtained provide grounds to draw the following conclusions:

- The mathematical models constructed within this study provide a reasonable description of the baseline configuration of the energy storage system. It is evident that they can be employed to conduct reliable simulation analyses.
- A considerable quantity of residual heat is emitted and dissipated during the discharge cycle. The temperature of the air prior to the heat dissipation stage is sufficiently elevated to provide thermal energy for the bottoming energy conversion cycle.

III. PHES DISCHARGE WITH BOTTOMING ORGANIC RANKINE CYCLE

In this section, an attempt will be made to enhance the baseline discharge configuration performance through implementation of a bottoming energy conversion cycle. A proportion of the dissipated thermal energy will be employed for this purpose.

The conversion of low-temperature heat into useful energy can be accomplished through the utilization of a multitude of thermodynamic cycles, each of which is optimized for distinct types of heat sources and applications. The Organic Rankine Cycle (ORC) and Kalina Cycle are two of the most popular thermodynamic cycles for the recovery of industrial waste heat and the utilization of geothermal energy. The Organic

Rankine Cycle (ORC) has been selected for this study on the basis of its status as a mature technology.

The Organic Rankine Cycle (ORC) represents a variation of the traditional Rankine cycle, which is widely employed in power plants. The principal distinction between the two lies in the utilization of an organic working fluid with a lower boiling point (such as pentane, butane, or refrigerants) in place of water. This renders the ORC particularly well-suited to low-temperature heat sources. Fig. 3 illustrates the conventional process flow diagram of an ORC.

In this configuration, the recuperator's airflow 5 is directed to the ORC evaporator, where its heat is transferred to the organic working fluid. This causes the fluid to evaporate into a vapor phase. Consequently, the temperature of the air entering the heat dissipation stage is reduced, resulting in diminished heat losses. A high-pressure vapor is expanded through an expander, whereby mechanical work is generated. This is used to drive a generator and thereby produce electricity in addition to that produced by the topping Brayton cycle. Once the working fluid has exited the turbine, a portion of the residual heat within the fluid is captured in the recuperator. This heat is then conveyed to the working fluid prior to its entry into the evaporator. Subsequently, the vapor is cooled in a condenser, returning to a liquid state. Thereafter, a feed pump transports the condensed liquid back to the evaporator, where the cycle repeats.

The selection of the working fluid in an Organic Rankine Cycle (ORC) is of critical importance with regard to the efficiency, performance and operational safety of the system. It is essential that the thermodynamic properties of the working fluid correspond to the temperature and pressure ranges of the heat source and sink, while also ensuring compatibility with the system components. In accordance with the recommendations set forth in reference [6], the current study employs the use of refrigerant R245fa. It is widely used in waste heat recovery and geothermal applications.

The Brayton discharge cycle simulation model, as previously described, has been enhanced with the integration of bottoming ORC. The combined discharge cycle, which has been elaborated in this way, has been simulated with a set of evaporating pressures in the ORC evaporator. The highest power output for the ORC was achieved at an evaporating pressure of 8 bar.

Table 5 provides the fundamental data required to evaluate the baseline PHES with the configuration that incorporates ORC during the discharge phase. From this analysis, it

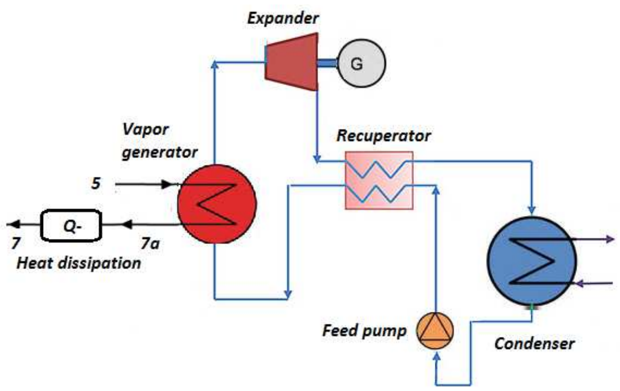

Fig. 3. Principal diagram of an ORC

becomes evident that the integration of ORC confers notable benefits.

The round-trip efficiency, as defined in the aforementioned text, is the ratio of the energy recovered from the PHES system during the discharge cycle to the total energy input during the charge cycle.

IV. Conclusions

This study has provided a comprehensive analysis of the pumped heat energy storage with air Brayton closed discharge cycle. In order to achieve this objective, both the charging and discharging cycles have been modelled using an appropriate simulation tool. The discharge cycle is distinguished by low efficiency, largely due to the dissipation of a considerable quantity of low-potential residual heat. Given its maturity, ORC has been selected as the technology for converting this thermal energy into electricity. In this combined configuration, the organic Rankine cycle functions as a bottoming cycle, while the air Brayton cycle operates as a topping cycle. The integrated operational approach resulted in a net increase in generated power by 6%, which also led to a 5.6% increase in round-trip efficiency.

TABLE V. Summary of the results C

Power and Energy, consumed/generated	BL	D
Compressor Shaft Power, MW	255.10	255.10
Gas Turbine Shaft Power, MW	70.67	70.67
Electric Motor Shaft Power, MW	184.43	184.43
Motor efficiency, %	97.34	97,34
Charge Power, kW	189.47	189.47
Charge Hours	10	10
Charge Energy, MWh	1894.71	1894.71
Compressor Shaft Power, MW	128.65	128.65
Gas Turbine Shaft Power, MW	231.62	231.62
Generator Shaft Power, MW	102.97	102.97
Generator efficiency, %	98.59	98.59
Bryton cycle Power, MW	101.52	101.52
ORC Turbine Shaft Power, MW	n.a.	5.98
ORC pumping power, MW	n.a.	0,30
Discharge Power, kW	101.52	107.21
Discharge Hours	10	10
Discharge Energy, MWh	1015.21	1072.05
Round-trip Efficiency, %	53.58	56.60

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