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Integrated High-Temperature Heat Pump and Thermal Energy Storage Laboratory Rig – Development of the validation plant and preliminary commissioning results

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Abstract

This work presents the design and preliminary commissioning results of an integrated heat upgrade system using a Stirling Cycle heat pump with two thermal energy storage units for industrial heat generation. Heat is upgraded to 200-250°C by the Stirling heat pump using a low temperature water vector as source. The heated stream is used for steam production, and for charging a thermal energy storage unit, which can also be discharged to continue with the steam production when it is too expensive to operate the heat pump due to higher electricity prices. The validation plant has two closed circuits filled with pressurized water, working as the heat transfer fluid. Both circuits are connected to the high-temperature heat pump. Additionally, two latent heat thermal energy storage units have been installed to increase the flexibility and overall performance of the system. One on the cold circuit, featuring a phase change material with a melting temperature of 75°C, enabling it to store waste heat when the heat pump is not operating. And the second, on the hot circuit, featuring a phase change material with a melting temperature of 227°C. Supporting equipment includes circulating water pumps, pressure vessels, electric heaters, control valves, temperature and pressure sensors, and the steam generator. The validation plant is designed to deliver about 200 kW of heat to the steam generator.

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Keywords: High-temperature heat pump; thermal energy storage; industrial decarbonization; Stirling cycle.

1. Introduction

High-temperature heat pumps and thermal energy storage are emerging technologies for decarbonizing process heat. The deployment of these technologies to deliver heat between 100 and 200°C could reduce the total CO_2 emission in the European industry to 26% [1]. Higher temperature applications in the range of 200-500°C need to be developed by experimental set-ups and validation experimental campaigns. A review of 34 technology suppliers in 2023 shows that available technologies with a TRL 9 correspond only to 40%, from which only 7 of these technologies can deliver heat above 150°C [2].

Previous experimental studies have investigated different cycle configurations using heat pumps. Liu et al built an experimental facility to recover heat from flue gases using an ammonia-water heat pump to produce steam at 154°C, reaching 70 kW of heating output, and a waste heat heating performance coefficient of 0.31 [3]. In another work, Ramirez et al presents a prototype of a high-temperature heat pump using R-1233zd(E) as refrigerant and a single-piston reciprocating compressor, achieving a sink temperature of 148.5°C with a maximum heat production of 66.1 kW by recovering heat at 80°C [4]. Other experimental studies have investigated heat pump/ORC test rigs for delivering heat at 120°C [5-6]. In this context, research efforts are being made to develop full-scale industrial heat pumps that upgrade waste heat to useful temperature levels between 130-160°C [7-8]. There is a clear gap of more validation experimental rigs above 160°C. This research presents an integrated system of a high temperature Stirling heat pump coupled with thermal energy storage units to effectively deliver process heat at 200°C in the context of SUSHEAT project [9].

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2. Methods

The development of the plant begins with the initial objective of the project: to design a system to deliver 200 kW of heat up to 250°C, as shown in Figure 1. At this stage, the position of the thermal energy storage units, heat transfer fluid, and auxiliary equipment is not yet defined. The system should be flexible enough to replicate different heat sources vectors, to store thermal energy through the latent heat thermal energy storage units, and to upgrade the heat in the range of 200-250°C.

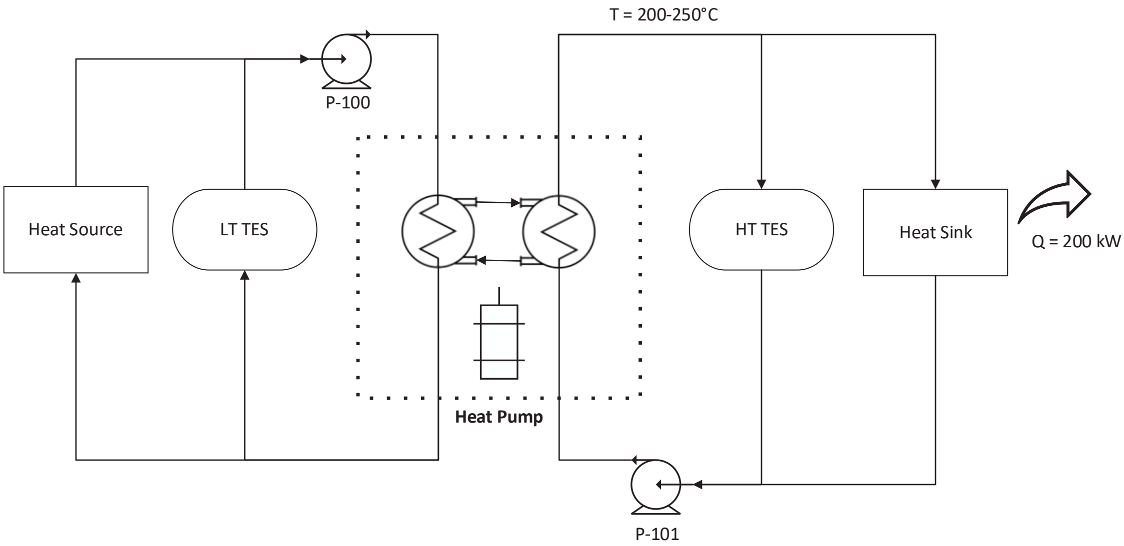

Figure 1. First idea of the validation plant.

2.1. Heat duty and boundary conditions

The boundaries of the system are determined by the general energy balance of the system. This includes the heat transfer fluid heat capacity and the heat pump operational boundaries. The Stirling-type heat pump is a double-acting gamma configuration machine that uses helium as refrigerant, which is non-toxic and does not have any global warming potential. Figure 2 shows a simplified schematic of the working mechanism: power piston (1), displacer piston (2), heat exchangers (3), and regenerator (4). During compression (upper heat exchanger), the gas releases heat to the heat transfer fluid that is moving through the shell, and helium is moving inside the tubes. During expansion, the gas absorbs heat from the heat transfer fluid in the same way. For this development, it is decided that the transfer fluid is to be pressurized water since it is a heat vector widely accepted by different industries (food, paper, chemicals).

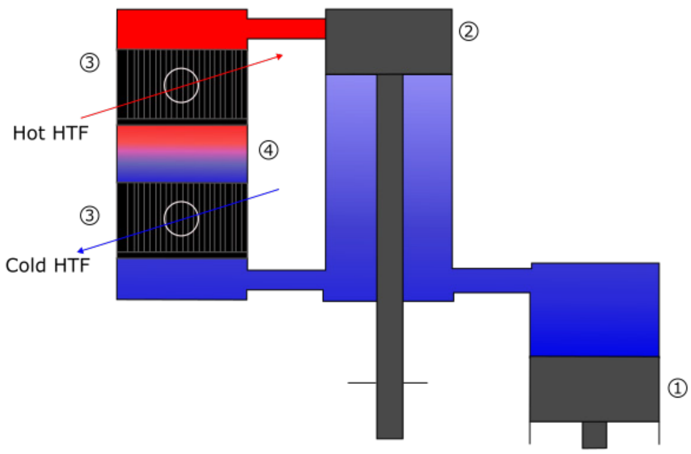

Figure 2. Simplified mechanism of the Stirling-type heat pump.

The main boundary conditions for the heat pump are volumetric flow rate and turbulence of water going through the heat exchangers, to guarantee a good heat transfer from the helium. The temperature glide of water is expected to be 3-5°C on both heat exchangers, and flow rate must be between 10-15 L/s. These characteristics will determine the available power that can be delivered to the system's heat sink downstream of the heat pump, since the circuit on the hot side of the heat pump is closed. This means that the water vector after the heat sink will come back to the heat pump to be heated again. The ideal situation is that the amount of heat delivered at the heat sink is close to the heat provided by the heat pump, avoiding major fluctuations in the system and leaving the heat pump to work in a steady state and therefore in a rather constant coefficient of performance (COP). With these premises, the energy balance can be done from the heat sink of the system backwards to the source, and it is expressed by Eq. (1).

$$\dot{Q}_{delivered} + \dot{Q}_{loss} = \dot{m}_{HTF} cp_{HTF} (T_2 - T_1) \quad (1)$$

where T_2, T_1 are temperatures of the HTF before and the heat sink respectively, and $\dot{Q}_{loss}$ are the segment thermal losses, determined by Eq (2)

$$\dot{Q}_{loss} = \frac{2\pi k D_i (T_2 - T_s)}{Ln\left(\frac{D_o}{D_i}\right)} \quad (2)$$

where T_s is the temperature on the outer diameter of the pipe. It is expected to be very similar to the temperature inside the pipe, since the system is thermally insulated. From this balance, it is also possible to calculate the pressure drop along the pipe segment. By knowing the mass flow of water, the Reynolds number can be calculated by

$$Re_D = \frac{\rho V D_i}{\mu_l} \quad (3)$$

where ρ is the density of the fluid, V is the velocity of the fluid, D_i is the internal pipe diameter, and μ_l is the dynamic viscosity of the fluid. For all the cases, Reynolds is > 200000; therefore, the flow achieves fully turbulent conditions. For each pump, the corresponding segment loss is determined by

$$\Delta P = \frac{f L \rho V^2}{2 D_i} \quad (4)$$

where L is the pipe length, and f is the friction factor for internal flow calculated by Petukhov [10] for a large Reynolds number range, and of the form

$$f = (0.790 ln Re_D - 1.64)^{-2} \quad (5)$$

for $3000 \leq Re_D \leq 5x10^6$. Then, the velocity of the water in the section is calculated by

$$V = \frac{\dot{v}}{\pi\left(\frac{D_i}{2}\right)^2} \quad (6)$$

where $\dot{v}$ is the volumetric flow rate in m^3/s, and the power of the water pump is calculated as

$$\dot{W}_p = \Delta P . \dot{v} \quad (7)$$

for the pressure vessel, the volume is calculated to have enough gas buffer volume to absorb fluctuations by

$$V_{pv} = 6\dot{m} \quad (8)$$

then for a 20% filled capacity, a vessel of 50 L is enough.

2.2. Thermal Energy Storage Units

For the thermal energy storage units, the selection of the phase change materials (PCMs) must correspond to stable materials that have a melting temperature around the desired temperature. In the case of the low temperature thermal energy storage (TES), a melting temperature between 40 and 80°C is reasonable since this range corresponds to typical temperatures of waste heat vectors. In the case of the high temperature TES, the melting temperature should be higher than the temperature of the heat delivered to the facility process. Process heat vectors are normally diathermic oil, steam, or hot air with temperatures around 175°C. Then, the sizing of the thermal energy storage units is based on the capacity of the tanks. This capacity can be equal to the peak demand or a percentage of the demand, depending on its magnitude. Then, the total energy is calculated and divided by the phase change enthalpy to obtain the amount of PCM needed. After knowing the volume of PCM needed, the corresponding tank can be designed for hosting the PCM and the corresponding heat exchanger that will be used for charge and discharge. Common latent heat TES units are made with seamless tubes or finned tubes, depending on the PCM material in the shell. Finally, after knowing the tank design, detailed piping and control valves can be placed to allow different operational modes such as charging and discharging.

3. Expected Results

In this section, the results of the finalized detailed engineering of the rig are presented with a respective 3D model. Additionally, expected results are mentioned since the experimental campaign will begin in the second half of 2026. The main KPIs of this research are:

- To investigate the performance of the system when upgrading the heat to 200-250°C.
- To achieve a COP of 2.8 using waste and ambient heat vectors as a source.
- To assess the charging and discharging efficiencies of the latent thermal energy storage units.
- To analyse experimental results for system up-scaling recommendations.
- Validate the technologies up to a TLR 5.

Figure 3 shows the finalized process flow diagram of the rig with the necessary equipment for the process. The heat pump is placed in the middle, where it is connected to the cold circuit (on the left) and to the hot circuit (on the right side). On the cold side, an immersion electric heater of 110 kW is used as a source of heat to replicate the different waste and ambient heat vectors. While on the hot side, a smaller electric heater of 20kW is placed before the heat pump to mark up the temperature of the water when it is returning after the steam generator, thus increasing the control of the temperature set points. On both sides, circulating water pumps and pressurized vessels are needed for adjusting the flow rate and keeping the pressure inside the circuits. The low temperature TES unit is placed after the electric heater so it can be charged. If the electric heater is off and the low-temperature TES is charged, it can be discharged to provide heat to the heat pump and increase its COP. The selected material for the low temperature TES is a bio-based commercially available PCM, Crodatherm 74 with a melting temperature of 75°C. The high temperature TES have been divided into two smaller tanks, and it features $NaNO_3$–KNO_3 (60–40 wt.%) with a melting temperature of 227°C as PCM. For the heat sink, a plate and shell steam generator was selected to produce steam at 2 bars (~120°C). Air coolers can also be used as a heat sink, but the area required is much larger and does not fit the space limitations in the laboratory. The energy balance has been solved for delivering 200kW of heat up to 250°C using source temperatures from 20 to 180°C. The COP of the heat pump has been calculated using interpolation from experimental data reported by [11]. Water flow rate has been determined to be between 10 and 12 L/s with a velocity of 4.5-5 m/s on both circuits. Pressure drop calculations and sizing of the pumps were developed using a DN50 pipe and valves, and pipelines have been placed to allow charge and discharge of the TES and for bypassing the heat pump. Figure 4 shows the 3D of the rig using solid modeling software. The area allocated for the validation plant is 6.5x4.35 m. The experimental campaign will begin in the summer of 2026, and data will be collected to validate the KPIs previously mentioned.

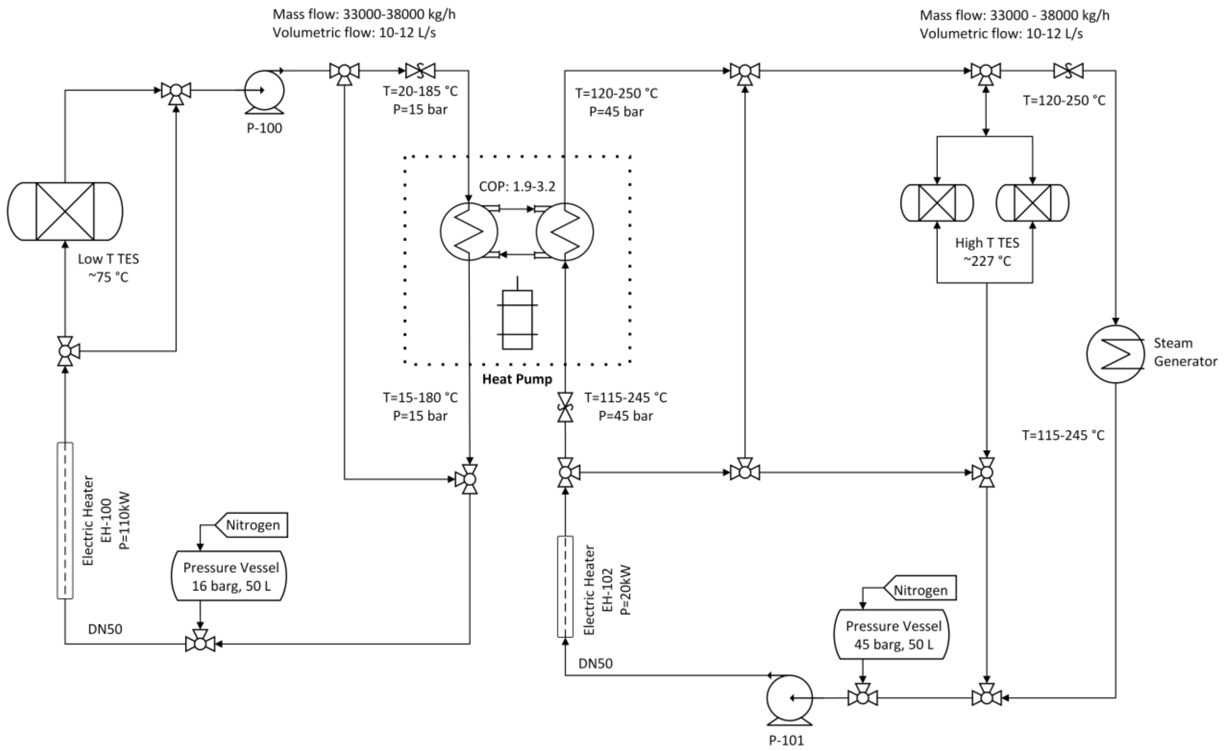

Figure 3. Finalized Process Flow Diagram (PFD) of the rig

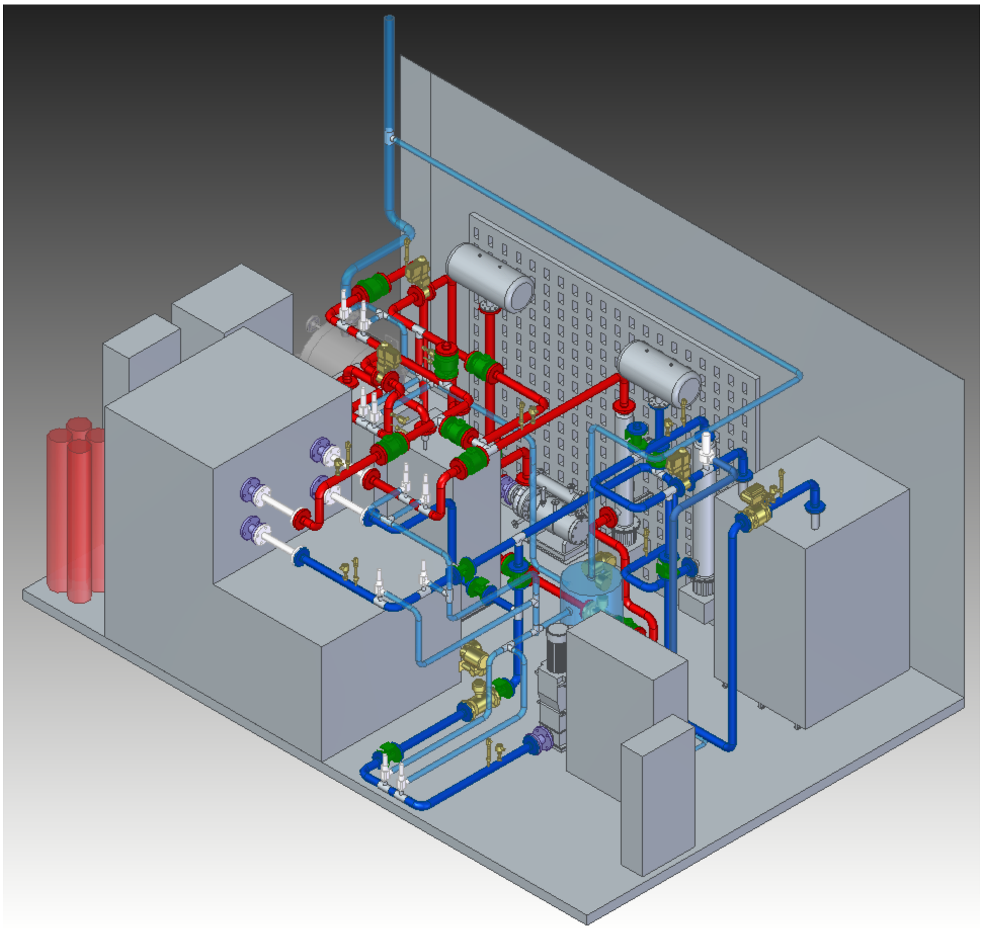

Figure 4. 3D model of the validation plant with real dimensions.

4. Conclusions

The current work has provided a methodology for designing an integrated system starting from the heat demand and the thermodynamic boundaries of the heat pump. The system features a novel high temperature Stirling heat pump coupled with an innovative latent thermal energy storage to upgrade heat to 200-250°C. Auxiliary equipment sizing is determined by the fluid dynamics of the heat transfer fluid, which in this case is pressurized water, and the design of the piping is done for allowing flexibility between the different operational modes. After the experimental campaign, KPIs of the system will be measured to validate the performance of the different components and the design itself.

Acknowledgements

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