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Integrated High Temperature Heat Pump and Thermal Energy Storage Laboratory Rig – Engineering Considerations and Preliminary Design

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Abstract

The present work introduces the preliminary laboratory design of a flexible process heat generation based on medium temperature stream heat upgrade via combination of a high temperature Stirling Cycle heat pump and a latent heat thermal energy storage. Heat is upgraded up to 250°C using a high temperature heat pump recovering heat from a medium temperature water vector. This heat is then used for charging a thermal energy storage that can be used for different industrial heat applications such as, steam production or drying hot air.

Keywords: Heat upgrade, thermal energy storage, heat pump.

Introduction

Heat upgrade technologies are becoming increasingly relevant as one of the ways to meet the high thermal energy demand required for industry. The development and installation of these technologies represents a double benefit. Firstly, by using renewable energy sources, it contributes to reducing fossil fuel consumption. Secondly, heat for industrial processes is a market where other renewable-based technologies, like solar, are having limitations for deployment [1].

Commercial technologies (heat pumps) can effectively upgrade heat up to 120°C [2]. Nowadays, a wide part of the industrial sector requires higher temperatures, 150-250°C [3]. In this work, heat upgrade up to 150-250°C is obtained thanks to the use of an innovative high temperature Stirling heat pump uses helium, which drastically enlarges the industrial exploitability of waste heat upgrade systems. The integration of innovative energy storage solutions will ensure a reliable, flexible, and customizable heat delivery with full decoupling from any waste heat recovery and renewables availability, which is a usual challenge in different industrial sectors. The authors introduce an innovative sustainable heat upgrade system using a hight-temperature heat pump (HTHP) integrating a high and low temperature thermal energy storage (TES) by describing the preliminary design and engineering steps for the installation and commissioning of such novel lab scale facility. This work is a stepping-stone toward a comprehensive experimental investigation.

Methodology

The schematic layout of the system is shown in Figure 1. The central units of the system are the HTHP, one low temperature (70°C) phase change material (PCM) based TES, one medium temperature (155°C) PCM based TES, and one high temperature (245°C) PCM based TES. The low temperature TES is located on the cold side (left) of the HTHP while the medium and high temperature TES units are on the hot side (right) of the HTHP as shown in Figure 1. The laboratory rig is designed to deliver about 200kW$_{th}$ of the heat to produce steam or drying air, for practical reasons the rig will use a heat exchanger as heat sink instead of a steam generator to avoid handling steam in the laboratory. In the same perspective, an electric tracing is used to simulate the heat coming from a waste heat stream. The system is designed to use pressurized water as heat transfer fluid (HTF) in both high and low temperature loops.

Figure 1 shows four modes of operation: (1) charging of the TES and producing steam with the black lines, (2) discharging with the red lines, (3) only charging without delivering heat to the heat sink with yellow lines, and (4) TES are fully charge and only heat is delivered to the heat sink with blue lines. It is important to highlight that the system is designed to be flexible and operate in different conditions.

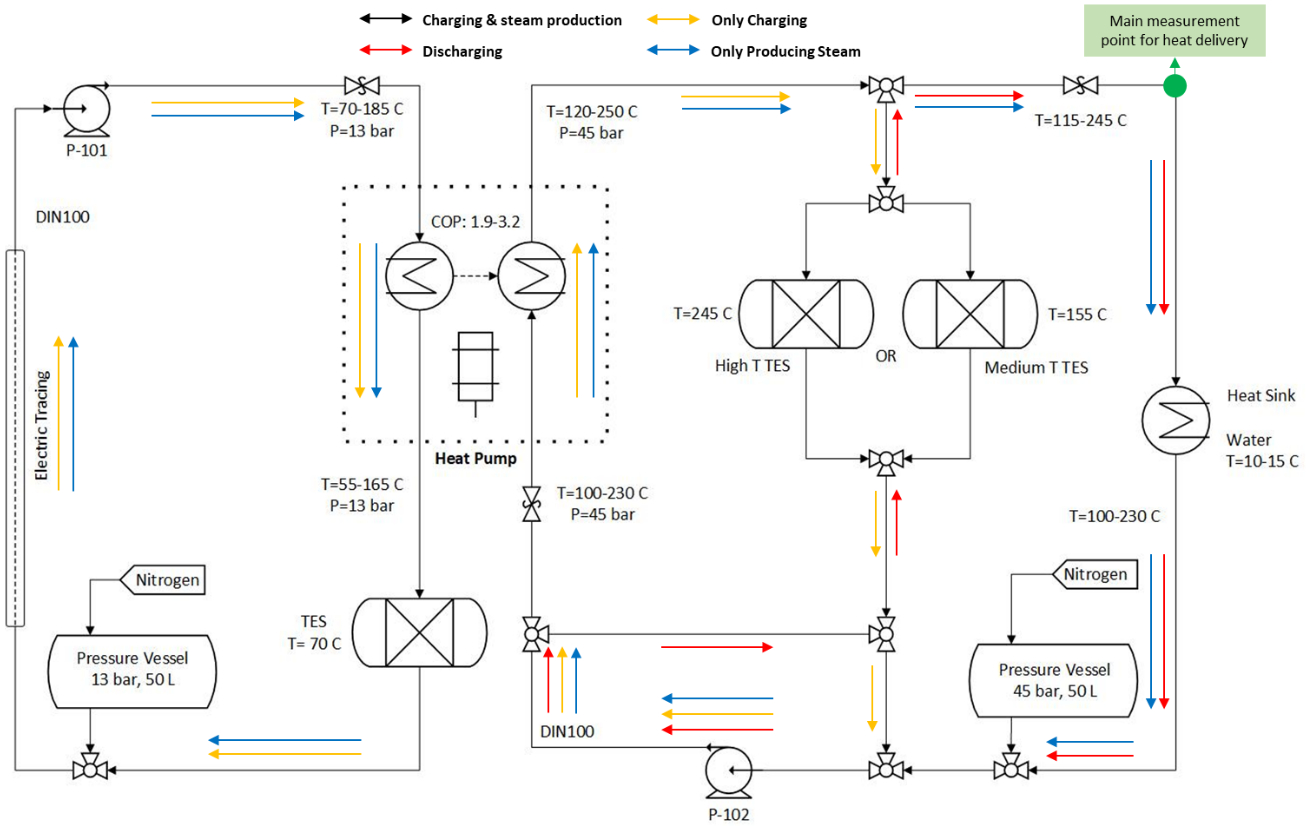

Figure 1. Proposed laboratory rig layout with different operation modes indicated in colours.

The system has three TES units, one with a phase change temperature at 245°C, a second with a phase change temperature at 155°C, and a third one at 70°C, giving the system more flexibility to storage heat in a wider range of temperature. For a giving running cycle only one TES on the hot side (i.e. medium temperature and hight temperature TES) can be use depending on the temperature of the system, meaning that only one can be charge or discharged at while running the system. The low temperature TES can be used at any time. It is also part of this research to investigate the performance of the system when upgrading heat up to 250°C while achieved a COP of 2.8, for this a maximum source temperature around of 185°C must be use for achieving the temperature lift.

When starting the system, the electric tracing on the cold side will heat up the HTF to the required inlet temperature of the cold side of the HTHP ($T_{C,in}$=70-160°C as seen in Figure 1) and pump-101 will start running the circuit through the cold heat exchanger of the HTHP, parallelly, pump-102 will also start running the hot circuit while a steady state of the cold side is reached. Then, the HTHP starts and heat is transferred by expanding the refrigerant, helium (non-toxic and inert, and has zero ozone depletion potential and zero global warming potential) at the low temperature heat exchanger and then transferring it to the pressurized water passing through the hot exchanger by compression until $T_{H,out}$=120-250°C is achieved, then heat can be deliver to the cooling district heat exchanger. When the hot circuit reaches temperatures from 160-250°C it can be used for charging the respective TES unit.

For this work, an industrial Stirling Engine as heat pump is selected. For each cold (source) temperature, several hot (sink) temperatures are applied using four different average COP: 1.95, 2.25, 2.55, and 3.0. It is known that for all heat pumps lower temperature lifts ($T_{H,out}$ – $T_{C,in}$) and larger ratios ($T_{C,in}$ / $T_{H,out}$) leads to higher coefficients of performance [4]. For example, with 120°C on the cold side it achieves a COP of 2.0 with a $T_{H,out}$ of 303°C and a COP of 3.0 with a lower $T_{H,out}$ of 178°C as shown in Figure 2 (clearest blue points on the left side).

When T_C and T_H have been defined for the boundary cases, it is possible to calculate the different parameters of the system by performing energy balances on the different components.

Heat Exchangers/District Cooling and pipe circuit

The design must start with the downstream heat exchanger (district cooling) which has been designed to deliver 200 kWth. The maximum inlet temperature is 250°C, and the maximum outlet temperature is 230 C. Thus, it is possible to calculate the mass flow of water inside the liquid and the heat losses by:

$$-Q_{exch} = \dot{m}_H cp_{H,out} T_{H,out} - \dot{m}_H cp_{H,in} T_{H,in} \quad (1)$$

Where $T_{H,out} = 250\ C, T_{H,in} = 230\ C$

By equalizing equation (1) to zero and solving for $\dot{m}$, the mass flow of the circuit can be obtained.

Then, for the Heat Pump:

ΔT between inlet and outlet temperature of the HTF on both heat exchangers of the HTHP is considered equal to 20 °C, which ensures keeping the corresponding COP. The design must consider the maximum and minimum conditions of temperature and COP (1.9-3.2) for calculating the mass and outlet temperature of water in the cold side of the heat pump. The equations involved for the heat pump are:

$$Q_H = \dot{m}_H cp_{H,in}(T_{H,out} - T_{H,in}) \quad (2)$$

Then using COP = 1.9 and 3.2 it is possible to calculate Q_C using equation (2), and to determine $T_{C,out}$ by

$$Q_C = \dot{m}_C cp_{C,in}(T_{C,in} - T_{C,out}) \quad (3)$$

Considering that $\dot{m}_C$ can be equal than $\dot{m}_H$, then $T_{C,out}$ is obtained. The mass flow can be set to keep a reasonable velocity of the fluid inside the circuit (0,25-1 m/s) and the $T_{L,out}$ is from the respective energy balance.

One important consideration for the laboratory rig design is the selection of the HTF, in this case, water which is non-flammable and does not have toxic risk. To work with water, the circuits must be pressurized above the saturation pressure at the given temperatures, otherwise vapour will be produced. Therefore, two scenarios for the pressure are foreseeing: the first one for building the desired pressure before the start-up; and the pressure lost within the circuit. For the latter, the segment thermal losses and the friction losses can be calculated for a given length, pipe diameter and flow regime.

Water pumps

For each pump the corresponding segment loss is determined by

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

Where D_i is the internal pipe diameter, and for laboratory purposes is established to be DIN100. Then, the velocity of the water in the section is calculated by

$$v = \frac{\dot{v}}{\pi \left(\frac{D_i}{2}\right)^2} \quad (5)$$

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

$$\dot{W}_p = \frac{\Delta P}{\dot{v}} \quad (6)$$

Pressure Water Vessels

For pressure water from heuristic the volume of water (% filled) is determined by

$$V_{pv} = 6\dot{m} \quad (7)$$

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

As main components for the system, two centrifugal pumps are used for controlling water flow and overcome pressure losses within the circuit. One pump in each side. As mentioned before, the water circuits must be pressurized above $P_{saturation}$ of water at the highest temperature, in the case of the cold side a pressure vessel at 13 bar will be used, and on the hot side a pressure vessel at 45 bar will be used to guarantee the circuit pressure. Minor components are valves and sensors as control equipment. The system foresees N1. T-port valve and N.1 Pressure relief valve (PRV) on the cold side, and on the hot side: N.7 T-port valves for connecting the circuit with the TES and N.2 PRV for pressure control, one before the HTHP and one before the district cooling unit. Key measurement point ought to be before and after the HTHP on both sides. The main temperature measurement point is indicated in Figure 1 as a green dot. Several manometers for pressure readings to be in the circuit. One flowmeter for flow control after each pump on both sides of the system. And pressure meters after and before the pressure vessel for regulating the air compressors and pumps according to pressure drop.

Boundary Conditions

The HTHP enables Coefficient of performance (COP) of 1.9 to 3.2 for temperature ratios between 1.1 to 1.5 (T/T °K) providing sufficient heat to produce high quality steam for heating, drying or distillation processes. The system will be investigated at thirteen (N.13) different operational scenarios as showed in Figure 2. where different temperature limits TC and TH define different COPs, and temperature ratios defining the boundary conditions of the system.

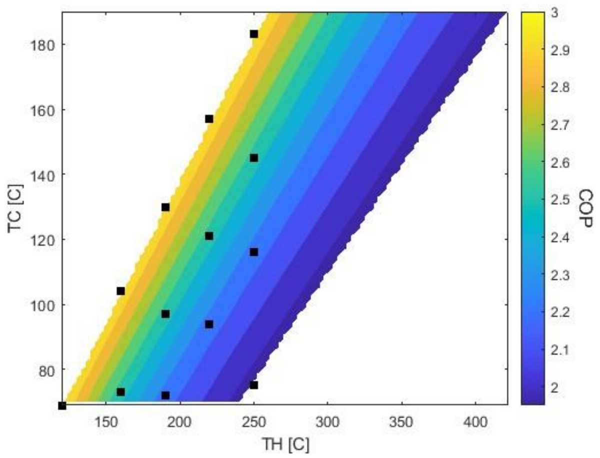

Figure 2. Operational scenarios for the HTHP

The thirteen cases have been chosen according to the industrial cases that this work is investigating. The first one, a dairy factory located in Greece, where the waste stream has a temperature of 70 C, and the hot stream is to be 200 C used for steam production. And a second, a fish related products in Norway, where the waste stream has a temperature of 90 C, and the hot stream is to be 250 C used for drying air. Within these cases, there are two that are considered as the limit cases to understand the minimum and maximum flowrate of water in the system as shown in Table 1.

Table 1. Boundary cases.

Case	Case 9	Case 13	Units
COP	3.0	3.0	
$T_{c,in}$	70	183	C
$T_{H,out}$	120	250	C
H_2O mass flow	8211	5145	kg/h
H_2O volumetric flow	2,42	1,79	L/s

Conclusions

The present work introduced the preliminary laboratory design of a flexible process heat generation based on low quality heat upgrade via a combination of a high temperature Stirling Cycle based heat pump and a latent heat thermal energy storage. The boundary conditions have been identified from the basic engineering and for going forward with the detail engineering. Future work will be focused on designing the control strategy, consolidation of the P&ID of the system including valves and sensors. And, finally, the start-up and operational manual of the system.

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