Latent heat storage for centralized heating system in a ZEB living laboratory: integration and design

To cite this article: Alexis Sevault *et al* 2019 *IOP Conf. Ser.: Earth Environ. Sci.* **352** 012042

Latent heat storage for centralized heating system in a ZEB living laboratory: integration and design

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Abstract. The ZEB Lab project, coordinated by SINTEF and NTNU, aims at building a ZEB (Zero Emission Building) in Trondheim (Norway) in 2019, to be used both as office building and living laboratory. An innovative latent heat storage (LHS) unit using phase change material (PCM) will be integrated in the centralized heating system. The LHS unit will be able to store excess heat from various heat sources connected to the heating system, when they are not required for space heating. One challenge is to make use of the full potential of the PCM latent heat to have a compact and effective unit, while the unit itself should have a low associated CO_2-footprint. The LHS system consists of two units designed for a total heat storage capacity of 0.6 MWh, corresponding to the heat needed on top of the heat pump to cover for up to 3 consecutive days in the coldest period of the year, with a maximum combined effect of 26 kW. A bio-based wax is used as PCM with melting temperature 37 °C and measured latent heat 198 kJ/kg. Dynamic system modelling is used to support the design of the LHS unit and ensure sufficiently high heat transfer rates.

Keywords. Phase Change Materials, PCM, Thermal Energy Storage, Heat Storage, Building, Centralized Heating system.

1. Introduction

1.1. Background

The ZEB Lab project (www.zeblab.no) , coordinated by SINTEF and NTNU, aims at building a ZEB (Zero Emission Building) in Trondheim (Norway) starting in 2019, to be used both as office building and living laboratory [1]. The building will rely on innovative technologies, both regarding building materials and energy system. One technology to be implemented and tested in the centralized heating system of the building will be a latent heat storage (LHS) unit using a phase change material (PCM).
The LHS unit will be integrated in the low-temperature heating system centred around a heat pump providing hot water for space heating. An essential asset of the LHS unit is to be able to provide high heat storage capacity and heat effect within the narrow temperature range offered by low-temperature heating. During hours of low heat demand, the LHS unit will be able to store excess heat. During high demand, the stored heat will be released either to provide heat directly to the heating circuit or to support the heat pump by compensating for a drop of return temperature below the optimal intake temperature. Since such LHS technology is not commercially available yet, an experimental unit has to be custom-designed following the best integration path in the heating system.

1.2. Objective
The main objectives of the present study are: (1) to assess one scenario for full integration of the LHS unit in the centralised heating system of the ZEB Flexible Laboratory; (2) to dimension the LHS unit according the building energy demand, while selecting the most appropriate PCM; and (3) to model the dynamic thermal performance of the designed LHS unit to validate its general design.

1.3. Literature review
A well-known challenge with using PCM for thermal energy storage is the poor thermal conductivity for available PCMs, limiting heat transfer rates [2, 3]. Comprehensive work has been done to increase the heat transfer rates within LHS systems by utilizing heat transfer enhancement techniques in numerical investigations and in experimental setups. However, only a few full-scale active LHS systems are in operation, making it challenging to document the potential upsides of coupling a LHS system to a heat pump for peak shaving and heating purposes.
Hirmiz et al. [4] studied the integration of LHS systems into heat pump systems to improve the demand side flexibility and, ideally, the strategy for the LHS system to cover the complete heat demand during peak periods. By utilizing a TRNSYS numerical model, it was concluded that a LHS system can completely offset peak heat demand periods within 2 to 6 hours, reducing peaks in the power grid. Through modelling and measurement data analysis, Jokiel et al. [5] evaluated a LHS system installed to reduce the required chiller capacity for three ammonia chillers/heat pumps covering the base load for heating/cooling at the Bergen University College (Norway). A dynamic system model was developed using Modelica [6] to better understand the dynamics of melting and solidification of the PCM. The model proved to correctly predict the measured data, within an acceptable accuracy, especially regarding the accumulated values of absorbed and released heat.
Bonamente et al. [7] studied the potential for system optimization in an existing ground-source heat pump heating system by implementing a TES unit. Computational fluid dynamic calculations were carried out and validated against measured data using two TES solutions: one using water as storage medium, and the other using PCM. Results showed that the COP of the system was increased from 2.9 to 3.2 and 3.4 for, respectively, heating and cooling modes when using water as TES medium. By using a PCM, the system COP was increased to 4.13 and 5.89 for, respectively, heating and cooling modes. In addition, the total volume of the PCM thermal storage was 10 times more compact compared to the water tank system making it more suitable for indoor installation and use.
Shifting the cooling load during simulated summer conditions was experimentally tested by Moreno et al. [8] by coupling a TES system to a heat pump. Thermal behaviour for the TES system was evaluated for cold storage and for space cooling. Two different TES configurations were tested, one using water and the other using PCM. The latter configuration utilized macro-encapsulated PCM with a phase change temperature of 10 °C. It was concluded that PCM storage is favourable to water storage. With identical volumes, the PCM tank was able to store 35.5 % more cold energy on average compared to the water storage tank. Other results indicated that by increasing the heat transfer rate for the PCM storage, it could store 14.5 % more cold energy, while delivering an acceptable indoor temperature for a 20.65 % longer duration compared to the water storage.

2. Integration of LHS unit

2.1. Centralized heating system
The building heating system is centralized and originally relies on a 25-kW heat pump system and several heat sources and heat sinks along the hot water heating circuit. The selected heat pump system is designed to provide a temperature lift on the hot side from 35 °C to 40 °C. Besides the heat pump, heat from the local district heating is used as heat source for the building, providing hot water at 45 °C. Throughout the heating circuit, preheating of domestic hot water, room radiators and heat exchangers providing heated air for ventilation are used as heat sinks to heat up the building. Additional components enabling research experiments in the different rooms of the buildings are also

planned but constitute minor heat sinks and heat sources on the heating circuit and thus are out of the scope of the present study. Without the LHS unit, the heat pump is meant to cover the maximum heat demand of the building, calculated to ca. 26 kW, necessary to maintain all rooms in the building at a comfortable temperature on the coldest days of the year in Trondheim (Norway). Using the LHS unit to support peak heating demands, the size and nominal effect of the heat pump can be significantly reduced, so that it operates more effectively. Combining a constant heat pump effect down to 12 kW with a charged LHS unit of 0.6 MWh would allow for a total heating effect of 26 kW for up to 42 h.

2.2. LHS unit integration

Among the possible scenarios to integrate the LHS unit in the centralized heating system, the integration enabling thermal buffering to support the heat pump was selected (see Figure 1). Depending on the heating demand in the building, the return temperature of the heating loop might be lower than 34 °C, and thus require additional power from the heat pump to sustain 40 °C as outlet temperature. Integrating the LHS unit downstream from the heat pump, with the option to circulate the return water through it or not, provides the opportunity to both charge and discharge the LHS unit, while smoothing the effect demand from the heat pump. Charging the LHS unit occurs when the heating demand is low, using 40 °C as inlet temperature, as it is generated by the heat pump. Using a PCM with phase change temperature within 34-37 °C, return water at lower temperature than 34 °C can circulate through the charged LHS unit and be heated up before entering the heat pump. Additionally, the LHS unit can be directly charged using the district heating loop providing hot water at 45 °C. Note that, as shown in Figure 1, two LHS units (LHS-1 and LHS-2) are integrated in the heating system, to ultimately allow for research experiments using various heat exchanger designs and test the thermal performance of several PCM. The present study focuses only on the design of LHS-1.

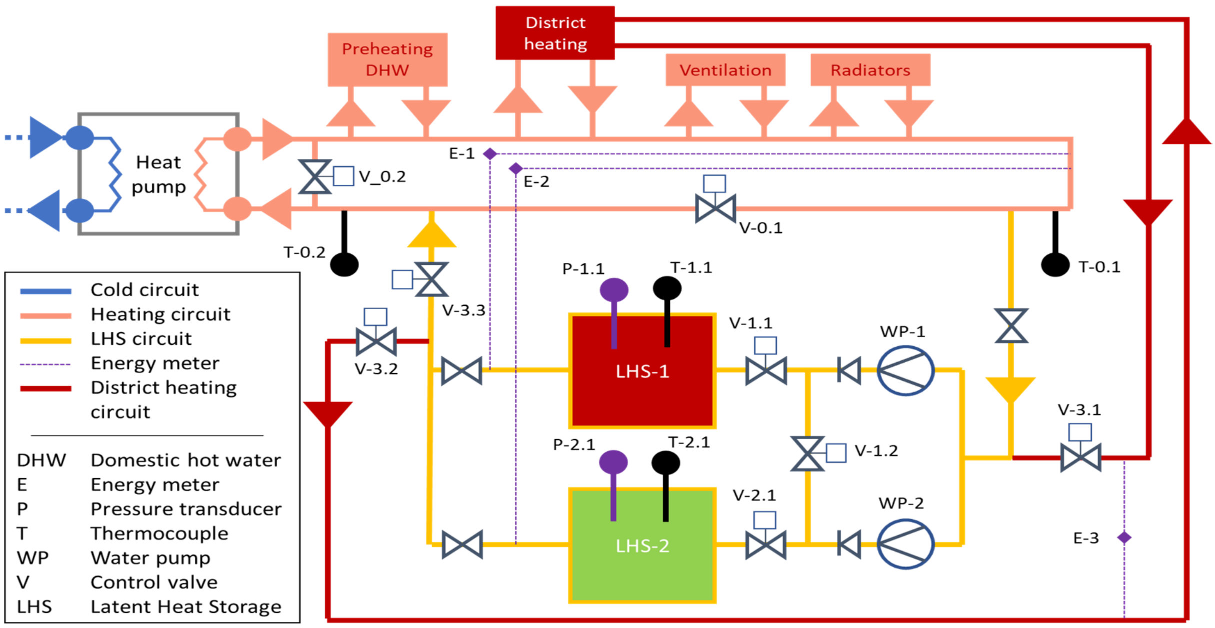

Figure 1. Process diagram of the centralized heating system showing the integration of the two LHS units. Only the instrumentation for control of the LHS units is shown for simplification.

Another feature available with this integration is the opportunity to use the LHS unit as a direct heat source in the building heating loop. This is meant to occur when the LHS unit is charged and the heating demand in the building is relatively low. Therefore, the heat pump can be bypassed, reducing significantly the energy use during these low-demand periods. This operational mode is especially interesting if energy price is integrated in the control system of the overall heating system.

2.3. Control strategy

As shown in Figure 1, the system allows for a variety of control strategies through a large number of control valves and two regulated water pumps. The selected strategy includes two levels: (1) a temperature-controlled strategy for charging and discharging using only the heat pump as heat source and heat sink; (2) a price-controlled strategy where the energy price is taken into account to decide when to harvest heat from the district heating network and when to use the LHS unit as direct heat sources for the building heating circuit. In both cases, the energy level of the LHS units is followed up using thermocouples located at various locations in the unit. Full charge is indicated by an average PCM temperature 4 K above its melting temperature range. Full discharge is indicated by an average PCM temperature 4 K below its solidification temperature range. In addition, three energy meters will enable to track the effect and accumulated transferred energy to follow up the thermal performance of the two units. The control system of the LHS system is to be fully integrated in the building control system, which will include a "researcher mode" to allow customizing and testing various control strategies.

3. Design and modelling

3.1. PCM selection and performance testing

The most suitable PCM for the LHS unit should primarily have a melting temperature within 34-37 °C, which yields only a limited range of commercially available PCMs. Table 1 lists a selection of commercial PCMs with melting temperatures ranging from 34 °C to 37 °C, as well as some of their thermodynamic properties given by the manufacturers.

Table 1. Selection of commercially available PCMs.

Product	*Type*	*Melting point [°C]*	*Heat of fusion [kJ/kg]*	*Thermal conductivity [W/(m.K)]*	*Manufacturer*
E34	Eutectic	34	240	0.54	PCM Products
A36	Organic	36	217	0.18	PCM Products
L36S	Salt hydrate	36	260	0.6	TEAP
E37	Eutectic	37	213	0.54	PCM Products
A37	Organic	36	235	0.18	PCM Products
CT37	Organic	37	202	0.24	CrodaTherm
PCM37	Organic	37	215	N/A	Microtek

After investigation of the pre-selected commercial PCMs listed in Table 1 for the melting temperature range 34-37 °C, the PCM CrodaTherm 37 (CT37) was selected. CT37 is a water-insoluble organic PCM, derived from plant-based feedstocks [9]. The PCM appears as a crystalline wax in solid state and oily liquid above melting temperature. The main arguments in favour of this PCM are its low degree of supercooling (cf. Table 2), its low-carbon footprint as well as its affordable cost. In addition, CT37 has low flammability, which is an essential criterion in buildings.

A sample of CT37 received by CrodaTherm was analysed by Thermogravimetric Analysis – Differential Scanning Calorimetry (TGA/DSC) at the SINTEF Energy Laboratory to evaluate the thermodynamic performance of the PCM. A measurement of 10 melting/solidification cycles was performed using a TGA/DSC SDT600 from TA Instruments, with controlled heating and cooling rates of 1 K/min ranging from 30 °C to 50 °C, in a nitrogen atmosphere. The results are shown in Figure 2. As indicated by the manufacturer, the first melting displays a significantly larger latent heat of fusion than the following melting/solidification cycles. Taking into account only the 9 following cycles, CT37 remains absolutely stable, yielding very similar heat flow patterns. The average latent heat of fusion is 198.6 kJ/kg (+/- 0.9 %) and the average latent heat of crystallisation is 196.4 kJ/kg (+/- 0.7 %). The average peak melting temperature peak is 36.5 °C (+/- 0.3 %) and the average solidification temperature peak is 34.5 °C (+/- 0.1 %). The weight loss is measured to 0.04 % along the first two cycles and then remains stable for the following 8 cycles. Note that thermodynamic property measurements might be variable from one device to another and is also known to depend on the sample mass and measurement procedure.

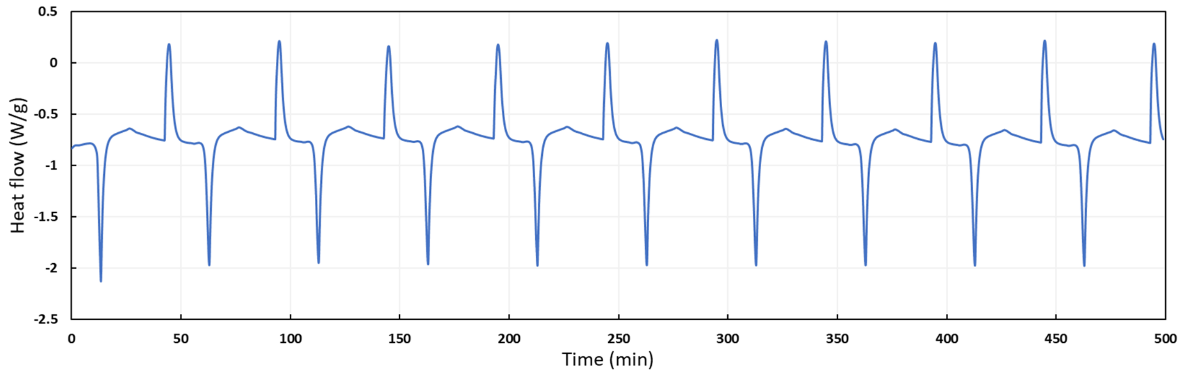

Figure 2. DSC measurements of heat flow absorbed and released by a sample of CT37 for 10 melting/solidification cycles.

3.2. *General design*

The general design parameter of LHS-1 are given in Table 2. The LHS unit dimensions are the first constraints to consider for the unit design due to the architecture of the building, limiting the access into the technical room through a 1.8-m wide corridor. This justifies the idea of having two LHS units whose dimensions allow to enter the building (see Table 2). The design of LHS-1, shown in Figure 3, is based on a fin-and-tube heat exchanger, filled with PCM. Water from the heating circuit circulates in the tubes. The design parameters of the fin-and-tubes heat exchanger are discussed in Section 0Headers at both ends of the unit enable a homogeneous distribution of the water across the tubes. A thick thermal insulation around the LHS-unit allows for a theoretical heat loss under 2 % per 24 h.

Table 2. General design parameters of LHS-1 using a fin-and-tube heat exchanger design.

Properties of LHS-1	***Values***
Dimensions of LHS-1 unit (height x width x length) *[m]*	1.5 – 1.4 – 2.5
Measured PCM melting temperature range and peak *[°C]*	35 – 39 (36.5)
Measured PCM solidification temperature range and peak *[°C]*	33 – 35.5 (34.5)
Measured PCM latent heat of fusion *[kJ/kg]*	198.6
Measured PCM latent heat of crystallisation *[kJ/kg]*	196.4
PCM density *[kg/m^3]*	957 (at 32 °C), 819 (at 75 °C)
PCM thermal conductivity *[W/(m.K)]*	0.24
PCM specific heat capacity (solid – liquid) *[kJ/(kg.K)]*	2.3 – 1.4
PCM degradation temperature *[°C]*	> 50
Total theoretical thermal storage capacity *[from 30 to 40 °C] [kWh]*	325
Ratio of latent heat to total heat storage capacity	90 %

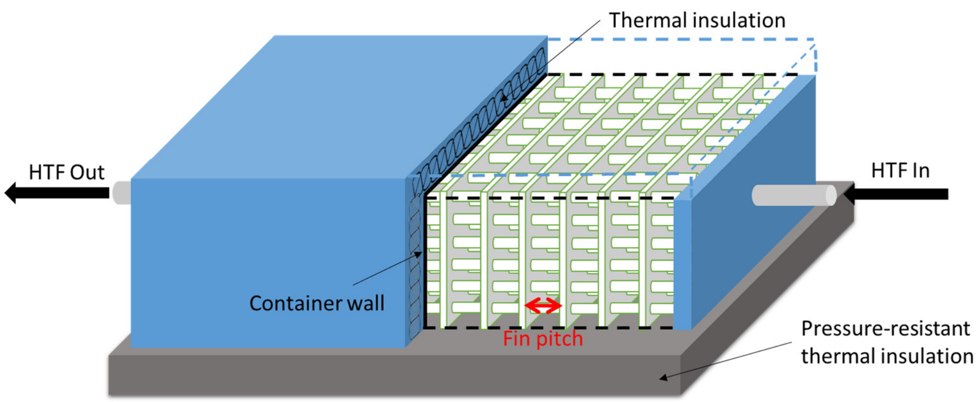

Figure 3. Simplified geometry of the LHS unit. The PCM (not shown here) will occupy the space between the plate fins.

4. Dynamic system modelling

4.1. Model description and assumptions

To investigate the transient nature of the charging and discharging processes of the LHS unit, a heat exchanger model using PCM for heat storage has been developed in the modelling and simulation software Dymola [10]. Dymola allows dynamic modelling of thermal systems with variable inputs. It is based on the open, object-oriented modelling language Modelica [6]. The specific heat exchanger model was developed using validated TIL libraries from expert thermodynamic model developers at TLK-Thermo GmbH [11]. The model is based on a solid-liquid fin-and-tube heat exchanger using PCM for TES custom-made by TLK-Thermo GmbH in 2017 and utilised in previous works [12]. This LHS model only has one mass flow, for the heat transfer fluid (HTF), water here. The model requires the following input parameters: heat exchanger basic geometry, fin geometry, HTF inlet temperature, HTF mass flow rate, PCM and material properties, as well as the initial temperature for respectively the HTF, PCM and heat exchanger. The LHS model consists of tubes and plate fins running through a tank filled with PCM. Heat transfer rates are calculated from the HTF to the PCM through tube walls and plate fins following a 1-dimension heat transfer model. The specific LHS model is still to be experimentally validated using PCM CT37, though it is assumed sufficient for a first assessment.

The dimensions of the tank are 1.5-m-high, 1.4-m-wide and 2.5-m-long. To be able to operate the LHS system during peak periods (e.g. cold days), the LHS unit should be fully charged in advance. The control strategy involves to fully charge the LHS unit during off-hours, typically 10 hours during the night or during warmer days since the storage capacity enable several worth of heat demand in the coldest days. The LHS model enables to evaluate the heat transfer area required to be able to fully charge the system in a predetermined number of hours, assuming a HTF inlet temperature of 40 °C as it can be delivered by the heat pump in off-peak hours. Note that this is the most critical case for charging processes since the connection to district heating enables quicker charge at 45 °C.

The following assumptions are made for the numerical simulations:

- Heat transfer coefficient between HTF and inner pipe wall are calculated using Dittus-Boelter correlation [13].
- Hysteresis effects are not accounted for in the PCM.
- No heat loss through tank wall.
- The PCM phase changes are complete during charging/discharging processes (no partial load).

Simulations are carried out for six different scenarios for both charging and discharging processes. Three scenarios are simulated with fin pitches of, respectively, 1, 2 and 4 cm, in the heat exchanger geometry for a constant HTF mass flow of 2 kg/s. The next three scenarios are simulated with a variable HTF mass flow of respectively 1, 2 and 3 kg/s, for a constant fin pitch of 2 cm. The heat exchanger consists of 225 pipes, arranged 15 x 15 across a transversal section, along the longitudinal axis, with 1 cm inner diameter. Fin thickness is set to 1 mm. The simulations are performed over a sufficiently long time period to allow for complete melting or solidification of the PCM. For the charging processes, the initial temperature is 30 °C in the whole LHS unit and the HTF inlet temperature is 40 °C. For the discharging processes, the initial temperature is 40 °C for the whole LHS unit and the HTF inlet temperature is 30 °C.

4.2. Results and discussion

Figure 4 shows the simulation results for one charging process with the tested fin pitches. Figure 4 (a) shows the average PCM temperature throughout the LHS unit. The charging or discharging time is defined here as the time required for the average temperature of the PCM in the LHS unit to reach the HTF inlet temperature. By reducing the fin pitch from 4 cm to 1 cm, the charging time is reduced from ca. 16 h to 9 h. This is mainly because of the increased surface heat transfer area from the increased quantity of fins, and because the shorter fin pitch also reduces the PCM volume in the LHS unit by 7

%. Figure 4 (b) shows the PCM liquid fraction and heat flow from the PCM to the HTF during the melting process. Heat transfer flows are initially high due to the larger temperature differences driving the heat transfer. They rapidly decrease within the sensible heat transfer region, until the heat flows stabilise when reaching the latent heat transfer region for fin pitches of 1 cm and 2 cm. The fin pitch of 4 cm does not yield such a sharp change during phase change.

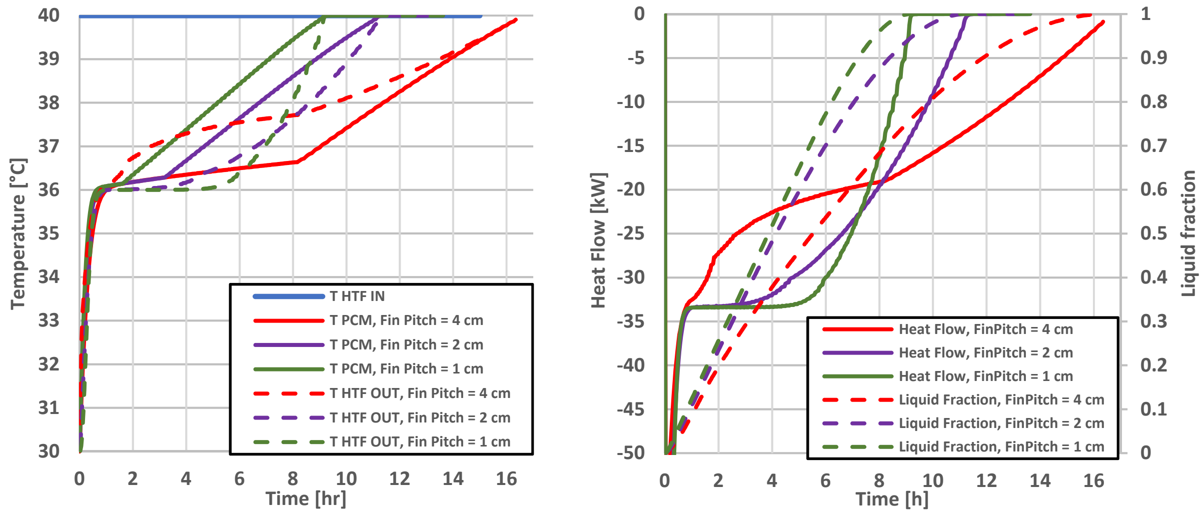

Figure 4. Charging process for various fin pitch with constant HTF mass flow of 2 kg/s, (Left) Temperature of HTF at inlet and outlet, and average PCM temperature; (Right) PCM liquid fraction and heat flow from PCM to HTF.

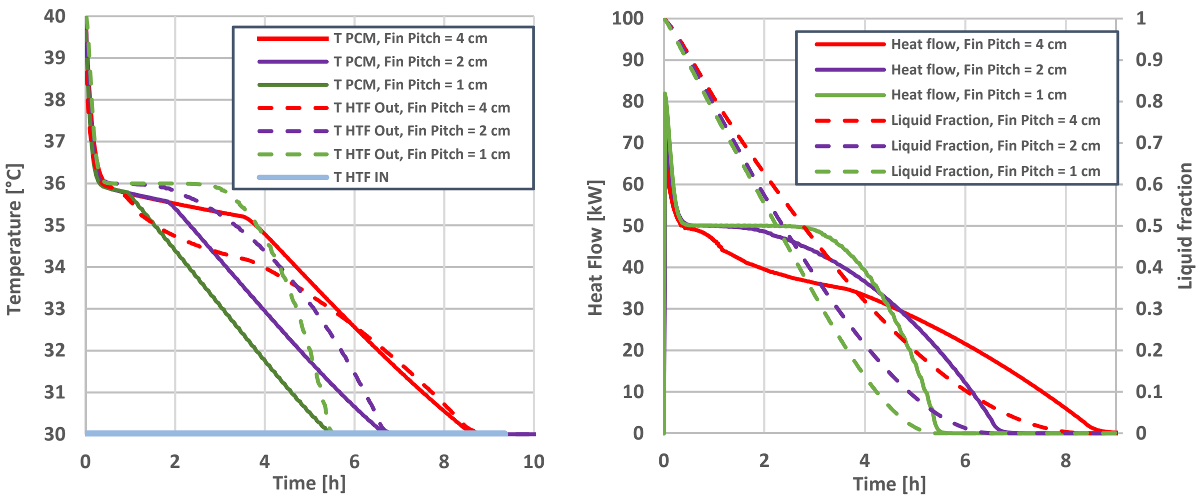

Figure 5. Discharging process for various fin pitches with constant HTF mass flow of 2 kg/s, (Left) Temperature of HTF at inlet and outlet, and average PCM temperature; (Right) PCM liquid fraction and heat flow from PCM to HTF.

Figure 5 shows the results of the discharging simulations for the tested fin pitches. As shown in Figure 5 (a), a fin pitch of 1 cm yields an HTF outlet temperature almost constant at 36 °C for 2 hours, which is optimal for the steady operation of the heat pump. Increasing the fin pitch results in steadily decreasing HTF outlet temperature during discharging. Figure 5 (b) shows that reducing the fin pitch provides higher and more constant heat flows in the latent heat transfer region, ultimately leading to shorter discharging times. According to the simulations, a fully charged system stores respectively 293 kWh, 286 kWh and 271 kWh for a fin pitch of 4 cm, 2 cm and 1 cm. This yields average heat transfer rates of ca. 35 kW, 44 kW and 50 kW for the various fin pitch distances during discharge.

These results validate the thermal performance for the proposed design of the LHS system, both in terms of charging times and heat transfer rates during discharge. Charging of the system may occur over night when the building is not in use, typically for a time period of maximum 10 hours. Figure 4

(a) shows that this can be achieve using a fin pitch distance of 1 or 2 cm. Figure 5 (a) confirms that the LHS unit can raise the return temperature of the heating circuit closer to design intake conditions for the heat pump (35 °C). Figure 5 (b) demonstrates that the LHS unit is able to deliver more than sufficient heat transfer rates, to meet the maximum heat demand in combination with the heat pump.

5. Conclusion

The aim of the present study is to evaluate the integration and design strategies for a LHS unit to be implemented in the centralized heating system of the ZEB Flexible Laboratory building in Trondheim (Norway). The LHS unit is designed based on a fin-and-tube heat exchanger filled with PCM whose phase change temperature is 35-37 °C. Using dynamical system simulations, the thermal performance of the proposed LHS design has been evaluated. The simulated LHS unit can store up to 325 kWh and simultaneously achieve sufficiently high heat transfer rates during discharge to successfully back up the heat pump during the coldest winter days.

Acknowledgements

The current study was carried out through the competence-building project PCM-Eff supported by SINTEF Energy Research using basic funding from the Research Council of Norway. The project aims at investigating novel phase change material solutions for efficient thermal energy storage at low and medium-high temperature (www.sintef.no/en/projects/pcm-eff/). The LHS unit is co-financed through the ZEB Flexible Laboratory by SINTEF, NTNU, Research Council of Norway and ENOVA.

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