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Dimensioning and techno-economic-assessment of thermal energy storages in the food processing industry using energy load profiles

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ABSTRACT

The food industry is a major consumer of electrical energy, which is required for cooling, freezing, drying and heating. Due to the production characteristics, high load peaks often occur in food processing. This leads not only to the need of oversizing the required equipment (e.g. compressors), but also to a shorter lifetime of these, as well as high peak load electricity prices. By integrating a thermal energy storage (TES), supply and demand for thermal energy can be decoupled, thus avoiding peak loads and ensuring a more stable operation of the refrigeration system. At the same time, TES ensures stable and low temperatures and thereby food quality and shelf life. Sensible TES are commonly used in the processing industry in the form of large water tanks, but latent TES using phase change materials (PCM) as storage medium are still under development for different applications. In particular, cold thermal energy storage (CTES) using PCM for storage temperatures below 0 °C are not widely used. In this paper, a python algorithm is presented that uses inputs from a process (hour-based thermal demand and electricity prices) to predict the impact of introducing TES in terms of reducing operating costs. The algorithm uses an optimization-based method to select and dimension the cost-optimal size of pillow-plate PCM thermal storage. In this paper, the Python algorithm is tested using load profiles from the pelagic fish processing industry, with ammonia refrigeration system, which is particularly challenging due to unpredictable and periodic production rhythm.

Keywords: food cold chain, energy efficiency, thermal energy storage, phase change materials, sustainability

1. INTRODUCTION

The European Union, together with Norway and Iceland, aims to reduce greenhouse gas emissions (GHG) by 55% below 1990 level by 2030 and to be carbon neutral by 2050 (European Commission, 2020). To achieve this, both electricity and thermal energy must be generated from renewable sources. In 2020, just 23.1% of the final energy consumption for heating and cooling purposes in the EU was covered from renewable sources (Eurostat, 2022). Thermal energy storage is a key technology for increasing this share as quickly as possible (Beck et al., 2021; Cirocco et al., 2022; Selvnes et al., 2022). This can be done, for example, in combination with renewable electricity-powered heat pumps and refrigeration systems with natural working fluids (Hafner, 2019).

The thermal energy demand of the food processing industry can vary considerably on a daily, weekly, or even seasonal basis, depending on the product, the production method, and the production schedule (Selvnes et al., 2022; Widell et al., 2022). By integrating TES into the factories, not only thermal but also electrical demand side management can be operated, thus avoiding peak loads. Peak load shaving and load shifting not only helps to reduce the load on the compressors and the electricity grid, but also increases energy efficiency and reduces energy costs. (Cirocco et al., 2022; Selvnes et al., 2022)

TES can be characterized by the method they use to store the energy (sensible, latent or thermochemical), by the temperature level (cold or hot) or by the type of heat exchanger that is used to charge and discharge the

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TES (Beck et al., 2021; Yang et al., 2021). This paper focuses on the integration of cold thermal energy storages (CTES) which uses a phase change material (PCM) to store thermal energy with the help of a pillow plate heat exchanger. In the pelagic fish industry, around 75 % of the electrical energy consumption is related to refrigeration systems and around 90 % to 95 % of this is related to the refrigeration compressors which mostly produce cold thermal energy to freeze the fish. To reduce peak power consumption, it is particularly suitable to take measures that reduce the energy consumption of the compressors or shift the load. In case of the pelagic fish industry, cutting the peak power demand by a production stop over a certain period isn't an option due to a negative impact on the product quality (Widell et al., 2022). For this reason, CTES technology with PCM can be an attractive option to achieve peak shifting of the refrigeration load handled by the compressors.

In the field of CTES, there have been some publications in recent years. Selvnes et al. (2021, 2022) experimentally investigated a novel pillow plate heat exchanger (PPHX) for use in a CTES unit using PCM as the storage material. This novel heat exchanger consists of two thin stainless-steel plates that are spot welded in a pattern that forms channels after the plates are inflated by a hydroforming process, forming the characteristic wavy surface. As the heat exchanger is fully welded, it is possible to use the internal flow channels for a high-pressure primary refrigerant such as CO_2 up to a pressure of 120 bar. If different refrigerants are used for charging and discharging, it is possible to use a so-called double-expanded PPHX with two inner channels.

In this study, a python algorithm is presented that uses an optimisation-based method to select and dimension a cost-efficient TES based on energy data. The aim was to develop an algorithm that can predict the techno-economic potential savings of implementing a TES in industrial processes, based on for example the energy price and the energy consumption of a production month. The algorithm was tested by using the cooling demand of the ammonia refrigeration system of the freezing tunnels of a pelagic fish processing plant as an input. A potential analysis of the possibility of installing CTES in this factory has already been carried out by Widell et al. (2022) based on an energy flow analysis.

2. METHODOLOGY AND SYSTEM DESCRIPTION

The cooling demand used as input to the python algorithm comes from the ammonia compressors associated with the freezing tunnels of a factory that processes pelagic fish (herring and mackerel). Strong seasonal and daily fluctuations in energy demand are typical of the pelagic fish industry. In spring (March to May) and in July, production usually comes to a standstill or almost to a standstill, because the required species of fish are not in season. Autumn and winter, on the other hand, are the main production seasons. Here, the variation in energy demand is related to the amount of fish bought by the processing plant. Figure 1 shows the seasonal and daily energy demand of a Norwegian fish processing plant, together with the monthly peak power for the year 2021. A detailed energy flow analysis of this factory was given by Widell et al. (2022).

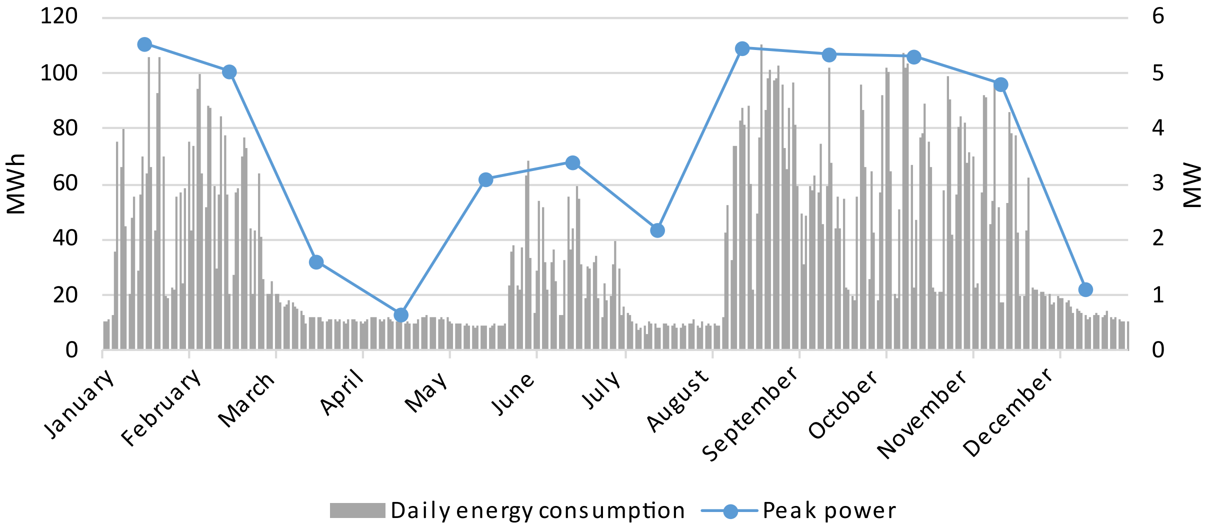

Figure 1 Total daily energy demand at the plant for 2021, including monthly peak power (Widell et al., 2022)

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2.1. Description of the ammonia refrigeration system

The installed ammonia cooling plant has a maximum cooling load of approximately 4 MW, which is only required on days with very high production. It has 5 tunnel freezers operating at an evaporating temperature of -40 °C and two freezing storages and a total charge of around 30 tonnes of refrigerant. The condensers are chilled with seawater, while the superheat of the refrigerant can be recovered to heat process water for cleaning purposes. For the freezing process, whole or filleted fish are packed in cardboard boxes of 20 kg each and stacked in the freezing tunnel. They are cooled to an internal temperature of -18 °C in a batch process, which can take up to 22 hours. Each tunnel can freeze 125 tonnes of fish per batch. An illustration of the refrigeration system is shown in Figure 2.

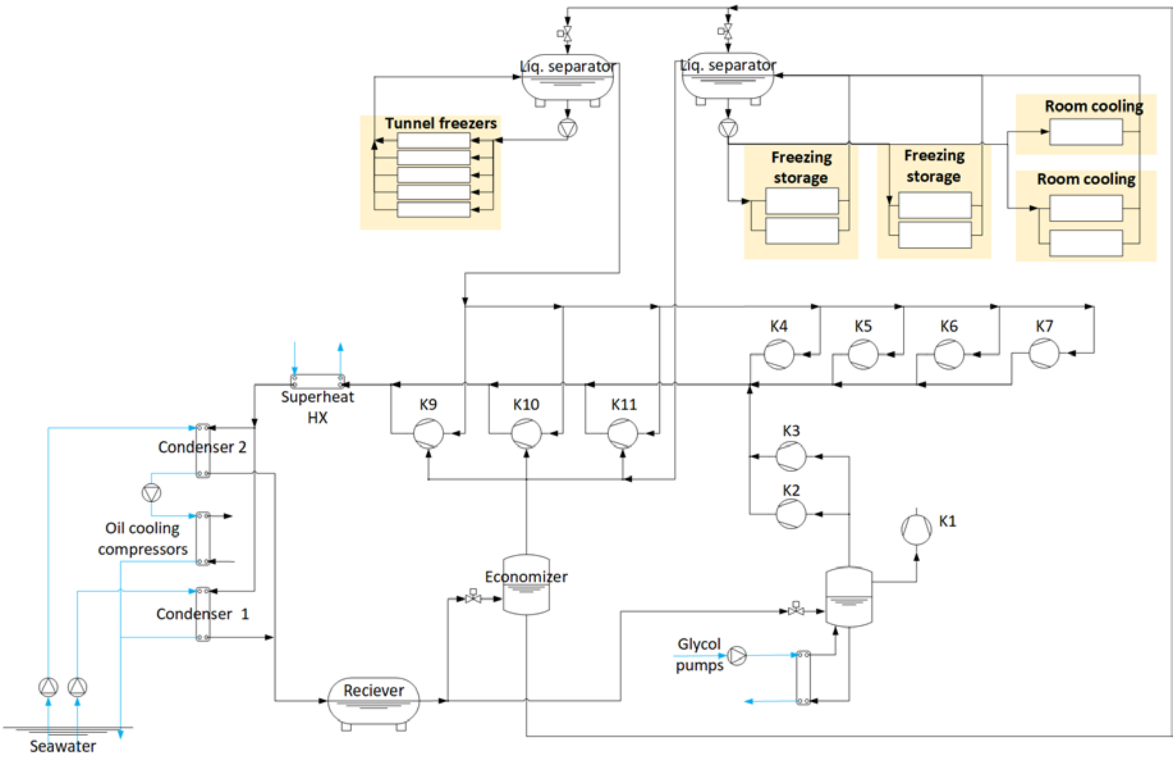

Figure 2 Illustration of the ammonia refrigeration system (Widell et al., 2022)

A suggestion on how to integrate a CTES into the existing ammonia refrigeration plant is shown in Figure 3. The evaporators in the tunnel freezers are operated at -40 °C. For the CTES, the melting temperature of the PCM should be at least 5 K below the evaporation temperature of the tunnel freezers to ensure a minimum temperature difference during the discharging process of the storage. Another 5 K below that is the temperature at which the CTES must be charged. Since ammonia cannot be operated at -50 °C evaporation temperature due to practical limitations, CO_2 refrigeration system should be used to charge the CTES. When the CTES is charged, the NH_3 loop bypasses the CTES via the dashed line and is operated as before the integration of the CTES. In the charging process, the CTES serves as an evaporator for the CO_2 refrigeration system. This means that heat is absorbed and the PCM can undergo a phase change from liquid to solid. Through the CO_2 condenser, the heat extracted from the CTES is then discharged into the ammonia circuit, where the heat is cascaded further into the system. When the CTES is to be discharged, the refrigerant first evaporates as usual in the evaporators of the tunnel freezers. Subsequently, the two-phase refrigerant flows through the CTES where it condenses and transfers the condensation heat to the cold storage tank, and thereby discharges it.

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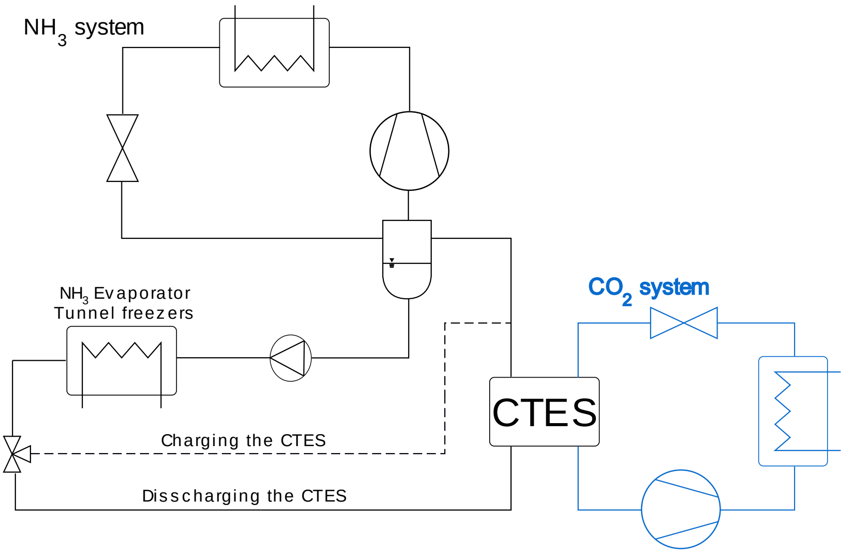

Figure 3 Simplified P&ID how an integration of the CTES into the existing ammonia system is possible.

2.2. The Pillow plate CTES

In this paper the PPHX was used in the refrigeration system as an evaporator (charge case) or condenser (discharge case). The PPHX offers a high heat transfer rate and is suitable for a variety of industrial applications. Depending on the capacity required, several plates can be stacked horizontally or vertically to form a PPHX which can be used as a CTES. The plate-to-plate pitch is critical as it is responsible for the charging and discharging characteristics of the CTES. A small plate pitch is suitable for unloading the compressors of a refrigeration unit when the application characteristic consists of short but sharp load peaks. By increasing the pitch, the CTES is suitable for longer lasting peaks. (Selvnes, Allouche, & Hafner, 2021)

While more and more research being done, published and made commercially available in the field of CTES, there is often a lack of experimental studies and findings for applications on an industrial scale in the temperature range between -25 °C and -50 °C (Yang et al., 2021). However, this temperature range is important for the food industry when freezing food. The PCMs available on the market with a phase change temperature around -50 °C are almost exclusively salt hydrates, which have various disadvantages. These are corrosive, suffer from phase separation and supercooling effects, and undergo incongruent melting, which is not optimal for a CTES unit. (Mastani Joybari et al., 2023; Selvnes, Allouche, Manescu, et al., 2021) This shows the need for further research into better PCMs and to test them in industrial applications.

In this work, a PCM with a melting temperature of -45 °C was used and the pillow plate design mentioned above for the heat exchanger was selected. Regarding the PCM, a modified version of ATS-40 from Axiotherm GmbH (2023) in which the phase change temperature has been lowered from -40 °C to -45 °C was used. All relevant material properties of the PCM are summarised in Table 1. The geometry was simplified as rectangular blocks in which canals for PCM and refrigerants alternate, which was suggested by Försterling et al. (2022).

A modular approach was taken for the TES to be able to easily implement (and visualize) the TES in industrial applications at a later stage. Each module consists of a 20-foot shipping container with internal dimensions of 5.9 m x 2.35 m x 2.4 m. As there must be enough space for the PPHX implementation with the piping etc., the dimensions for the storage unit were set at 5.5 m x 2.2 m x 1.8 m, as shown in Table 1. The TES can thus be easily pre-assembled, transported and later extended if more storage capacity is required at the plant. On site, the containers have to be connected to each other and to the existing refrigeration system according to the proposed integration in Figure 3.

2.3. Dimensioning- and decision-making tool

A dimensioning and decision-making tool has been developed to perform a techno-economic assessment of the TES integration in industrial processes. The tool has been built upon a previous work of Beck et al. (2021).

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The tool's purpose is to identify an optimal solution for TES integration into a thermal process, i.e. selecting a PPHX with optimal dimensions in terms of storage capacity, heat loads (charge/discharge) and storage operation. In short, the tool requires some input parameters that describe a case (thermal demand and energy price profile for a given period, and some case-specific parameters) which is used to create an optimization model, in which the cost function can be defined as minimizing the total energy costs for the simulation period. The optimization is done by a MILP/MIQP (mixed integer quadratic programming) model. For dimensioning of PPHX, the tool iterates a number i of different geometries in a discrete manner, and calculates the heat load and capacity per unit length for each iteration. Each solution is then rescaled j times with respect to capacity, heat loads and volumes, which results in a set of $i \cdot j$ solutions. Lastly, an elimination procedure to remove suboptimal solutions is performed leaving the final set of solutions to $n < i \cdot j$. Suboptimal solutions are solutions yielding the same performance in capacity and heat load, but at somehow worse geometrical definition (assumed to increase capital cost). A more detailed description of the tools internal procedure was given by Beck et al. (2021), while the thermal calculations for PPHX are described here.

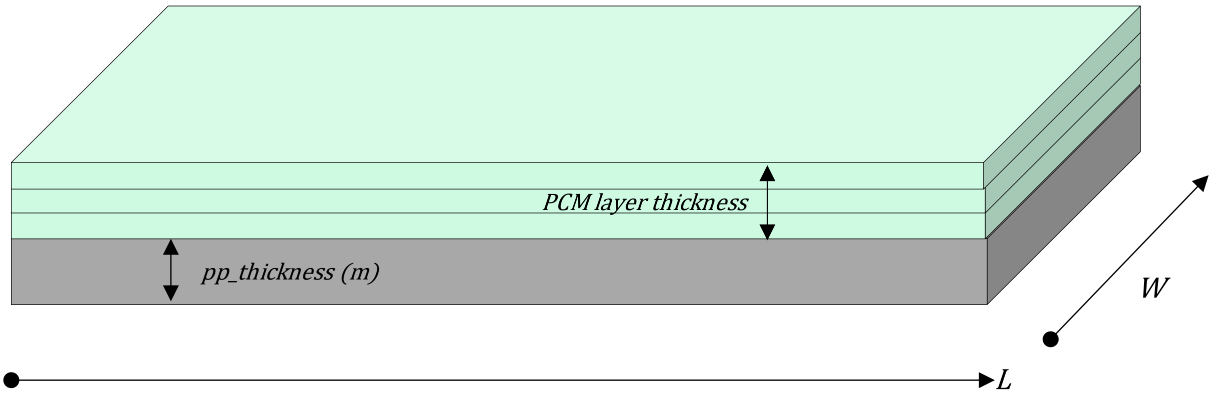

Figure 4: Schematic drawing of pillow plate with PCM layer. Refrigerant flows internally through the pillow plate along the length axis

Figure 4 shows a schematic representation of a pillow plate in contact with PCM on the top side. The plate thickness (*pp_thickness)* and width (*w*) are constant parameters, defined by the real boundaries of a container vessel as described in 2.2. Thickness of the PCM layer is discretized into *i* ranges between a user-defined range which describes the thickness as a fraction *f* of the plate thickness (e.g., *i=1, layer thickness=5*$pp_{thickness}$* given that the range starts with a factor of 5). The layer thickness can be interpreted as plate distance in a real PPHX, which is an important parameter for attaining desired heat load rates. For each layer thickness, the energy capacity is calculated per unit length as:

$$Q_{capacity} = V_{PCM} \cdot (E_{lat} + E_{sens} \cdot dT \cdot \eta_T) \left[\frac{W}{m}\right]$$

Where E_{lat} $(\frac{W}{m^3})$ and E_{sens} $(\frac{W}{m^3 K})$ are PCM specific parameters, dT (K) is a case specific parameter describing the temperature difference between the refrigerant and melting temperature of the PCM and η_T is a temperature efficiency factor describing the boundary reach of the temperature range. PCM volume (V_{PCM}) is calculated for each iteration as:

$$V_{PCM} = W \cdot (pp_{thickness} \cdot f - pp_{thickness}) \left[\frac{m^3}{m}\right]$$

To calculate the heat load, a quasi-stationary node model using an enthalpy approach was used. In short, the PCM layer was divided into several nodes, in between the heat transfer by conduction and temperatures were calculated in time increments Δt. Some simplifications were: a constant heat transfer coefficient and PCM conductivity were assumed for the entire process, for both charging and discharging, and it was assumed that the refrigerant mass flow was sufficient to ensure full evaporation or condensation within the plates. These assumptions were made to generate rapid (with respect to computation time) results.

After rescaling and elimination of suboptimal solutions, the resulting set consisted of solutions with scaled capacities and heat load rates, but also scaled plate lengths since the internal procedure only considered one, long plate. To transform the dimensions to conform with the boundaries of a real storage vessel, number of plates were first calculated by restricting the plate length to a maximum of 5.5 m, and number of storages were calculated by ensuring that the height of plates and PCM layer did not exceed the height dimension of 1.8 m.

$$n_{storages} = roundup\left(\frac{n_{plates} \cdot pp_{thickness} + \left(n_{plates} + 1\right) \cdot layer_{thickness}}{1.8}\right)$$

It was assumed that each storage had identical configuration with respect to dimensions, number of plates, storage capacity and heat load rates.

Finally, the energy costs were calculated as the product of $\dot{P}_{el}$ and $c_{energyprice}$ for each timestep for the base case and all iterations. For the iterations, the optimization model aimed to minimize the overall energy costs, which means that it aimed to charge the storage during off-peak hours and discharging during high peak hours, by looking at the relative price difference over time in the inserted energy price profile.

2.4. Case definition

For this paper the process data from the pelagic fish industry was used as a case. This included the thermal loads that occur in the freezing tunnels during the freezing of the pelagic fish and a 24-hour average electricity price profile that the company had to pay. The ammonia refrigeration system had a maximum cooling capacity of 4000 kW, an evaporating temperature of -40 °C and a condensing temperature of 25 °C as shown in Table 1. When considering an isentropic compressor efficiency of 70 % a refrigeration COP of 1.9 was calculated using *CoolProp.* Since the refrigeration system of the pelagic fish processing industry contains losses, the COP used in the algorithm was set to 1.7 which is assumed to be constant over the period considered.

Table 1 Initial values

Storage parameters	Value	Unit
Container length	5.5	[m]
Container width	2.2	[m]
PP thickness	0.004	[m]
Container height	1.8	[m]
PCM parameter		
Heat conductivity	0.60	[W/(mK)]
Latent heat capacity	245	[MJ/m^3]
Sensible heat capacity	4380	[kJ/m^{3}K]
Melting temperature	-45	[°C]
Density (liquid/solid)	1450/1363	[kg/m^3]
Ammonia refrigeration system		
Heat transfer coefficient	3500	[W/(m^{2}K)]
Evaporation temperature	-40	[°C]
Condensation temperature	25	[°C]
COP	1.7	[-]
Maximum cooling capacity	4000	[kW]
Simulation parameter		
Max. storage temperature	-40	[°C]
Min. storage temperature	-50	[°C]
Temperature efficiency factor	0.8	[-]

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A period of 100 h was selected as a case to test the model and illustrate the intention. Cooling demand and electricity costs for this period is shown in Figure 4. In the first 20 hours there is a high thermal energy demand and at the same time a high electricity price. With the help of a TES the load on the refrigeration system could be shifted to a less expensive time, thereby reducing costs. The average cooling demand for this period 1200 kW, with a minimum of 300 kW and maximum at almost 3800 kW. Energy prices for the same period range from 0.28 to 0.55 NOK/kWh (26-51 EUR/MWh) in a repeated 24hr pattern. These prices were based on real hourly prices from the region in which the plant operates, averaged for 3 months during autumn.

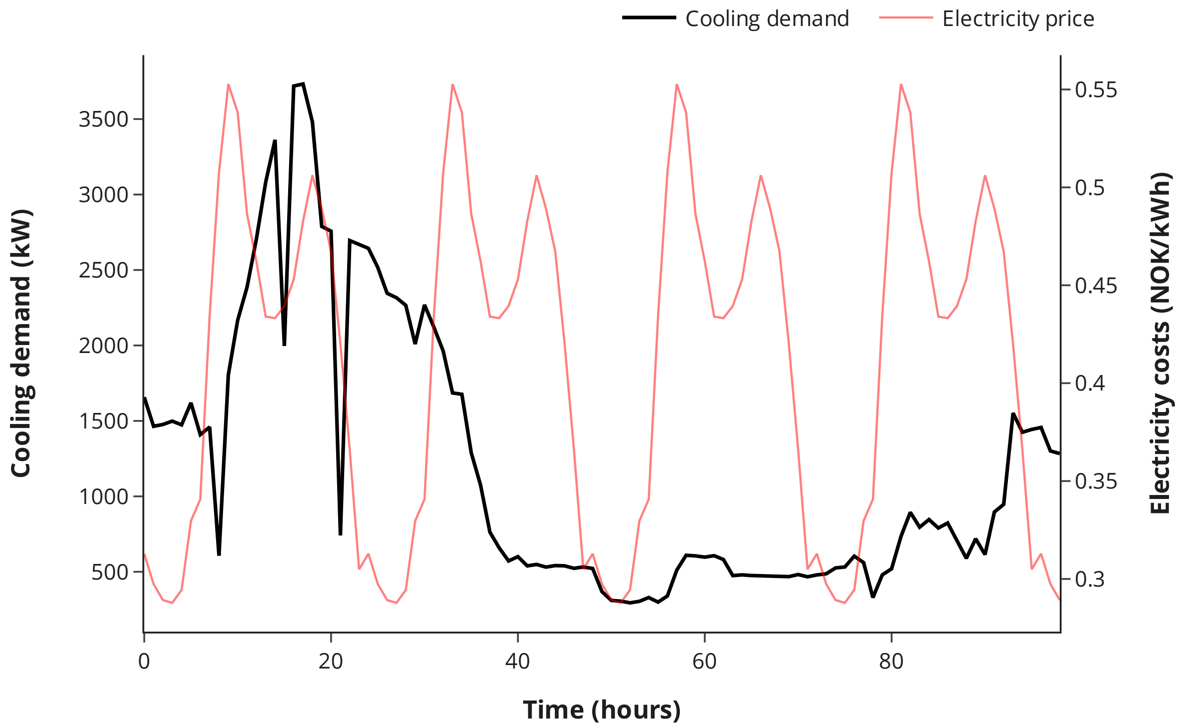

Figure 5 Cooling demand of the freezing tunnels and the electricity price over an illustrative period of 100 h

3. RESULTS

The results from the case simulation (input data presented in Figure 4) are shown in Figure 5. The graph includes the heat flows of the CTES (’TES’ in graph) and the cooling capacity of the refrigeration system (Ref. sys.) designed using the python algorithm. The negative heat flow of the CTES represents the charging, and the positive heat flow the discharging of the CTES. In times of low electricity prices, the refrigeration system should cover the cooling demand of the freezing tunnels. If the cooling demand is lower than the capacity of the refrigeration system, the surplus capacity is then used to charge the CTES which can be seen in the first 5 h of Figure 5. There, the cooling capacity requirement is around 1.5 MW. However, due to the favourable electricity price, the cooling plant is operated with a cooling capacity of around 3.7 MW. This means that the TES can be charged with about 2.2 MW. Given the favourable electricity price of 0.3 NOK/kWh, the full cooling capacity of 4 MW is not used since both the charging and discharging heat flow in the TES is limited to 2.17 MW. As can be seen around hour 20, the TES also reaches its limit when discharging. Therefore, the storage is not sufficient to completely cover the high cooling demand of approximately 2900 kW at a high electricity price period of 0.51 NOK/kWh. In this case, as much cooling capacity as possible is provided with the help of the CTES and the remaining cooling demand is provided with the help of the refrigeration system which operates at a reduced capacity. In the 100-hour period under consideration, the cooling demand was always below the maximum discharge capacity of the TES at a peak price of 0.55 NOK/kWh. Therefore, these particularly costly periods could always be fully covered by the TES. If this had not been the case, the financial savings that could be achieved with the help of the storage system would probably have been lower. Based on the operation of the TES and the refrigeration system as shown in Figure 6 the energy cost reduction would amount to 10.9 % compared to the current mode of operation without a TES. This corresponds to a reduction in energy costs of NOK 3127 from NOK 28586 to

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NOK 25459. Based on the inputs, the simulation tool suggests a storage with a capacity of 5.28 MWh, translating to 4 modules/containers. Each container has a capacity of 1.32 MWh and consists of 55 plates with plate pitch of 27.6 mm.

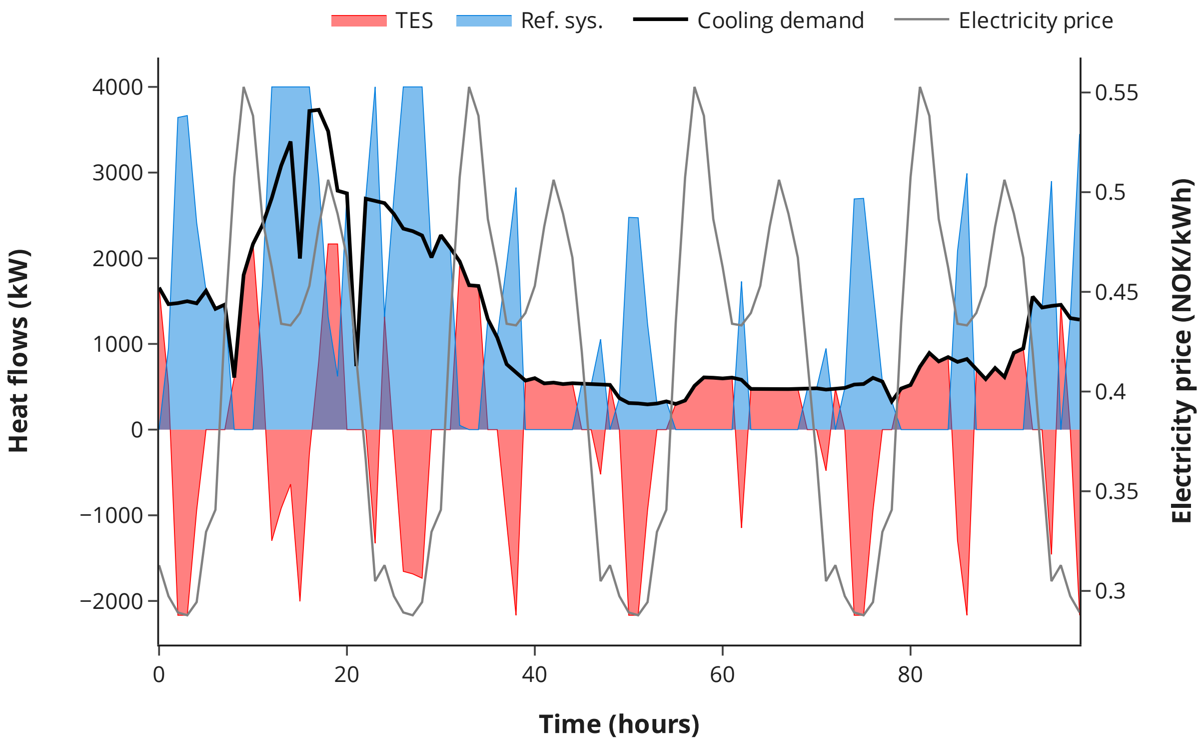

Figure 6 Thermal load of the freezing tunnels, heat flow of the CTES, the cooling capacity of the refrigeration system and the electricity price over an illustrative period of 100 h

Figure 7 shows the state of charge (SOC) of the CTES over a period of 100 hours. There are four complete charge and discharge cycles during the period considered, with the SOC fluctuating between 0 % and 100 %. In the period between 15 and 20 hours, the storage was not fully charged because the cooling demand was very high and the electricity price started to rise again. Therefore, to keep energy costs low, the load on the cooling system was reduced and the CTES started to discharge.

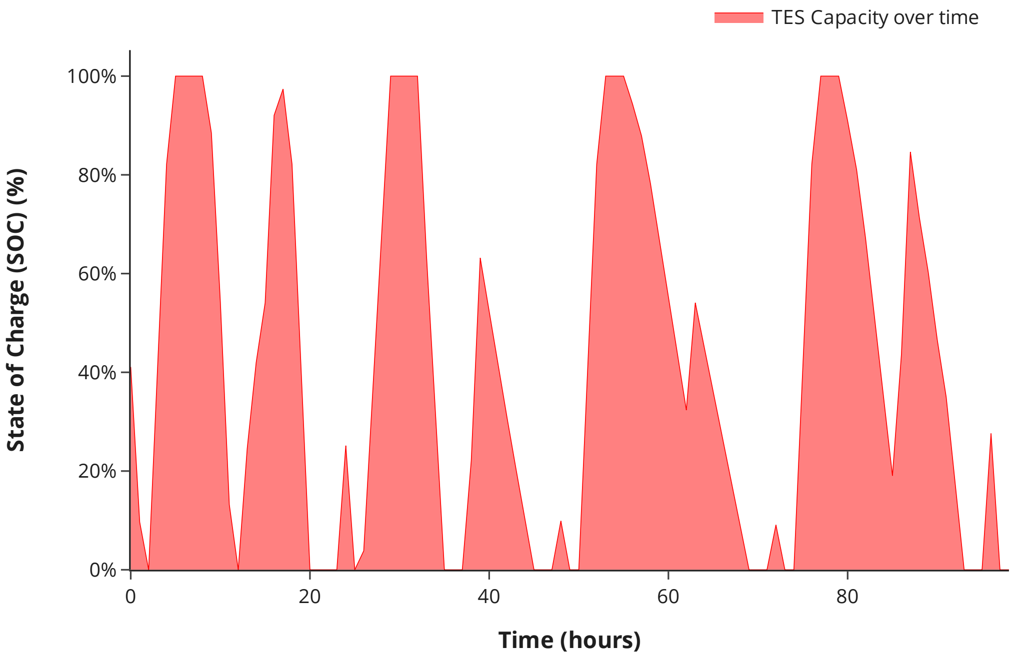

Figure 7 State of charge of the CTES over the period of 100 h

4. CONCLUSION AND FURTHER WORK

In this paper, a python algorithm was presented that uses an optimization-based method to select and dimension a cost-efficient thermal energy storage (TES) based on energy data. The aim was to simplify the implementation of TES in industrial processes to drive decarbonization forward. In particular, cold thermal energy storage (CTES) using a PCM as a storage medium are not yet very widespread but will play an increasingly important role in the future, especially in the food industry. The CTES unit stores thermal energy when electricity is cheap and uses this stored energy to shift cooling loads that occur during peak price periods. To illustrate how the algorithm works, it was run on the cooling demand of an ammonia refrigeration system and the average hourly electricity price from the region over three months in the fall from the pelagic fishing industry. These cooling demand occurs in tunnel freezers during the freezing of pelagic fish at an evaporation temperature of -40°C.

The storage system, suggested by the algorithm will be modular and consist of four 20-foot containers with a combined storage capacity of 5.3 MWh. This is distributed equally among the 4 containers. Each container holds 55 pillow plates which are stacked with a plate pitch of 27.6 mm in which the PCM is located. As a phase change temperature of the PCM a temperature of -45 °C was considered. Based on the operation of the TES and the refrigeration system, the energy cost reduction would amount to 10.9 % compared to the current mode of operation without a TES. Since in the period under consideration all peak price periods could be covered by the CTES due to the low cooling demand at that time, the energy cost reduction may be lower when looking at a longer period (e.g. one month or one year). Therefore, for better reliability on the results, the tool should be operated with seasonal data in the future.

There are still some functionalities that remains to be added to the python algorithm, partly to improve the optimization model and partly to make it more user friendly. For the former, the optimization does not take demand charges into account (peak power fines), i.e. the resulting optimal operation can lead to higher peak powers than the base case. In reality, this would in many cases (dependent on demand charge structure) lead to an increased demand charge from the grid operator, and there exists an optimum balance between this charge and the electricity cost. Furthermore, at present stage the likely adverse effect of charging at reduced evaporation temperature is not reflected in the energy cost calculations which will be implemented. As for user-friendliness, the goal of this tool is to rapidly perform the feasibility of TES integration in general (i.e., flexible with respect to cases). Several features are considered for this, like including capital cost (CAPEX) prediction, an evaluation of the environmental impact of the CTES integration and automatic selection/suggestion of PCM.

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NOMENCLATURE

COP	Coefficient of performance	i	Number of iterations
CTES	Cold thermal energy storage	j	Times of rescaling
PCM	Phase change material	L	Length
PPHX	Pillow plate heat exchanger	$n_{Storages}$	Number of storages
TES	Thermal energy storage	n_{Plates}	Number of plates
		$Q_{capacity}$	Energy capacity
dT	Temperature difference	$pp_{thickness}$	Pillow plate thickness
E_{lat}	Latent heat capacity	V_{PCM}	PCM Volume
E_{sens}	Sensible heat capacity	W	Width
f	Fraction of the plate thickness	η_T	Effective temperature range

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