A NOVEL PASSIVE DEFROST SYSTEM FOR A FROZEN RETAIL DISPLAY CABINET WITH A LOW EVAPORATOR

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ABSTRACT

An energy efficient thermo-siphon method of defrosting the air coils on a commercial half glass door (HGD)/well retail display cabinet has been developed (FrigescoTM) and the performance compared with the existing electric defrost system under EN ISO 23953 test room conditions.

Previous work by (Foster et al, 2013) used a passive thermo-siphon to defrost the top evaporator inside the top (glass door) section and a pump assisted thermo-siphon to defrost the well section. This was due to the head being too low to adequately defrost the well evaporator with a passive thermo-siphon.

This work describes a passive thermo-siphon with no pump. To enable the thermo-siphon to operate efficiently the design of the evaporator was optimised. The thermo-siphon heated quicker and melted water faster than the electric defrost. The thermo-siphon used less electrical heat and had an added benefit of free sub-cooling.

1. INTRODUCTION

To maintain food at acceptable temperatures, both frozen and chilled refrigerated cabinets run their evaporative coils at temperatures less than 0°C. Because of this they need to defrost at regular intervals to remove any ice build-up. With chilled cabinets this can usually be achieved with an off-cycle (or passive) defrost, where the refrigerant flow is stopped and the evaporator allowed to warm naturally to above 0°C, melting the ice. With frozen cabinets this is not possible, as it would be extremely slow and cause the food to defrost.

The two most common defrost methods used in supermarket applications for defrosting frozen cabinets are:

1. Resistive electric heater imbedded in or at the edges of the evaporator.
2. Hot refrigerant gas from the compressor or receiver is diverted into the evaporator.

Electric defrost heaters use a significant amount of energy. Due to the inefficiency of getting heat from the defrost rods to all of the iced fins, much of this energy goes into the cabinet (overhead), rather than into melting the ice. Lawrence and Evans (2008) found the overhead to be around 85% of the energy for a 2.5 m frozen food well display cabinet at climate class 3 (temperature of 25 °C and relative humidity of 60%). This extra heat warms the product and needs to be removed by the refrigeration system. Therefore, the defrost has direct electrical energy from the resistive heaters plus an indirect refrigeration energy required to remove the extra heat. The cabinet often has to run at a reduced set point to allow for the increase in product temperature during the defrost, increasing refrigeration energy consumption.

Fricke and Sharma (2011) estimated the total electrical energy consumption for the electric defrost of a low temperature glass doored reach-in case to be 176 kWh/(year ft) or 577 $kWh.m^{-1}$ p.a. This energy consumption includes both the direct energy associated with operating the defrost heater and the energy consumed by the compressor to remove the excess defrost heat from the display case.

Gas defrosts have the advantage that they are heating the refrigerant pipes directly, evenly and the maximum temperatures are more limited than electric defrosts. During a hot gas defrost, refrigerant gas is taken from the compressor and passed through the evaporator. In both cases the gas condenses in the evaporator giving up heat to melt the ice. The advantage with this system is that the defrosts are fast and efficient. The low temperature of the heat, compared to electric defrosts results in less steaming. However, certain problems are associated with hot/cool gas defrosts. These include extra piping and valving, thermal shocks caused by rapid temperature changes which can cause pipes to leak and the need for head pressures to be high to force the gas through the pipes to the evaporator.

The Kramer-Trenton Company patented a heat bank defrost (Thermobank method), where the discharge of the compressor heats a water store (Dossat). During a defrost the heat from the water bank is used to re-evaporate the refrigerant condensed in the defrosting evaporator.

Previous work by (Foster *et al*, 2013) used a passive thermo-siphon to defrost the top evaporator inside the top (glass door) section and a pump assisted thermo-siphon to defrost the well section. This was due to the head being too low to adequately defrost the well evaporator with a passive thermo-siphon. The same system has been employed on a walk in freezer (Campbell et al, 2014).

This paper describes a novel phase change material (PCM) thermo-siphon defrost system (Frigesco™) which is attached directly to the suction and liquid pipes of a refrigerated display cabinet and does not require the compressor or a pump to run during the defrost. It uses the exchange of heat from the liquid line to the evaporator during a defrost to sub-cool the liquid line, allowing a reduction in refrigeration energy from the defrosts, as opposed to an increase, with other defrost systems.

2. EXPERIMETAL METHOD

2.1. Cabinet

The cabinet tested was an Epta XE0046 2.5 m long remote half glass door (HGD) and well frozen cabinet. The cabinet had a single refrigeration system for both the HGD and well. The evaporator lay in the base of the well, air was ducted up the rear of the cabinet and discharged into the HGD section at the top front. Air was returned at the bottom front and into the well section at the top rear. This air was returned back to the evaporator at the top front of the well. The evaporator was fed with R404A refrigerant from a remote refrigeration compressor. The expansion device was a thermostatic expansion valve (TEV). Two fans at the front (upstream) of the evaporator forced air through the evaporator and around the cabinet.

The cabinet had an electric defrost system which consisted of 3 resistance heater elements, one at the front (upstream) and two at the rear (downstream) of the evaporator. Fans were off during the defrost.

The supermarket settings for the cabinet were 2 defrosts per day (every 12 hours) with a minimum and maximum defrost time of 20 and 45 minutes respectively and a termination temperature of 5ºC. However, for the electric defrosts tests the termination temperature was increased to 10 ºC as 5ºC was not high enough to clear all the ice.

2.2. Thermo-siphon defrost

The novel defrost consisted of a heat exchanger (HE) placed within a stainless steel tank containing a phase change material (PCM) (Puretemp 15, Entropy Solution Inc.) with a melting point of 15ºC and a heat storage capacity of 182 $kJ.kg^{-1}$. The PCM was designated as 100% Bio-based (composed of agricultural, forestry or marine ingredients) by the US Department of Agriculture (USDA). During normal running, liquid refrigerant from the condenser passed though the HE and melted the PCM, sub-cooling the refrigerant before it passed into the cabinet evaporator (Figure 1a). During a defrost, valves were actuated such that the cold evaporator and the HE formed a closed loop (Figure 1b). To instigate an effective thermo-siphon an appropriate sequence and duration of valve opening and closing was determined. The evaporator was now fed by refrigerant gas from the HE. This hot gas condensed in the evaporator, heating it. The liquid from the evaporator drained naturally (due to the higher height of the evaporator) to the HE, solidifying the PCM. The thermal capacity of the PCM was such that a temperature gradient was formed between the HE and the evaporator allowing a thermo-siphon to exchange heat between the two.

When the ice on the evaporator was melted and the PCM was solidified, the valves returned the system to normal operation. Refrigerant was pulsed back to the suction of the compressor at a rate which avoided liquid/compressor issues. The PCM now started to melt again, and when melting was complete a defrost could again be activated.

The PCM HE was placed underneath the cabinet, with a height of 190 mm between the top of the PCM and the bottom of the evaporator.

The length of the defrost cycle was adjusted to allow the PCM to fully solidify and thus exchange all its latent heat with the evaporator during the defrost. The number of defrosts per day were increased to 6, to make best use of this sub-cooling effect. The defrosts were terminated by time, with a 35 minute duration.

As there was less overall heat during the defrost, it was necessary to heat the base plate to a temperature above 0ºC. To do this a temperature controlled heater mat (0.5 kW) was fitted to the base plate and a heater tape (0.2 kW) run along the drain channel and the return grille. These heaters were activated only during the defrost period.

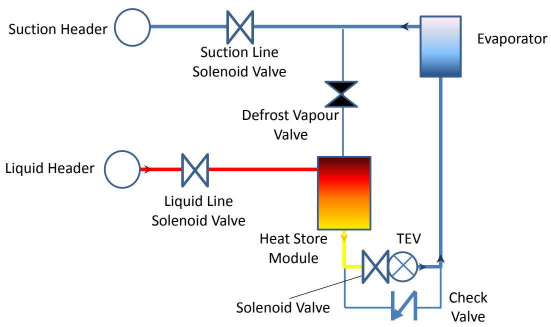

Figure 1a. Schematic of system on charge.

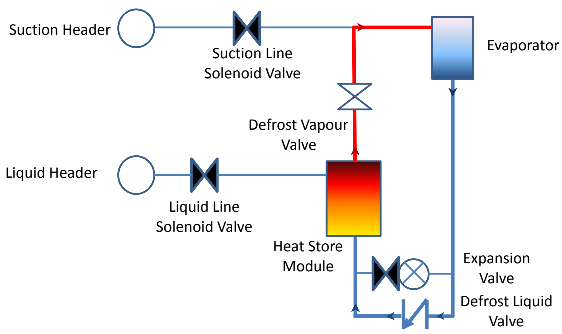

Figure 1b. Schematic of system during defrost.

2.3. Evaporator

The original evaporator was of dimensions 2.06 x 0.144 x 0.4 m. It was 8 tubes (5/8 inch) deep and 3 tubes high with 3 circuits. Fin spacing was 7.5 mm. The evaporator sat on the base of the cabinet which was at angle of 5º, such that water from defrosts drains to the front of the cabinet and out of the drain hole (Figure 2).

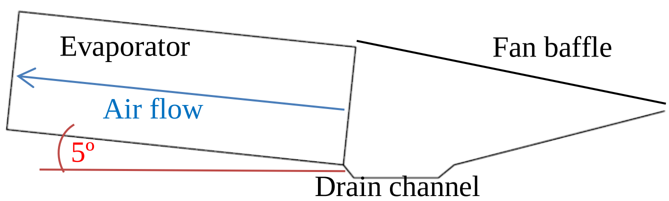

Figure 2. Vertical section of original evaporator layout.

During thermo-siphon, the refrigerant gas entered the front of the evaporator (fan end) and liquid exited at the rear. Due to the angle of the base (5º), the exit was higher than the entrance. It was necessary to slope the evaporator upwards by 10º by raising the front to allow the liquid refrigerant to flow downwards during thermo-siphon (Figure 3). A baffle was placed underneath the evaporator to stop air by–passing the evaporator and a new fan baffle was made which angled the fans more to the vertical than previously.

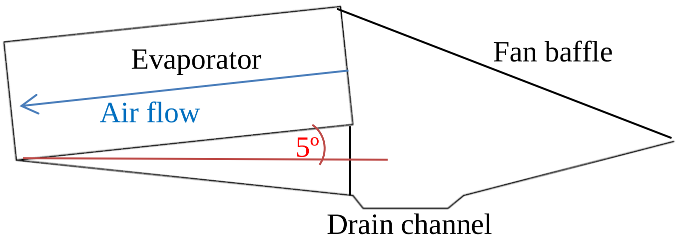

Figure 3. Vertical section of evaporator after tilting upwards.

The thermo-siphon was slower than hoped, due to a perceived large pressure drop caused by the distributor pipes, restricting the mass flow of the thermo-siphon. A gas defrost header (GDH) was added to the evaporator. This system is used in hot gas defrost systems to increase the rate of defrost and stopping condensed refrigerant logging in the coil, preventing even defrosting. A schematic of a GDH evaporator is shown in Figure 4 with flow direction as during a thermo-siphon defrost. The returning liquid by-passes the distributor and flows to the PCM/HE through the check valve.

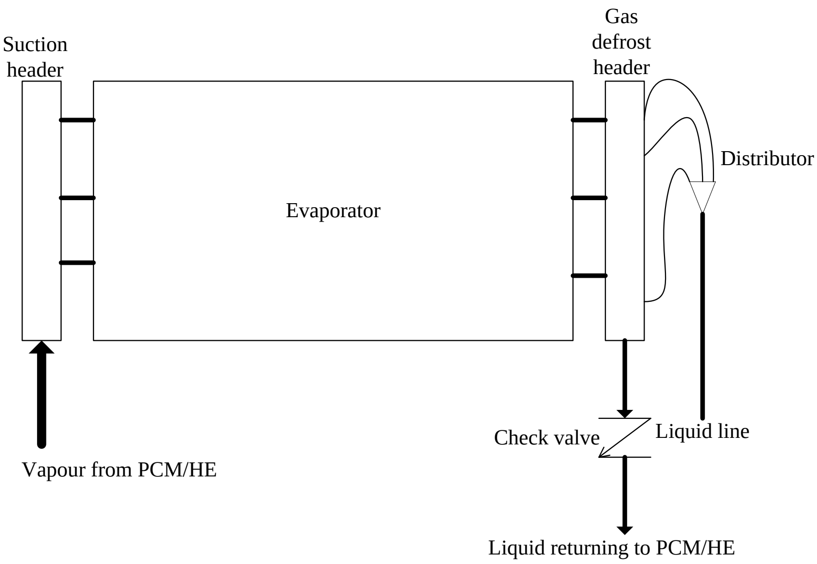

Figure 4. Schematic diagram of evaporator with gas defrost header (GDH) Arrows show direction of flow during a defrost.

2.4. Instrumentation

Weighing scales were positioned under the defrost drain pipe. The weighing scales contained a pump and controller/timer, such that the water was pumped out of the weighing scales 1 hour after the defrost had finished. The mass of water was logged every 20 s.

Temperatures of the evaporator and liquid refrigerant into and out of the PCM/HE were measured to an accuracy of ±0.5°C using calibrated ‘t’ type thermocouples. Thermocouples were strapped tightly to the ends of the evaporator and the pipes into and out of the PCM/HE and recorded every 1 minute

A power meter (Northern Design, MultiCube) was connected with the stabilised mains electrical supply (230 V) to monitor and record electrical power consumption of all parts of the cabinet except the remote refrigeration system (lights, trim heaters, defrost heaters, controllers, solenoid valves, tray heaters).

The mass flow rate of refrigerant was measured using a calibrated Coriolis mass flow meter (Krohne Optimass) with an accuracy of ±0.1%. This was on the liquid line (sub-cooled) upstream of the cabinet.

2.5. Test

The half glass door section of the cabinet was loaded with Tylose test packs and loaded according to EN23953:2005.

Tests with both the traditional (electric) and thermos-siphon defrost system were carried out in a test room conforming to EN23953:2005 standards. During the test, the room conditions were maintained within climate class 3 (25°C and 60% RH).

The condensing pressure was controlled using a condensing pressure regulator. The liquid temperature was maintained to approximately 25ºC to adequately melt the PCM between defrosts.

3. RESULTS

Figure 4 shows mass of water during both electric and thermo-siphon defrost. Water exits the drain pipe 7.6 minutes earlier for the thermo-siphon compared to the electric defrost. The thermo-siphon in this experiment removes slightly more water than the electric; however, this is due to experimental irreproducibility.

Figure 4 also shows the electrical power during the defrost. There was a base power of 1000 W which included trim heaters and lights. During both defrosts (for a time of 23 minutes), the heater mat was activated. For the length of the entire defrost the heater tape was activated. For the electric defrost there was an extra power of 3.65 kW for a period of 23 minutes. The thermo-siphon used 1.3 kWh during a defrost.

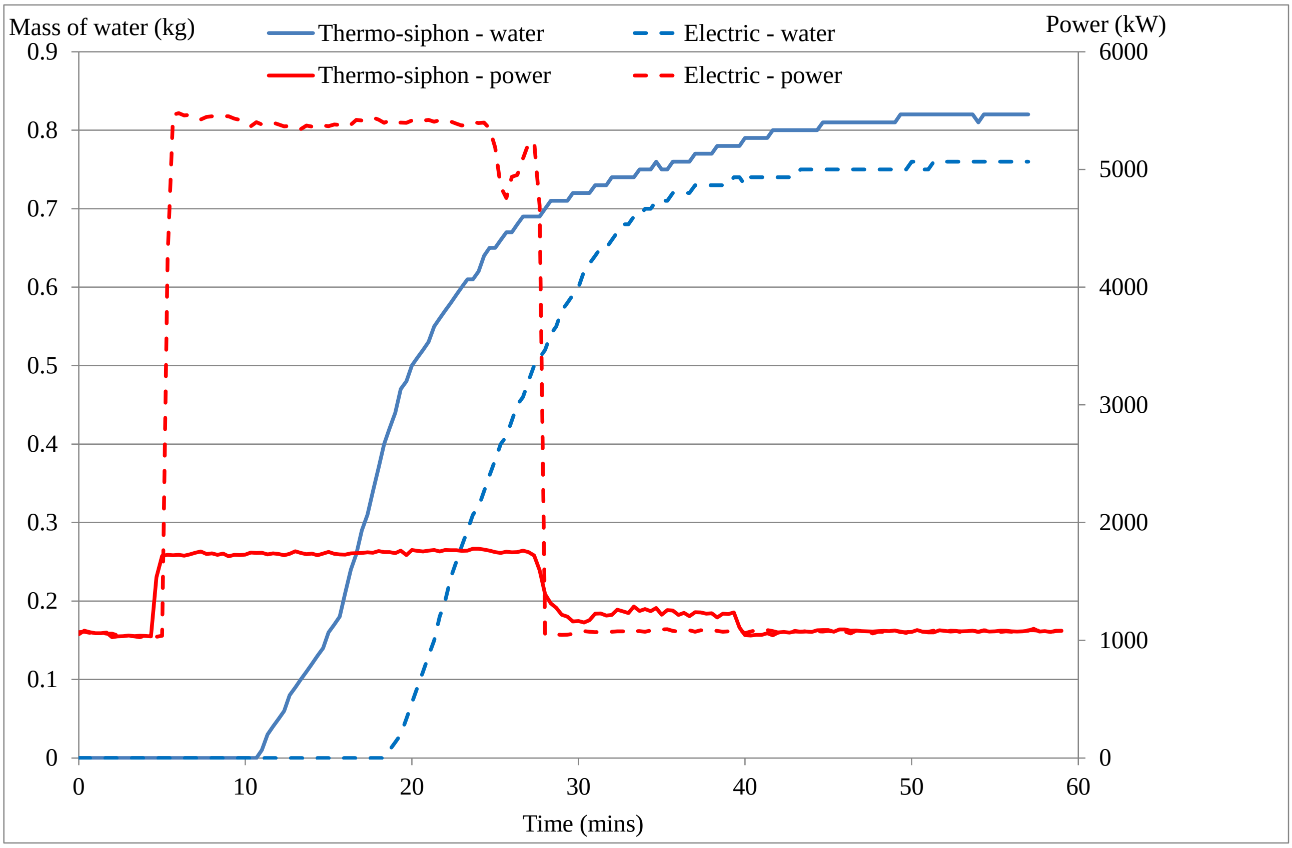

Figure 4. Mass of water exiting the defrost pipe for both the thermo-siphon and electric defrost.

Figure 5 shows temperatures at the end of the evaporator at the fastest and slowest positions to heat up during a thermo-siphon defrost. The fastest position was the top front of the evaporator and the slowest was the bottom rear. The front was the entry of the thermos-siphon and the rear the exit.

Temperatures started to rise sooner for the thermo-siphon than the electric defrost, reaching 0ºC, 10.5 minutes quicker at the front and 8 minutes earlier at the rear. The front of the coil reached 0ºC, 6.5 minutes quicker than the rear for the thermos-siphon defrost. The temperature during electric defrosts continued rising higher in the electric defrosts, reaching a maximum of 22.5ºC during electric compared to 11.1ºC for the thermo-siphon, both at the front.

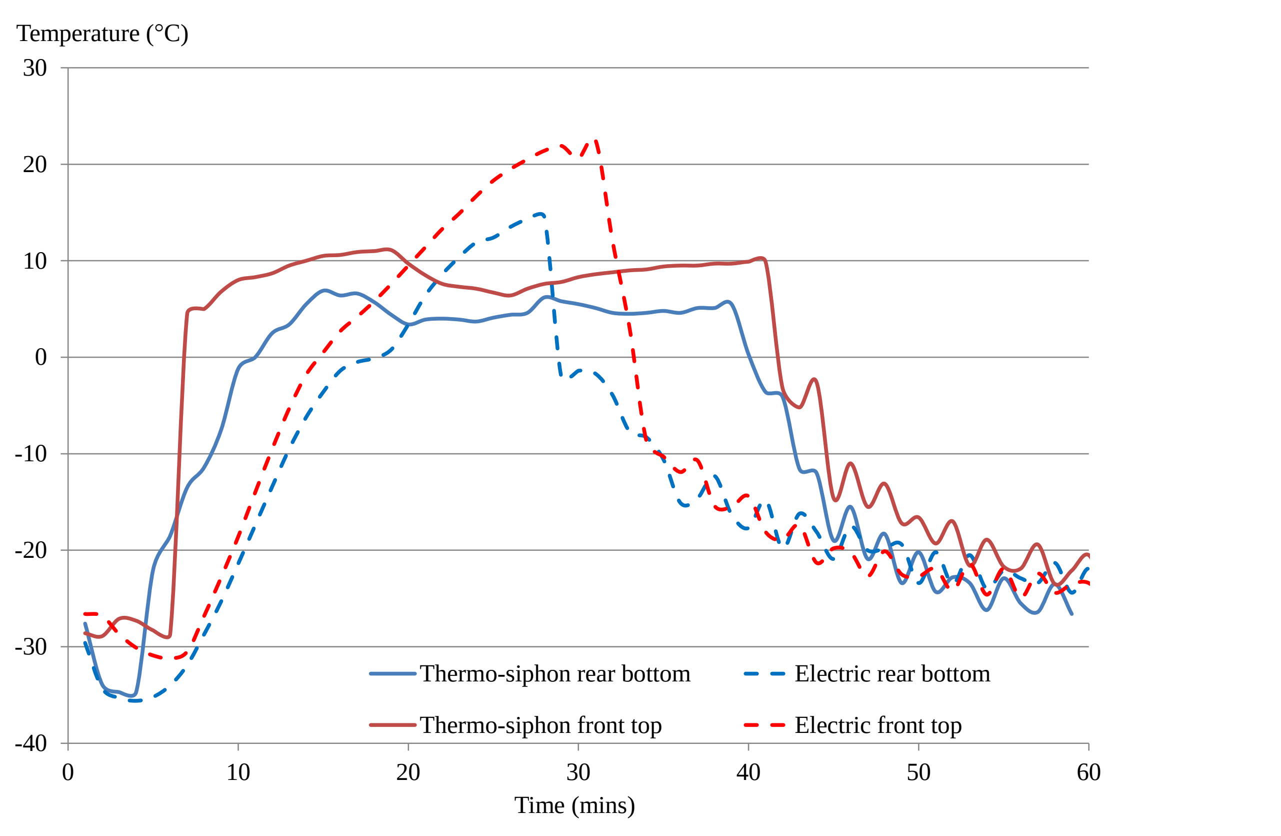

Figure 5. Temperatures at the end of the evaporator at the fastest and slowest positions to heat up during a thermo-siphon defrost.

Figure 6 shows the temperature of the liquid refrigerant into and out of the heat store, the subsequent level of sub-cooling and the mass flow. Based on an average mass flow rate of 14.4 $g.s^{-1}$, sub-cooling of 6.2 K and a specific heat capacity of liquid R404A of 1.5 $kJ.kg^{-1}.K^{-1}$over the period between defrosts, we can calculate an average reduction in cooling duty of 134 W between defrosts, which equates to 0.446 $kW.h^{-1}$ per defrost. At 6 defrosts per day, this would equate to 2.68 $kW.h^{-1}$.

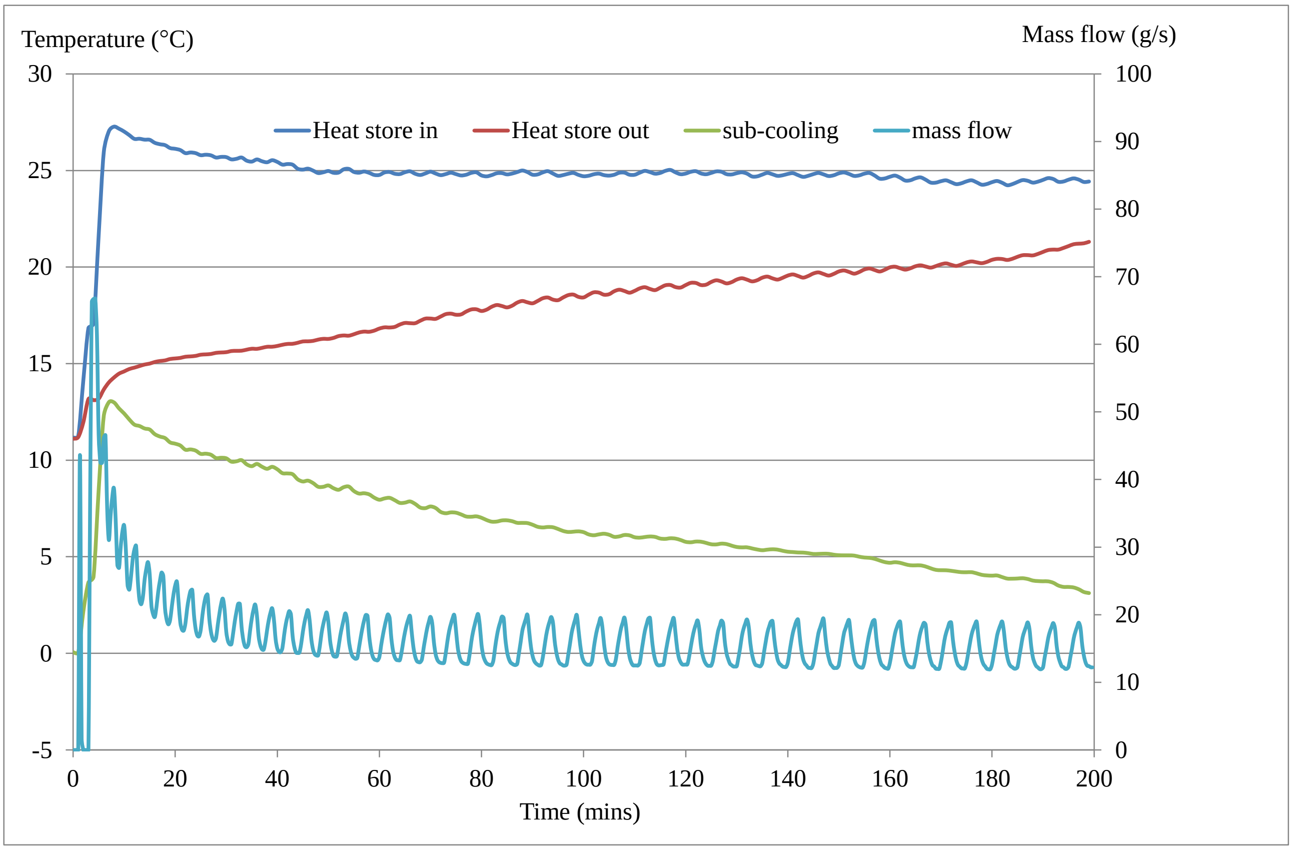

Figure 6 Temperature of the liquid refrigerant into and out of the heat store, the subsequent level of sub-cooling and the mass flow.

4. CONCLUSIONS

A novel thermo-siphon defrost has been shown to be able to defrost a refrigerated display cabinet using heat stored from the liquid line passing through a heat exchanger immersed in a PCM.

Faster heating of the evaporator resulted in water melting more quickly.

The benefits of the thermos-siphon are;

- it does not require electric defrost heaters (particular benefit with hydrocarbon refrigerants)
- uses less electrical energy
- uses less refrigeration energy due to sub-cooling
- higher duty (due to sub-cooling) directly after defrost allows a quicker reduction in product temperature.

Further work is required to test the overall energy savings of the system. It is expected than an extra saving on refrigeration energy is produced due to the more efficient defrost.

A positive consequence of the inefficient high temperature electric defrost is that more heat is available to melt ice away from the evaporator. With the thermo-siphon defrost, methods need to be employed to make sure all ice is removed from the cabinet between defrosts.

The work needs to be repeated with other refrigerants which have a longer life regarding the F-gas regulations (EU, 2014), e.g. R407. Parallel studies on a cold store have shown good results with R407F.

5. REFERENCES

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