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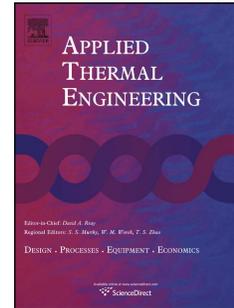
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A thermographic study of the on-off behavior of an all-refrigerator

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1. Abstract

In this work a thermographic camera is used to observe the temperature distribution of a household refrigerator cooling system operating at on-off cycling conditions. This technique offers an alternative method to analyze the cooling system compared to conventional thermocouples. In particular it is interesting to view the overall picture of how the refrigerant charge is distributed over the cooling system at transient conditions. In additions, four sources of energy losses were indentified and discussed. Out of these losses, two would have been difficult to find using conventional thermocouples temperature measurements.

Keywords: household refrigerator; thermography; heat exchanger; cooling system

2. Introduction

The global annual production of household refrigerators and freezers is more than 80 million units. With an expected lifetime of 10 to 20 years the number of units in use is enormous and therefore their impact on the global energy consumption significant. It follows that domestic refrigeration is a natural target for energy consumption controls in the EU and in many countries around the world.

Most studies that have expanded the knowledge of the working principle of small, on-off controlled, capillary tube throttled, vapor compression cooling systems (i.e. household refrigerators & freezers) are based on thermocouple recordings. Useful experimental data have for instance been provided by Wang and Wu [19], Janssen et al. [12], Rubas and Bullard [18], Jakobsen [11], Krause and Bullard [15], Coulter and Bullard [7] and Björk and Palm[5]. Experimental charge inventories were reported by Kuijpers et al. [13-14], Mulroy and Didion [16], Björk and Palm 2006 [3-4]. X-ray was used by Inan et al. [10] to demonstrate the refrigerant movement in selected parts of a household freezer cooling system. Björk [1] visualized the refrigerant flow in a household refrigerator using a transparent glass lid evaporator.

Evidence of using thermography in household refrigeration is sparse. Cristalli et al., [8] proposed an online method based on thermography to identify manufacturing and assembling defects during production of household refrigerators. Herolf [9] used an infrared camera to visualize the air temperatures in a vertical cross section of the cabinet.

In this study the results from an unusual experimental run is presented. A household refrigerator cooling system is observed using an infrared video camera during normal

on-off cycling operation. The purpose is to assess an alternative technique for analyzing the transient behavior of household refrigerators.

2.1 The typical refrigerator/freezer

Household refrigerators and freezers typically operate by the vapor compression cycle. They are characterized by low cooling capacities (50–250 W), low quantities of refrigerant charge (20–200 g), a refrigerant accumulator located at the evaporator outlet, a hermetically sealed cooling system, a capillary tube expansion device which is in heat exchange with the suction line, and a cooling capacity control by on-off control.

2.2 The on-off cycling

The most common way to control the cooling capacity in household refrigerators and freezers is by on-off control (cycling or intermittent operation). The compressor starts as a preset temperature in the refrigerated space is exceeded (cut-in temperature) and shuts down as a low temperature (cut-out temperature) is reached. A typical course of events at compressor start and stop is as follows: At the moment before the compressor starts, the evaporator is at its highest temperature. Most refrigerant, by mass, is located in the evaporator and compressor whereas the condenser and the capillary tube contain only superheated gas. As the compressor starts, the initial compressor mass flow is high (high evaporation pressure) whereas the capillary tube mass flow is low (superheated gas). The result is that refrigerant is displaced towards the condenser which may starve the evaporator with refrigerant liquid. As a result the

evaporator pressure decreases thereby lowering the compressor mass flow. A liquid pool is formed at the condenser outlet which increases the mass flow through the capillary tube. Now the evaporator “refilling” begins. This means that the dry-out point (position in evaporator where superheat starts) moves downstream gradually ‘activating’ the evaporator as the wetted area, active for evaporation, increases. It is important that this process is short in comparison to the compressor on-period since the system during this time period operates inefficiently due to the superheat and lowered evaporation temperature.

When the cut-out temperature is reached the compressor stops and the pressure equalizes within the system as refrigerant continues to flow through the capillary tube. The condenser pressure decreases and slowly approaches the saturation pressure in the evaporator while refrigerant is forced into the evaporator. During this migration process it is important that the refrigerant liquid is freely drained towards the condenser outlet. If pockets of liquid remain in the condenser as the capillary tube inlet liquid seal is broken a reversed heat transfer follows, where liquid in the condenser evaporates and condensation occurs in the evaporator.

The charge displacements at startup and shut-down affect the performance negatively. Totally, efficiency losses of 5–37% have been reported [3,7,11,12].

3. Methods

A single compartment refrigerator with natural convection (passive) heat exchangers was studied using thermocouples and an infrared camera. At steady cycling conditions

(each on-off period at similar length) the variations in the surface temperatures of the evaporator and condenser were recorded. The temperatures of the lower half part of the condenser were also separately recorded from close range to give a more detailed picture. The experiments were carried out in the following way: At 25°C ambient temperature and the refrigerator thermostat set to warmest possible (averaged air temperature in the cabinet 7.2 °C) the refrigerator was running until it was steadily cycling. Then the evaporator was filmed during one on-off period. Next, the video camera was repositioned to capture one on-off period of the whole condenser and thereafter the lower part of the condenser. Finally, the three video recordings (evaporator, overall condenser and condenser close-up) were synchronised so that images of the evaporator and condenser could be viewed side by side displaying the same moment in the on-off period. This was done by synchronising the condenser films to the evaporator film. They were synchronised at two moments in the on-off period; the moment when the compressor started, and the moment when the compressor stopped. This means that the images showing the evaporator and condenser comes from similar, but not the same, on-off period. However, the temperature recordings and the evaporator films come from the same on-off period. Figure 1 shows an example of the synchronised videos.

Figure 1 Images from the infrared videos captured five minutes after a compressor start. From the left: evaporator, the entire condenser and a close-up of the lower half of the condenser. The compressor is seen as a large warm object at the bottom of the condenser.

3.1 Test object

The test object was an Electrolux ER8893C, single door and single compartment refrigerator with the following data (see also Figure 2):

- **Cabinet:** (1.75×0.6×0.6 m), 0.350 m³ internal volume, UA value 2.3 W/K (experimentally measured by a reversed heat leakage test).
- **Evaporator:** free convection, (0.66×0.49×0.0014 m), aluminium, plate type, back-wall located (20 mm distance to wall), integrated downstream located accumulator. Refrigerant line length (including accumulator) 6.22 m. Hydraulic diameter 3.2 mm. Total internal volume 116 ml whereof accumulator volume 46 ml in which approximately half the volume can store liquid at steady state condition. UA value estimated to 3.7 W/K. Weight 1.35 kg.
- **Condenser:** free convection, (1.33×0.51×0.008 m), steel, wire on tube (53 vertical wires on each side of the tubing, each of diameter 1.5 mm) located with 25 mm distance to the cabinet back wall. Refrigerant flow horizontally downward. Internal volume 135 ml. Internal/external diameter 3.5/5.0 mm. UA value estimated to 7.7 W/K.
- **Capillary tube:** (2.54 m length and 0.6 mm internal diameter) with coaxial type suction line heat exchanger of 2 m length. The capillary tube adiabatic inlet and outlet sections are 0.5 and 0.04 m respectively.
- **Filter dryer:** molecular sieve with internal free volume 11.3 ml.
- **Piston compressor:** (Electrolux HL 60AH) with low pressure oil sump.
- **Refrigerant:** 36 g of Isobutane (R600a)
- **Capacity control:** by intermittent run with self-defrosting in every off-period

Figure 2 Cooling system and thermocouple locations

In Figure 2 the overall cooling system is displayed in a schematic view together with the thermocouple locations.

3.1 Modifications to test object

The IR-camera used was an AGEMA Thermovision 400 with a 40 mm/40° lens. To avoid thermal reflections influencing the IR camera, the evaporator and condenser were painted dull black. This modification did not change the refrigerator behaviour in any noticeable way.

This type of refrigerators that has natural convection condensers located at the back of the refrigerator is optimised in terms of refrigerant charge and capillary tube capacity for a condition where they are placed against a wall [5]. The reason is that the “chimney” effect enhances the air flow across the condenser in this position.

However, this would make IR recording of the condenser difficult. Instead, during the experiments a thin transparent plastic film, of the type used for food storage, was stretched over the cabinet’s back side forming a channel with geometry equal to that formed when the refrigerator is positioned against a wall. Through this, a chimney for the condenser heat was formed while at the same time IR recording of the condenser was enabled. It was experimentally verified that the system behaviour was similar to the wall positioned refrigerator. The error by filming the condenser through the plastic film was measured to be less than 0.1 K (by quickly removing and inserting the plastic film).

Due to the rather narrow angle 40° lens, the camera needed to be at some distance to capture the entire evaporator. Therefore an inspection opening was made in the refrigerator door and some insulation material was used to locate the camera at the

needed distance from the evaporator. This is why the insulation material sticks out somewhat from the door (Figure 3). Even though this opening was as much as possible insulated, an increased heat leakage was unavoidable. The result of this was a prolonged cycle length in comparison to the unmodified system. The on-period was extended from about 700 s to about 1000 s, the cycle length from about 2700 s to about 3200 s.

The experimental set-up with the IR-camera pointing through the inspection opening in the door can be seen in Figure 3 and Figure 4. Temperatures were measured by thermocouples located as seen in Figure 2.

Figure 3 IR-camera, AGEMA Thermovision 400 with a 40 mm/40° lens, in position for recording of the evaporator temperatures.

Figure 4 IR-camera in position for recording the condenser temperatures (close-up recording of condenser lower part)

4. Uncertainties

The estimated uncertainties in temperature were 0.5 K (U=95%) for the thermocouples and 5 K (U=95%) for the IR camera. The difference in time was less than 2% (U=95%) of the different on-off periods. Since the condenser films were synchronised to the evaporator films at the moment when the compressor started and stopped, the largest uncertainty between the two films would be at the end of the on and off-period. In the presented images, the largest uncertainty will occur at 720 s. The uncertainty in time at this moment is about 20 s (U=95%) between the evaporator

and condenser images (calculated as root-sum-square of the individual uncertainties of the two films).

5. Results

Firstly some comments will be made about the thermocouple temperature recordings that, as was mentioned before, are taken from the same on-off cycle as was used for the IR film of the evaporator. After this the thermo graphic images will be presented and discussed.

Figure 2 Temperatures during one on-off cycle. For thermocouple locations see Figure 2.

5.1 Temperature recordings

In Figure 5 the temperatures during one on-off cycle are shown. It is seen that the cycle length is more than 3200 s (3240 s). It is also seen that the on-period is approximately 1000 s (994 s).

Following a startup, the suction line temperature (T8) immediately falls from 25°C to below 10 °C, after which it increases to above the ambient temperature. This temperature dip is caused by liquid refrigerant, accumulated in the evaporator during the off-period, is being drawn into the suction line where it evaporates on its way towards the compressor. This liquid suction, occurring at compressor startup, was previously observed by Rubas and Bullard [18], Krause et al. [15] and Björk and Palm [3]. It is explained by the initially “overcharged” evaporator and the initially high

evaporation pressure that causes the fixed volumetric compressor to operate with a high mass flow.

The condenser temperature (T_3) first rises sharply. It reaches a peak value of just above $40\text{ }^\circ\text{C}$ in the middle of the on-period after which it slowly decreases to $40\text{ }^\circ\text{C}$ just before the compressor stops. This decrease follows upon a steadily decreasing evaporation temperature which lowers the compressor capacity. The condenser outlet temperature (T_4) parallels the condenser temperature (T_3) but at a lower value. This is a clear sign of condenser sub-cooling.

On the evaporator side the inlet temperature (T_5) falls sharply at startup, whereas the accumulator inlet temperature (T_6) is at a higher temperature until almost 700 s when they coincide. This is the time needed for the refrigerant liquid to reach the accumulator inlet thus “activating” the evaporator after the initial starving at the compressor startup. In other words, this is the time needed for the dry-out point to move the distance from the evaporator inlet to the accumulator inlet. The evaporator outlet temperature (T_7) is at a higher temperature than T_5 and T_6 throughout the on-period. This superheat is almost 3 K at the moment when the compressor stops.

In the off-period the condenser temperatures (T_2 - T_4) asymptotically approaches the ambient temperature. The evaporator temperatures (T_5 - T_7) first increase to $0\text{ }^\circ\text{C}$, where they level out while the evaporator defrosts. Then they increase again to slowly approach the cabinet air temperature. It is seen that the defrost time at the T_7 location is shorter compared to T_5 and T_6 . This may be explained with less frost at this location. The temperature at this location is higher during the on-period which means

that less frost is formed here. It can be mentioned that the frost layer was very thin after the on-period (estimated by visual inspection to be < 0.5 mm).

The compressor temperature (T_1) varies between 35 and 45 °C during the complete on-off cycle. It is worth noting, that this thermally heavy component will always stay significantly above the ambient temperature. The heat dissipated from the compressor shell is used to evaporate the defrost water collected in a plastic container on top of the compressor (see also Figure 4).

The time to “activate” the evaporator in the on-period was unexpectedly long. The typical activation time for this kind of refrigerator is less than 180 s. In this case it was about 700 s. The explanations are the high temperature setting in the refrigerator and the increased heat load due to the modified door (inspection opening). Together these two factors give a higher evaporation temperature than under normal operation, thus resulting in an increase of the compressor mass flow. Since the capillary tube has a fixed flow resistance, longer time will be needed to activate the evaporator at startup.

5.2 Thermographic images

Images from the IR video recordings are displayed in Appendix A. On the right side of the images the time is shown referring to the same time scale as in Figure 5. For instance, 0 s is the time when the compressor starts (frame picked out at the moment before any thermal change due to the compressor start can be seen). 994 s is the time when the compressor stops (frame picked out at the moment before any thermal change due to the compressor stop can be seen).

At 0 s the evaporator is isothermal (within the resolution of the measurements). The condenser is also more or less isothermal. The compressor can be seen as a warm object at the bottom of the condenser images. Since this image shows the temperatures at the moment when the compressor starts but just before any thermal change is visible the compressor heat stems from the previous on-period.

At 10 s it is seen that cooling occurs in parts of the evaporator. One can identify the location of the accumulator and the vertical tube at the left side of the evaporator. Obviously this is liquid which is accumulated in the evaporator during the off-period that evaporates as soon as the compressor starts. Apparently, part of this liquid was accumulated in the bottom part of the evaporator and most likely the rest in the accumulator itself.

The suction line, which is seen left of the compressor, is also rapidly cooled. Taking into account that the refrigerant exiting the evaporator has to pass through a two meter long coaxial type suction line heat exchanger (SLHX) - at ambient temperature - before reaching this location it is obvious that some refrigerant liquid is exiting the evaporator. In fact, in a previous work by Björk and Palm [3] it was found from experiments using a refrigerator at similar conditions that about 15 g of refrigerant, or about 40% of the entire charge, was exiting the evaporator during the first 20 s following a compressor start. In the same article it was calculated that a cooling power of 1.7 kJ/cycle was unaccounted for when performing an energy balance over the evaporator and SLHX at startup. A cooling power of 1.7 kJ/cycle corresponds to 4.7 g of refrigerant liquid passing un-evaporated through the SLHX into the suction line

just upstream of the compressor. This would explain the rapid cooling of the suction line observed.

Another observation from the image at 10s is that the compressor discharge pipe is rapidly heated.

At 20 s refrigerant has made the full loop from the evaporator, through the condenser and back to the evaporator again. This can be seen at the evaporator inlet which cools down rapidly. From now on it is possible to see the dry-out point moving downstream gradually activating the evaporator. The accumulator is cold but not as cold as the evaporator inlet. Furthermore, in the following images (30 and 60 s) the accumulator will heat up. Apparently all liquid remaining in the accumulator from the off-period, has already exited the evaporator.

At 60 s sub-cooling is seen in the condenser approximately covering the four lowest horizontal pipes. From 60 to 180 s this sub cooling area decreases. At the same time the dry-out point advances forward in the evaporator. Apparently liquid is being transferred from this sub-cooled part of the condenser to the evaporator. After 180 s the sub-cooled area is no longer changing, and covering approximately the lowest two horizontal pipes in the condenser. This is in agreement with the T.C. temperature recordings (Figure 5) which revealed a constant sub-cooling in the later on-period. Apparently, the liquid needed to further refill the evaporator is not taken from the condenser. Previous studies [3, 12] have shown that this liquid comes from the compressor oil, where it is dissolved during the off-period, and from the low side vapor whose pressure (density) decreases with the decreasing evaporator temperature.

At 720 s the accumulator can still be seen as an area of somewhat higher temperature than the rest of the evaporator. At this time the T.C. temperature recordings (Figure 5) show that the dry-out point has moved into the lower part of the accumulator.

At 994 s, which is the moment when the compressor stops, the evaporator is more or less isothermal. A vague color difference indicates the accumulator which at this time has a 3 K superheat at its outlet (Figure 5). It is possible that the temperature range (lower limit set to $-15.8\text{ }^{\circ}\text{C}$) was selected too high making this superheat difficult to spot. It is also seen that the evaporator edges on the lower left side are at a higher temperature. Apparently the refrigerant channels (see Figure 2) are too distant from the edges. The result is that the evaporator plate is poorly used for heat transfer which leads to lowered system efficiency.

At 1054 s, which is 60 s after the compressor has stopped, the heat exchanger temperatures have changed significantly. It is interesting to note that there is no local heating at the evaporator inlet as a result of refrigerant migrating from the condenser into the evaporator. Intuitively it may feel like “hot” refrigerant liquid coming from the condenser would cause a heating effect. However, any liquid passing the capillary tube will depressurize to the evaporator pressure and enter as a two-phase mixture at the saturation temperature corresponding to the pressure in the evaporator. The result is that the entering vapor increases the evaporator pressure so that some condensation occurs in the evaporator. This heating will not affect the evaporator locally at the evaporator inlet but rather globally all over the evaporator where condensation occurs.

At 1114 s the image shows a higher temperature, evenly distributed, where the refrigerant lines are located. This can be attributed to heating by condensing refrigerant in the channels, as was explained above.

At 1134 s (the last image) a design weakness is revealed in the mid picture. Even though the condenser design admits a free fall towards the condenser outlet to avoid liquid pockets during the off-period, the filter dryer (lower left side) is at a significantly lower temperature than other parts of the condenser. Apparently some liquid is trapped in this device where it is evaporating during the off-period. It was estimated, based on the IR-films and the physical size of the filter dryer, that less than 0.3 kJ of heat was transferred to the filter dryer during the ten minutes period when this cooling effect existed. This heat will add to the cabinet thermal load in the following way. When the liquid seal at the capillary tube inlet is broken so that the capillary tube only contains vapor, any liquid left in the condenser will evaporate and, just like in a thermo-siphon, the corresponding heat of evaporation will be released during condensation at the coldest point in the evaporator. This thermal transport is obviously an energy loss. In previous articles [3,18] this phenomenon was estimated to result in an additional heat load of 2–7%.

6. Discussion

6.1 The thermographic technique

An infrared camera is a fast online method to analyze a temperature distribution. One of the advantages over conventional thermocouples technique is that it gives the full thermal landscape and not only point values. In the present study two sources of

energy losses were found that would not easily have been found using thermocouples (if not, by chance, the thermocouples would have been located at the right spots). Some experiences from the experiments are that the test objects (evaporator and condenser) must be painted dull black to avoid thermal reflections. If possible, a wide angle lens should be used so that the evaporator could be filmed from a location within the cabinet. A thin plastic film is a useful tool to create a transparent “chimney” (see section 3.1). A limitation would be that only plate evaporators are suitable to analyze using IR-technique.

6.2 Identified losses

Four sources of energy losses were identified and will be discussed: 1. The evaporator activation process at startup. That is the time before the evaporator is fully charged with boiling refrigerant. 2. Evaporator edges poorly used for heat transfer during steady state. 3. Evaporation in filter dryer during the off-period. 4. Liquid refrigerant exiting the evaporator at startup.

1. The evaporator activation process at startup.

Firstly, the evaporator studied was remarkably slow to activate at compressor startup (about 700 s). Even if this long time was partly a result of the experimental set-up the phenomenon is typical for household refrigerators. High evaporator superheat in the early on-period has been observed in earlier work [3,7,15]. Björk and Palm [3] estimated this loss to be 3.6% for a household refrigerator similar to the one tested in the present study. A possible way to overcome this weakness and to speed up the activation is to redesign the evaporator refrigerant lines. Carrera [6] showed that evaporator lines in a downward inclination resulted in a much faster evaporator

activation. He also found the same result by decreasing the cross section area or to increase the mass flow by increasing the flow capacity in the expansion device. In another study by Björk and Palm [5] it was found that the energy consumption was independent of the expansion device flow capacity within certain ranges. Thus, it is suggested that a combination of increased expansion device flow capacity and rearranged refrigerant lines with a downward flow could be used to improve the energy efficiency. It should also be possible to reduce the internal refrigerant channel cross section area at the most upstream part of the evaporator without any significant additional pressure drop, as the vapor velocity at this location is low.

2. Evaporator edges poorly used for heat transfer during steady state.

Secondly, the IR-images show that the evaporator's edges were at higher temperatures than the central part of the evaporator. Thus, the edges were poorly used for heat transfer. To overcome this, the refrigerant channels should be moved closer towards the edges. Björk [2] estimated the fin efficiency, calculated as the ratio between actual heat transfer and the heat transfer that would occur if the evaporator was isotherm at the refrigerant channel temperatures, of a similar evaporator to be 97%. This can be recalculated to a temperature loss of about 0.6 °C which corresponds to an efficiency loss of approximately 1.2% (1K added temperature difference is about 2% lowered efficiency for the given compressor).

3. Evaporation in filter dryer during the off-period.

Thirdly, the filter dryer acted as a liquid pocket during the off-period. In this pocket evaporation occurs resulting in a transport of thermal energy into the cooled space. It may be difficult to overcome this problem since the highly porous particles inside the

filter dryer are needed to absorb any water in the system. Assuming that the only water that enters the hermetically sealed cooling system comes during manufacturing, a more controlled manufacturing process would make the dryer (but not the filter) unnecessary. The loss was estimated to be 0.3 kJ/cycle or about 0.4% in efficiency (total cabinet heat leakage 73,6 kJ/cycle).

4. Liquid refrigerant exiting the evaporator at startup.

Finally, the fourth source of energy loss was found at startup where it was observed that liquid refrigerant was exiting the evaporator. The classical solution to this problem would be to design the evaporator accumulator as a liquid separator so that boiling would solely occur in the evaporator. However, a liquid separator may not be easily designed within the dimensions of the plate. Moreover, if such separator was possible to design, it would still be unfavorable to boil refrigerant locally in the evaporator accumulator compared to have it boiling in the entire evaporator. The refrigerant needs to be distributed over the entire evaporator to give a high evaporation temperature and hence a high efficiency. Another solution would be to quickly recharge the evaporator after a startup. It has been found that a relation exist between the exiting refrigerant at startup and the evaporator activation time [5, 18]. With a large quantity of liquid exiting the evaporator at startup the liquid seal at the condenser outlet is quickly restored which permits faster evaporator activation. Moreover, a closer analysis suggests that the problem with liquid refrigerant exiting the evaporator to evaporate in the suction line heat exchanger (SLHX) at startup is not as large as it first appears. Part of this cooling is regenerated back to the cooled space via the SLHX itself. At similar conditions Björk and Palm [3] calculated that the cooling energy accumulated in the SLHX material when exiting refrigerant evaporates

at startup in the SLHX was about 2.0 kJ. This can be compared to 1.7 kJ of cooling energy accumulated in the evaporator material at startup, and the 1.7 kJ that was estimated to be the loss of cooling energy when liquid refrigerant liquid exits the SLHX at startup (passes un-evaporated through the SLHX). In other words, the cooling energy accumulated in the SLHX appears to be an important factor to decrease the cyclic losses. Perhaps a redesign of the SLHX, making the passage of liquid refrigerant at startup more difficult, would be a way to decrease the cyclic losses even more. Such design change should also be beneficial for the compressor that would receive less refrigerant liquid at startup. It is important, however, that the suction line pressure drop not increases with such design change.

7. Summary and Conclusions

Infrared thermography was used as an alternative technique to observe a household refrigerator cooling system during on-off cycling conditions. The experimental technique proved to be useful. It gave a clear overview of the cooling system temperature field. Overall, the experimental results confirm findings from previous research. It is for instance shown that refrigerant is displaced towards the compressor and condenser at startup which results in a starved, superheated evaporator and a drop in efficiency. However, the study also reveals sources to energy losses that would have been difficult to find if the conventional temperature measurement technique (i.e. thermocouples) is used. It is shown that evaporator edges are poorly used for heat transfer and that evaporation occurs in the filter dryer during the off period. The identified losses are discussed and possible means to overcome them are suggested.

It is concluded that:

- The experimental technique is useful to analyze the cooling system behavior. However, it is limited to plate evaporators.
- The observed evaporator is slow to activate at startup. It is suggested to rearrange the refrigerant lines into a downward slope and to decrease the internal diameter at the upstream part of the evaporator.
- The observed evaporator edges are poorly used for heat transfer at steady state conditions. It is suggested that refrigerant lines are moved closer to the plate edges.
- Refrigerant evaporates in the filter dryer during the off period. In a more controlled manufacturing process the dryer could possibly be removed. However a filter is still needed.
- Liquid refrigerant exits evaporator at startup. However, it was calculated that a substantial part of this liquid is regenerated in the SLHX.

8. Acknowledgement

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9. Animations

The IR stills can also be viewed as animations at the following web-addresses.

First 12 minutes following a startup:

[http://www.energy.kth.se/proj/Thermo/WWW/bjork/Start-up first 12](http://www.energy.kth.se/proj/Thermo/WWW/bjork/Start-up%20first%2012%20minutes.avi)

[minutes.avi](http://www.energy.kth.se/proj/Thermo/WWW/bjork/Start-up%20first%2012%20minutes.avi)

First 4 minutes following a shut-down.

[http://www.energy.kth.se/proj/Thermo/WWW/bjork/Stop first 4 minutes.avi](http://www.energy.kth.se/proj/Thermo/WWW/bjork/Stop%20first%204%20minutes.avi)

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Appendix A

0 s

10s

20 s

30 s

60 s

120 s

180 s

240 s

300 s

360 s

420 s

720 s

994 s

1054 s

1114 s

1134 s

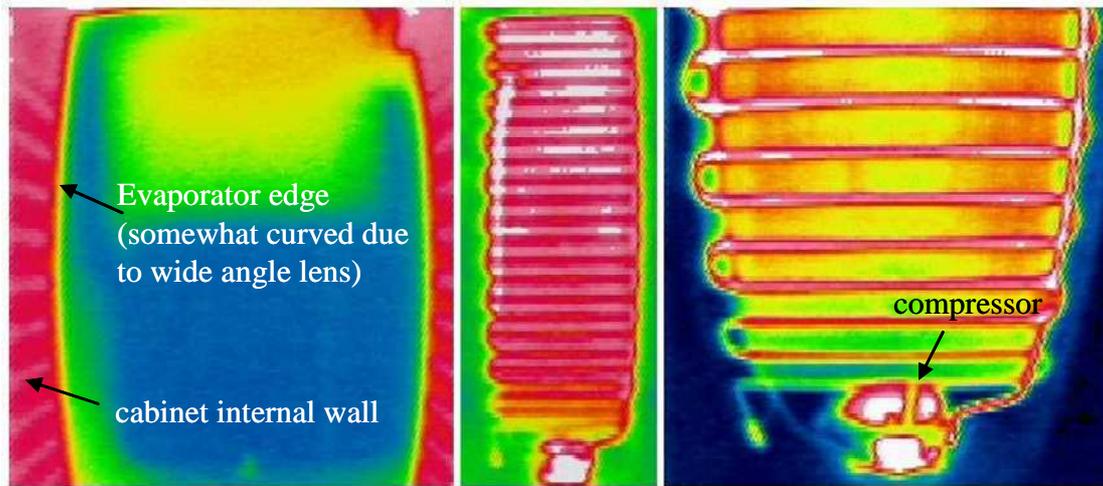


Figure 1 Images from the infrared films five minutes after a compressor start. From the left: evaporator, the entire condenser and a close-up of the lower half of the condenser. The compressor is seen as a large warm object at the bottom of the condenser.

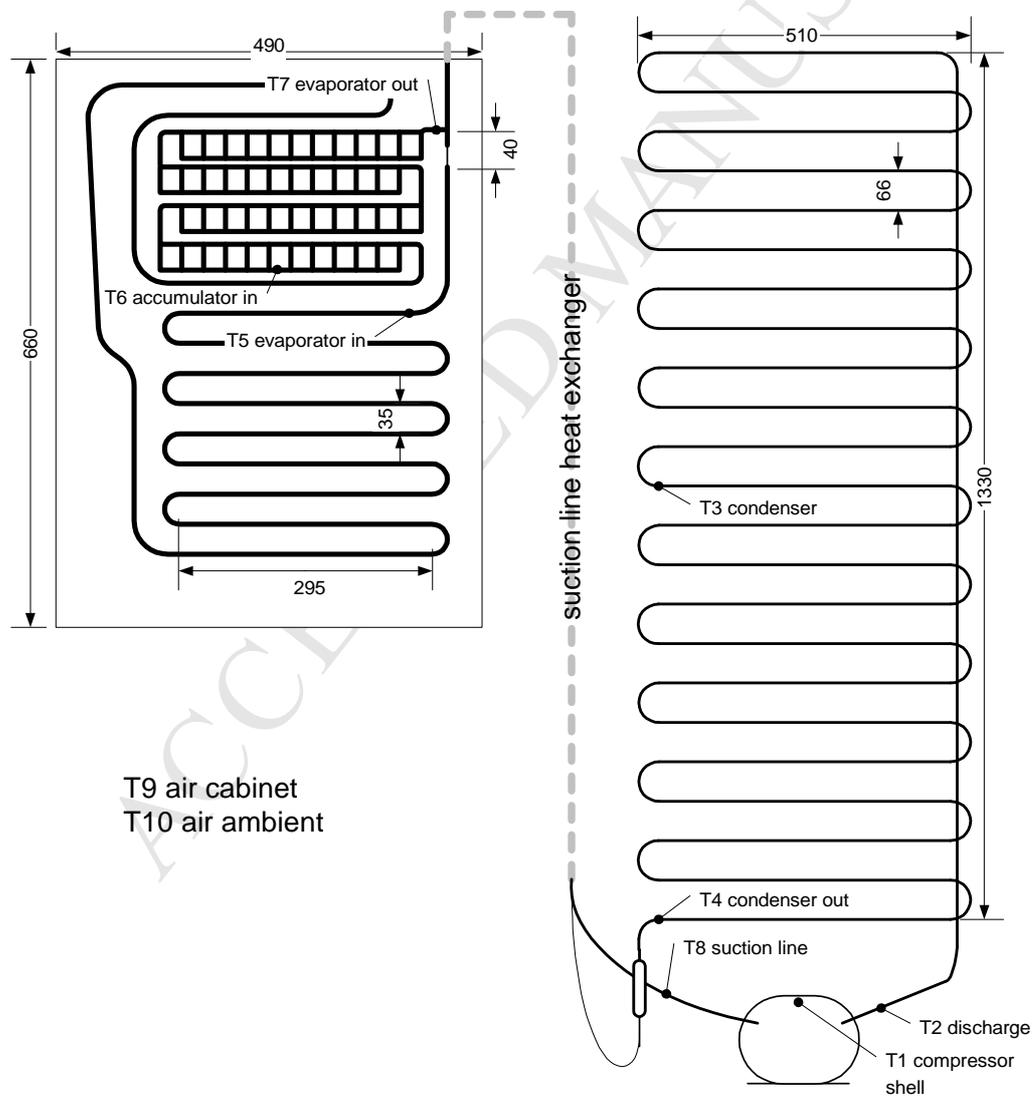


Figure 2 Cooling system and thermocouple locations



Figure 3 IR-camera in position for evaporator filming



Figure 4 IR-camera in position for condenser filming (close-up of lower part)

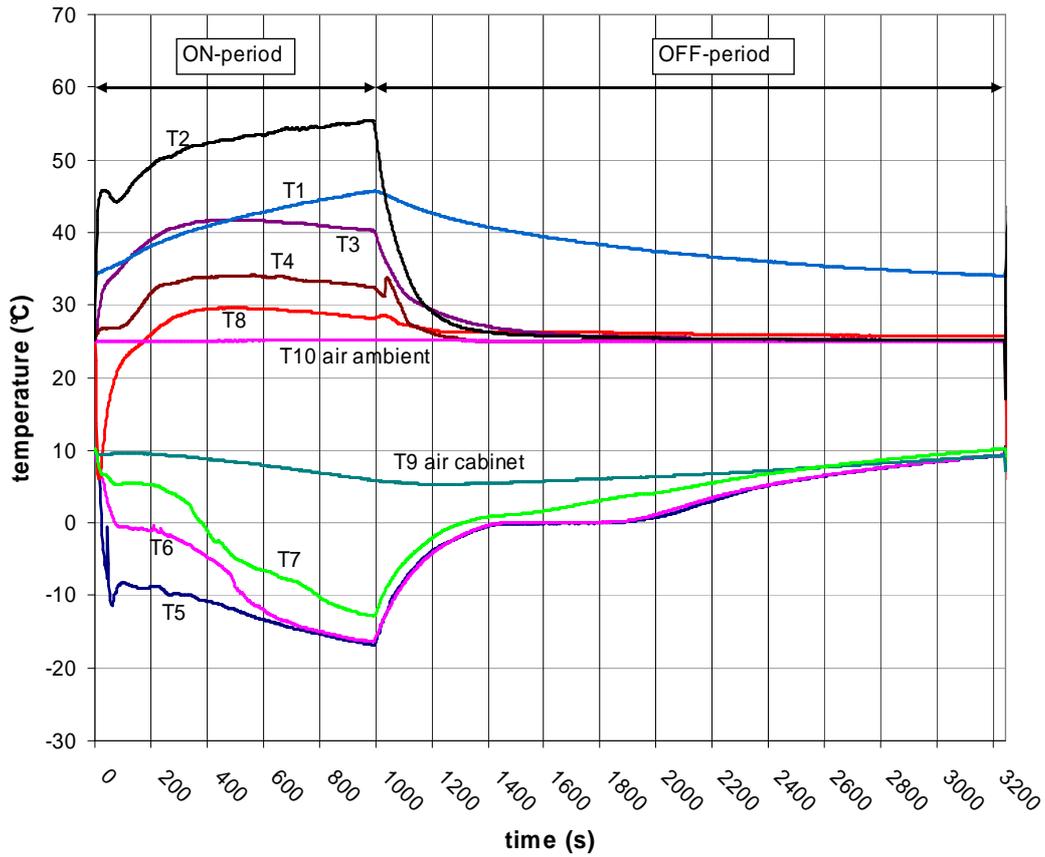


Figure 2 Temperatures during one on-off cycle. For thermocouple locations see Figure 2.

Appendix A

