Aspects of Household Cooling Technology

Matthias Mrzyglod
BSH Bosch und Siemens Hausgeräte GmbH, Germany, matthias.mrzyglod@bshg.com

Stefan Holzer
BSH Bosch und Siemens Hausgeräte GmbH, Germany, stefan.holzer@bshg.com

Aspects of Household Cooling Technology

Matthias Mrzyglod[1]*, Stefan Holzer[2]

[1]BSH Bosch und Siemens Hausgeräte GmbH, PRF/DE-GICS2,
Giengen, Germany
+49 7322 92 3398 matthias.mrzyglod@bshg.com

[2]BSH Bosch und Siemens Hausgeräte GmbH, PRF/DE-GICS1,
Giengen, Germany
+49 7322 92 2711 stefan.holzer@bshg.com

* Corresponding Author

ABSTRACT

Actually available household cooling appliances of the highest efficiency class may consume less then 10W average electrical power. To achieve such low power consumption special challenges for the cooling system had to be overcome. The related cooling system design has to deal with several effects, which arise from the corresponding low cooling capacity demand, start/stop cycles and additional power consumption by control accessories.

The paper presents symptomatic aspects of cooling technology, which have not been fully considered in the past or had only a negligible impact on electrical energy consumption. Those aspects are presented as qualitative descriptions of the phenomena, not as definite data or solutions.

Some items can be discussed as evolution of key performance data during the last years, like efficiency of compressors at specific operation points or power consumption of electrical components.

1. INTRODUCTION

Reduction of electrical energy consumption for household refrigerators and freezers has been performed very successfully in the recent years. E.g. the consumption of a typical household cooler application dropped within 15 years by approximate 65%. In order to keep a storage volume of 100 l at a temperature of 5 °C 460 Wh/d were required in 1997 whereas in 2012 only 160 Wh/d of electrical energy had to be used.

One major effect was the improvement of the insulation, but also the enhancement of the cooling system contributed significantly to this energy saving.

In 2013 appliances with energy class A+++ (EU-label system) became available, which involved further steps in energy savings. The following discussion in this paper on challenges and effects is done on a well insulated cooler application with 350 l inner volume.

According to the calculation scheme of EU-directive 1060/2010 the refrigerator must not exceed 207 Wh/d which can be converted in an average electrical power consumption of 8.6 W. Other electrical consumers have to be considered before the remaining electrical power may be used to drive the vapor compression cycle to compensate the heat flux through the cabinet insulation.

- Fans in order to increase the heat transfer coefficient on the evaporator /condenser surface
- Electronics, e.g. for appropriate control of the cooling system, display information for the user, drive the compressor motor (e.g. Inverter for BLDC motor, PTC starter), heaters, magnetic valves…

In this example the approximate remaining electrical average power P_{el} to operate the cooling cycle is estimated as $P_{el} = 7.5W$.

The heat flux through the cabinet depends mainly on the thermal conductivity of the insulation, but also other influences have to be considered:

- Heat load created by the electrical components.
- Heat bridges of gaskets, tubes, wire harness, …
- Heat flux due to increased temperature differences at specific areas (e.g. compressor compartment, evaporators, condensers).
- Heat flux due to radiation of surfaces.
- Heat flux due to specific heat transfer coefficients of internal / external air flows.
- Air movements ("breathing") of the cabinet due to temperature variations through water drain and door gasket.

In this example the total heat load for the cabinet $\dot{Q}$ is approximate $\dot{Q} = 24W$ at the standard ambient temperature of 25°C. This leads to a demand of a minimum Coefficient of Performance $COP = \frac{\dot{Q}}{P_{el}} = \frac{24W}{7.5W} = 3.2$ of the cooling system.

2. COOLING SYSTEM DESIGN

2.1 Compressor

The selection of the compressor has to consider the varying demand which results from different ambient temperatures of the appliance (e.g. climate class SN-T). Therefore the related heat flux through the cabinet (UA= ~1.2W/K) is within the range UA*(10°C-5°C) = 6W up to UA*(43°C-5°C) = 46W for ambient temperatures of 10°C and 43°C, respectively.

Compressors with such small and flexible cooling capacities are not available. Standard compressors with asynchronous motor (2950rpm at 50Hz / 230V Input, R600a) with 4cm³ displacement provide already more than 120W of cooling capacity (-10°C evaporating, 32°C condensing temperature).

A solution is the application of a variable speed compressor with BLDC motor design, which decreases the cooling capacity below 50W (1000rpm at -10°C, 32°C). Smaller displacements or speeds show a significant decrease in efficiency η_C. η_C is defined as the share of the isentropic power to the real required input power of the compressor system. It can be interpreted also as share of the realized Coefficient of Performance (COP) to the maximum $COP_{\max}$ of a reference cooling cycle at operation condition.

$$\frac{w_{is}}{P_{el}} = \frac{COP_{real}}{COP_{\max}} = \eta_C$$

A benefit to use efficiency η_C instead of COP at specific rating conditions is a simplified simulation of compressor behavior also at transient conditions as shown by [Negrao et al.]. To achieve the target energy consumption of the appliance, the COP of the compressor has to match the required COP of the system of 3.2 in this example.

Figure 1 shows exemplary compressor efficiencies for displacements of 3 up to 7cm³ (at 50Hz input) for actual compressors with asynchronous and BLDC motors at variable speeds.

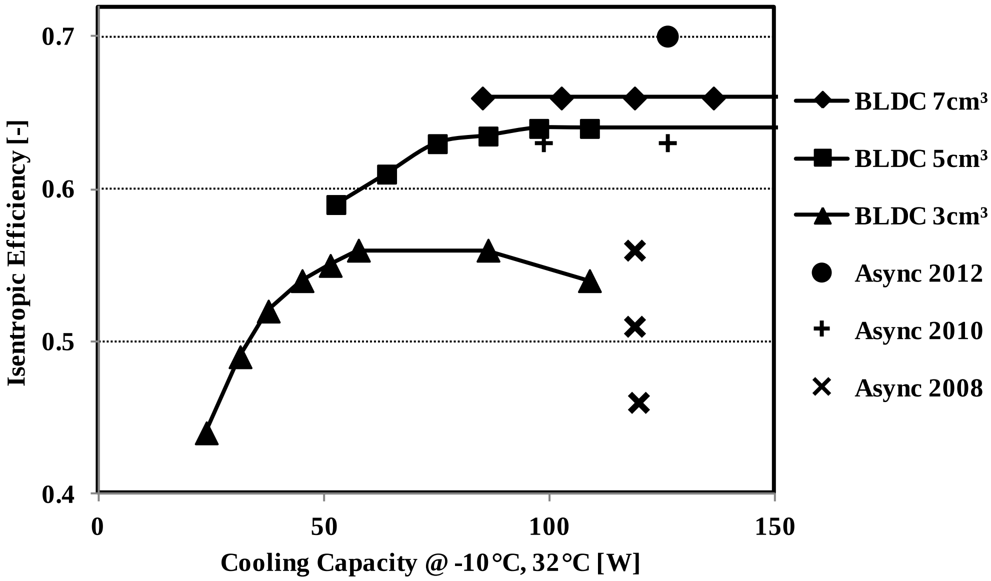

Figure 1: Examples of compressor efficiencies at low cooling capacities

The evolution of the isentropic efficiency η_C in a longer period shows figure 2. Compressors with small displacements showed better improvements than those with big displacements. Especially the focus in development for low cooling capacity demand decreased condensing and increased evaporating temperatures (often discussed as tailor made compressors) made those improvements of modern compressor generations possible.

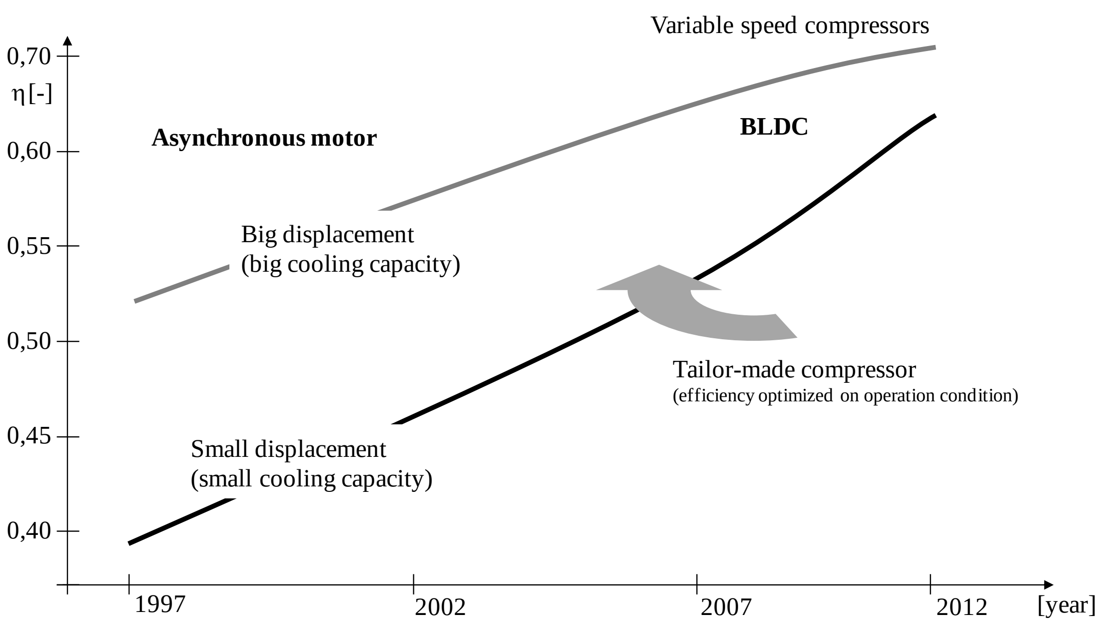

Figure 2: Evolution of household compressor efficiency

Even with an appropriate selection of compressors with high efficiencies, variable (low) speed and small cooling capacity it is obvious that the temperature controller of the appliance has to switch off the compressor, when the temperature limit in the cabinet is achieved. (Cooling capacity exceeds demand) Due to the hysteresis of the temperature controller (~1-2K), the switching occurs quite often. In this case specific care has to be taken, that the start of the cooling system does not require a significant amount of energy. Asynchronous motors used specific resistors (PTC-starter device) to start the motor by an additional auxiliary winding. This concept has the disadvantage to use high currents for a short time (~0.5s-1s).

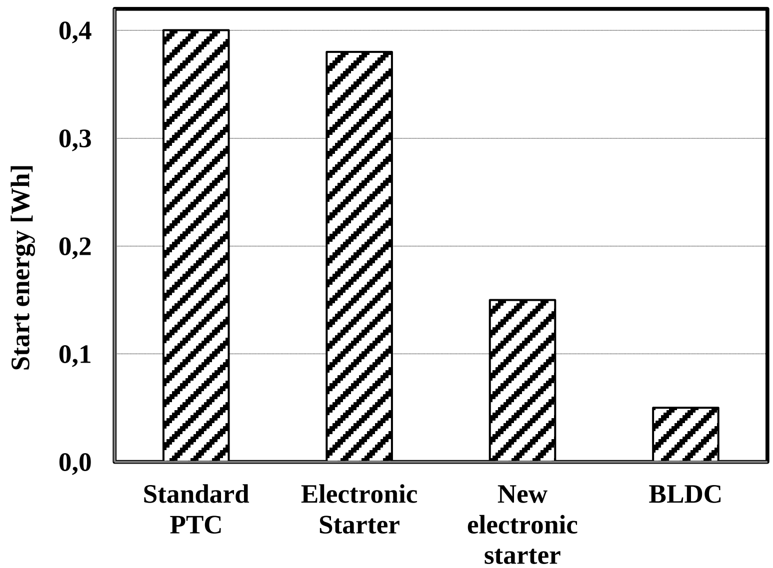

Figure 3: Evolution of compressor Start-up energy, from former PTC starter to recently available electronic starters

If the compressor is switched on 3 times per hour, the start-up energy (e.g. 0,4Wh) of the compressor sums up to 1,2W additional average power consumption.

The availability of electronic instead of the PTC-starter devices reduced significantly the permanent electric power (~2W) in order to heat the PTC. Those starters had also the potential to reduce the starting loss of asynchronous motors. A significant reduction of the start-up energy for asynchronous motors has been achieved recently by new electronic starters, modified auxiliary windings and adapted start-up times. Compressors with BLDC motors in general use only a small amount of start energy, depending on the used inverter electronic.

2.2 Evaporator

Except for NoFrost applications, a common design of the evaporator for household refrigerators is the cold-wall design, which consists of a tube attached to a metal plate, hidden behind the cabinet inner liner. In principle this design has to balance air-side heat transfer, refrigerant-side heat transfer, thermal resistances, homogeneous temperature distribution on the surface and pressure drop

Whereas static (free air convection) designs were state of the art in the past, nowadays more applications with fans and air guidance are used.

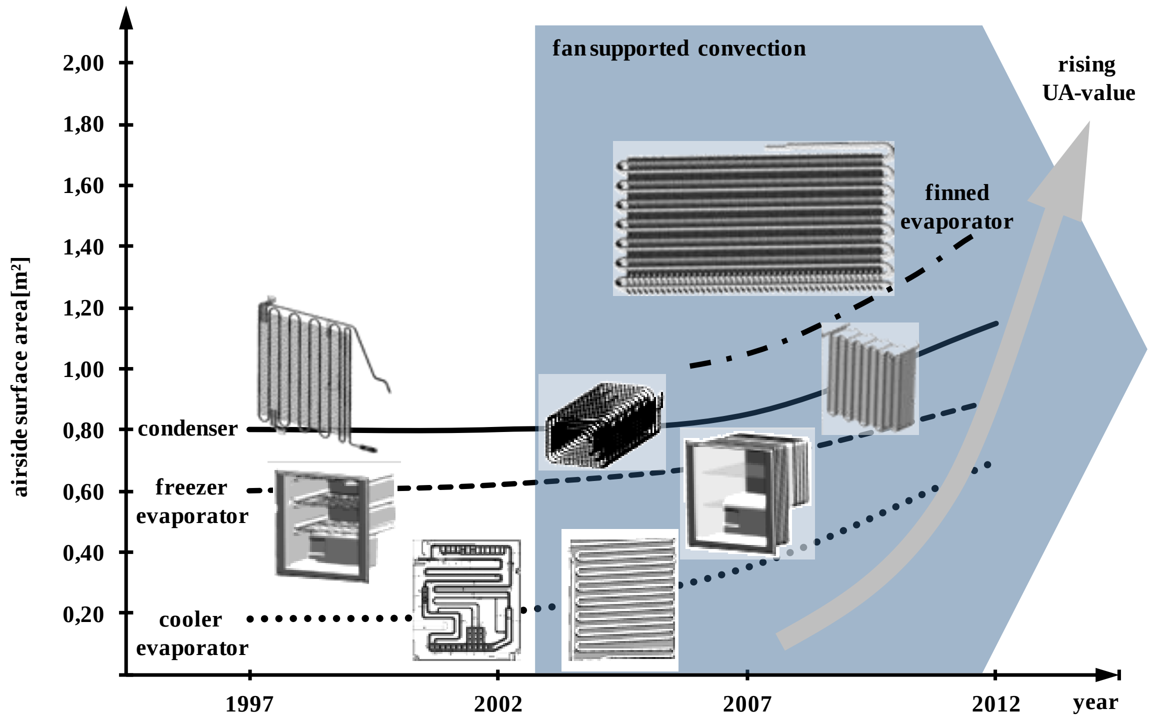

Figure 4: Trend to improve heat exchangers by increasing their area (and HTC for air flow)

One could state, that by introduction of fans the air side heat transfer coefficient (HTC) has been improved significantly. In order to gain a benefit from the improved HTC for energy saving, the electrical energy to drive the fan (and the generated heat) has to be smaller than the power saving to operate the vapor compression cycle. Therefore the efficiency for small fans has been rising continuously over the past years. Current designs show fan efficiencies up to 25% with electrical power consumptions less than 1W. The indicated fan efficiencies in figure 5 are shown as the ratio $\frac{\Delta p \cdot \dot{V}}{P_{el}} = \eta_F$ of achieved technical work of the air flow $\dot{V}$ at a pressure difference Δp to the required electrical power P_{el}

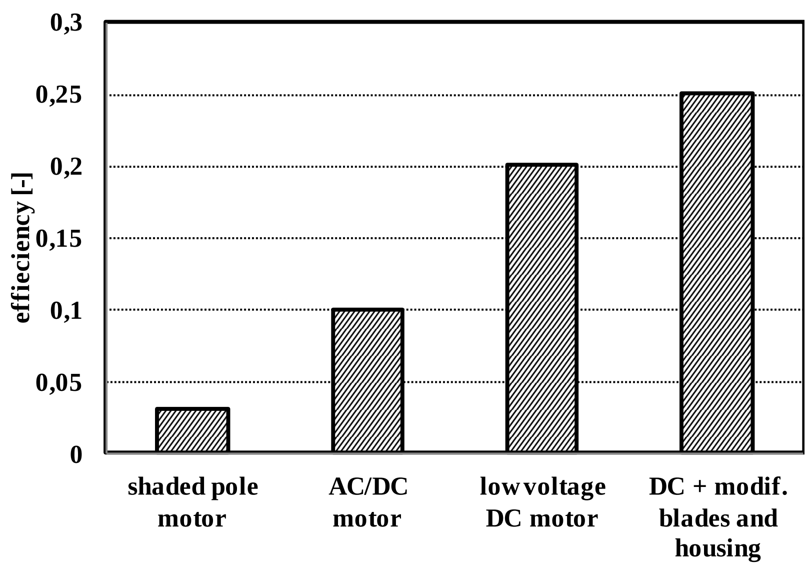

Figure 5: Trend for fan improvement

On the contrary due to the improvements in insulation and the availability of compressors with variable speed, the refrigerant side HTC has been decreasing because of small flow rates.

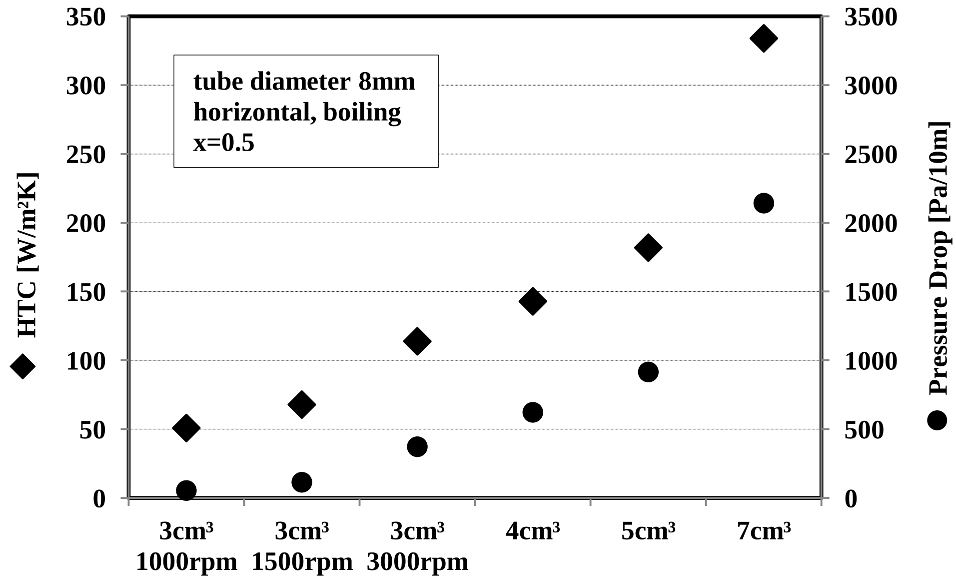

Figure 6: Calculated values of HTC and pressure drop according to [Thome] of refrigerant R600a in a tube with 8mm diameter and different compressor displacements and speed.

Figure 6 should be considered as a prognosis, how the HTC could develop with the change to small cooling capacity demand. This is mainly caused by changing of the flow regime to stratified flow of the refrigerant inside the tube. Some transparent tube sections or x-ray analysis indicate the occurrence of pure stratified flow regime, but the drop in HTC of practical tube layout configurations with turns from horizontal to vertical and vice versa may be in reality not as severe as indicated.

A positive influence on the overall efficiency is the decreasing pressure drop with small flow rates. The expected improvement in pressure drop will not compensate the loss by the decreasing HTC. Simulation and optimization of the complete layout of an evaporator, which considers all significant thermal resistances is a major challenge for small flow rates. The benefit of this optimization is an increase in the evaporation temperature, which results in a better operation condition for the compressor. A side effect is the designation of the compressor from low back pressure (LBP) to medium back pressure (MBP) application.

2.3 Condenser

Condensers in household refrigerators are available in many variants. Besides the optimization target to maximize the heat transfer capacity there is a common understanding to keep the refrigerant quantity as low as possible. The containing refrigerant could transport at every stopping period some extra thermal energy to the evaporator and hence to the cabinet. As an example, one could assume a condenser design which keeps 10gr of liquid refrigerant in the moment of compressor stop. This amount of refrigerant evaporates in the condenser, flows via the capillary and condensates in the evaporator. This yields an energy transport of 1,2Wh/stop. With the assumed switching period of 3 times/h this heat flux sums up to 3,6W, which has to be added to the previously calculated heat flux through the cabinet insulation of 24W. Current condenser designs avoid a major share of this extra heat load.

Besides the condenser itself also other parts like the connected frame heater tube and dryer may also accumulate significant amount of liquid refrigerant which contribute to this extra heat load. Appropriate countermeasures are only able to reduce the effect, but not totally avoiding it:

a) Condenser design with a flow configuration that the liquid refrigerant instantly flows to the evaporator without significant phase change.
b) Condenser with tubes and attached components in a configuration that the involved volume for liquid refrigerant is kept low.

An alternative approach is to block the refrigerant flow during off time by an electro-magnetic valve placed in series with the capillary. This solution works well, if the compressor is able to start against unbalanced pressure and the compressor restricts reliably the back-flow of refrigerant. Currently, only a few available compressors for household application meet this requirement.

An appropriate method in order to reduce the energy consumption is to lower the condensing temperature. If the appliance design allows it is a possible approach to use a bigger condenser surfaces or ventilation, like indicated in figure 4. This is well known for steady state conditions [Admiraal, et. al.]. However household appliances which are operated at normal room temperature normally show a transient behavior and the relative compressor cycle is often below 50%. With a standard control device the compressor runs typically less than 30 minutes until the compressor off condition is reached. For these appliances a condenser with substantially increased mass yields a lower condensing temperature (figure 7). The averaged condensing temperature during compressor run-time falls from approximately 41°C down to 37°C.

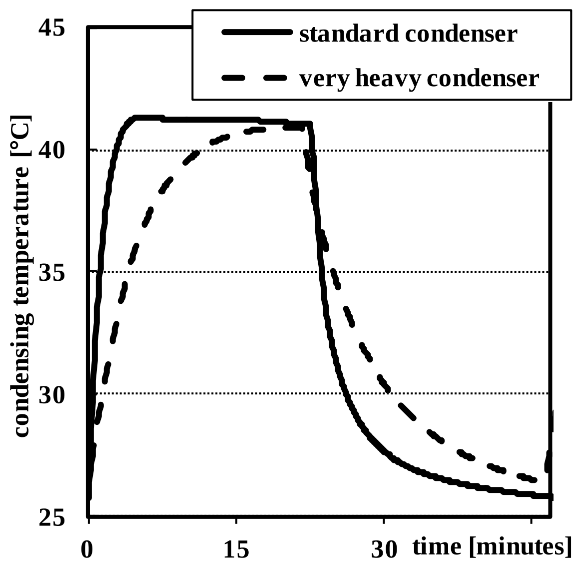

Figure 7: Comparison of a standard condenser vs. very heavy condenser

An even more effective way to achieve a lower condensing temperature is to reduce the cycle time considerably [Ilic et.al]. This became possible with modern electronic controllers. If the compressor run-time in the example above is reduced from 22 minutes to 4 minutes the average condensing temperature during compressor run-time drops down to approximately 31°C (figure 8).

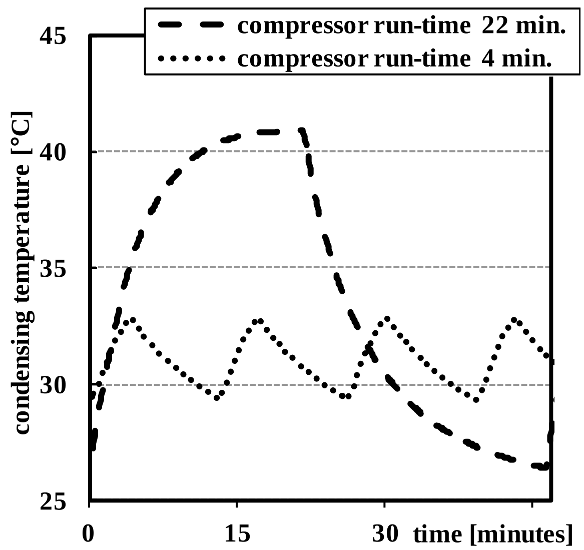

Figure 8: Comparison of a standard vs. short compressor cycle duration

The shorter cycle period requires more frequent compressor starts. It is essential for such a control scheme that the starting losses of the compressor are extremely low. Therefore a compressor with BLDC motor would be the best choice because it shows the smallest starting losses.

As discussed before, the cooling circuit itself shows some losses caused by undesired refrigerant flow from the condenser to the evaporator during compressor off time [Katipamula et.al.]. It depends on design of the cooling circuit whether an energy saving can be achieved by short compressor cycles. If the cycle period is too short, the start-stop-losses will be higher than the advantage gained by a lower average condensing temperature and the energy consumption of the appliance will rise (figure 9).

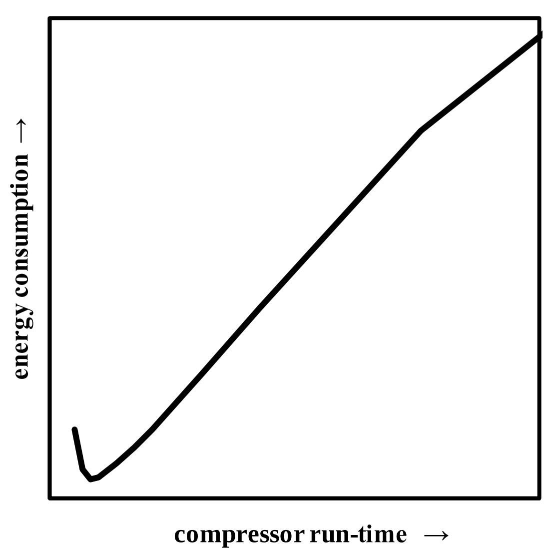

Figure 9: Energy consumption of a household appliance vs. compressor run time

As there is no general rule to select the optimal compressor run-time each appliance has to be considered separately. The refrigerator considered in figure 9 achieves a minimum of energy consumption with a compressor run-time of 4 minutes and thus more than 100 on-off cycles per day. With this new control strategy the energy consumption of this appliance drops significantly.

2.4 Refrigerant/Oil

Those applications require only a small amount of refrigerant. In the discussed refrigerator filling quantities below 50gr R600a are quite common. For operation of the system the location of the refrigerant in the system has to be understood and optimized. Many publications and specific dynamic simulation tools help to modify the cooling system for optimization related to energy efficiency and functionality.

Only one topic regarding oil and refrigerant shall be discussed here. The refrigerant mass which is used for evaporation and condensation is lower than the filled in quantity, because of the refrigerant dissolved in the compressor oil. If the oil discharge of the compressor cannot be neglected, a dynamic simulation faces the additional challenge to consider the presence of oil in other components (e.g. evaporator), too. A further difficulty is that a reliable fluid transport (oil, refrigerant) in vertical rising sections may not be ensured at small flow rates [Sethi].

Household compressors contained up to 300ml oil in the past which could hold approximate 20g R600a temporary solved in the oil. The amount of dissolved refrigerant varies strongly with suction pressure and temperature and could create severe control problems in a refrigerator design. New household compressor generations use less oil in their reservoir and have very small oil discharges, which helps to operate the appliance at changing ambient temperatures and different load conditions.

3. CONCLUSIONS

Good insulated appliances which have to handle also small cooling capacity demands have to consider several effects of small magnitude. If they are not handled properly, they could sum up to significant negative impacts on the energy efficiency of the appliance. Even with most of those effects being known and countermeasures being implemented, further activities have to follow to achieve higher-efficient cooling systems.

REFERENCES

Admiraal, D. M., Bullard, C. W., 1993, Heat Transfer in Refrigerator Condensers and Evaporators. *ACRC TR-48*, University of Illinois.

Ilic, S. M., Bullard, C. W., Hrnjak, P. S., 2001, Effect of Shorter Compressor On Off Cycle Times on A C System Performance. ***ACRC CR-43***, University of Illinois.

Katipamula, S., 1989, A study of the transient behavior during start-up of residential heat pumps. *Thesis*, Texas A&M University

Negrao, O.R. Cezar, Erthal, Raul H., Andrade, Diogo E.V., Silva, Luciana W., 2010, An Algebraic Model for Transient Simulation of Reciprocating Compressors, In: International Compressor Engineering Conference at Purdue

Mohanraj, M., Jayaraj, S., Muraleedharan, C., 2008, Comparative assessment of environment-friendly alternatives to R134a in domestic refrigerators. In: *Energy Efficiency (2008) 1*:p. 189–198

Sethi, Ankit, 2011, Oil Retention and Pressure Drop of R1234yf and R134a with POE ISO 32 in Suction Lines, Thesis, http://www.ideals.illinois.edu/bitstream/handle/2142/26242/Sethi_Ankit.pdf

Thome. R. John, 2004-2010 Engineering Data Book III, Wolverine Tube, Inc., http://www.wlv.com/products/databook/db3/DataBookIII.pdf