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Calibration Strategies And Limitations Of Cycle Simulations Representing Complex Domestic Cooling Devices

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ABSTRACT

Transient cycle simulations are commonly said to be a tool for improving and understanding the operating behavior of cooling cycles. Assuming a properly working simulation tool, simple cycle configurations which are driven by On/Off-controlled compressors can meanwhile be modelled with relatively high effort. Given the growing complexity of today's modern refrigeration appliances – in particular cooling/freezing combinations – it is getting more and more challenging to represent these in a satisfactory manner. Fans for ventilation, variable speed drives and switchable capillary tubes are in interaction and follow a sensor-controlled, prescribed logic.

The focus of this paper lies on the simulation of a particular real world appliance, which features all before-mentioned actuators. The simulation tool comprises component models beyond state of the art which have been validated both independently and in a simple cycle configuration. The attempt of modelling the cooling/freezing combination leads to new knowledge concerning the approach of calibrating the model. Measurement data (temperature, pressure, electric power, state variables) is available and used as reference. It can be shown, that it is possible to shape the model to follow the order of switching commands of the real appliance. Effects like heat conduction from walls to the sensors, delay caused by the thermal inertia of the sensors or temperature stratification pose the main obstacle on the way to obtain a comparable energy consumption.

1. INTRODUCTION

A refrigeration cycle embedded in foam and metal or plastic casing, like it is the case for a simple household cooling/freezing combination, is literally nontransparent. Especially the time dependent state of the refrigerant along its path can't be observed well without changing the system's characteristic and energy consumption significantly. From this point of view, a transient, 1d-simulation sounds interesting in order to gain insight into the cycle dynamics. Secondly, the determination of the efficiency depends on the outcome of the energy consumption test. Therefore, efforts at the cost of several days can be estimated if both preparation and measurement are counted. If a simulation tool could help getting a good prognosis of the energy consumption, parameter-variations or other analysis could be accomplished much easier. These two arguments certainly lead to constant improvement in numerical analysis of cycle simulations.

Based on reviews of Bendapudi and Braun (2002), Hermes (2008) and Ding (2007), cycle simulation tools in this area started being used up to 40 years ago, where most simplifying assumptions were made concerning heat exchangers, compressor, and expansion device. By the time more and more evolved models were coupled: 1d separated fluid representations of the heat exchanger domain, compressor models which take the oil/refrigerant interaction and the thermal inertia of the compressor into account, or detailed cabinet representations are examples of complex transient cycle simulation sub-models. As impressive the description of such models sounds, not only advantages come along with growing complexity. Longer calculation times, convergence issues, and certainly growing calibration efforts come along during model development with more degrees of freedom. The benefit of a more realistic, physically more exact model has to be weighted carefully against the mentioned drawbacks. Calculational speed and convergence are closely connected to the mathematical formulation of the models and the solving-approach. But irrespective of the chosen discretization scheme, the solver settings or the programming style, the calibration of several unknown parameters has to be carried out carefully. This procedure demands critical analysis of physical phenomena, especially in thermal analysis, where various heat transfer mechanisms are possible and interfere. To point out the importance of such consideration, a cycle simulation (Heimel *et al*., 2015) is used in order to reproduce the transient behaviour of a high-end cooling/freezing combination. The real world device features a variable speed compressor, a magnetic valve for feeding two capillary tubes, 3 independent temperature zones, a fan for forced air-convection as well as several temperature-sensors to supply the onboard logic with information.

2. COMPONENT MODELS

First of all, the component models of the cycle simulation are explained to get an idea of the interaction between these components. These sub-models are compared and validated according to real devices independently, before they are used as part of the cycle simulation.

2.1 Compressor

A hermetic reciprocating variable speed compressor is used. Despite good experience with semi-empirical correlations for fixed speed drives, a different solution is applied for the variable speed drive (VSD). Comprehensive –measurement data is necessary to calibrate semi-empirical formulations. The influence of a variable rotational frequency (40-150 Hz) would have consumed a multiple of this time. Therefore, an interpolation/extrapolation scheme is used in combination with values from the datasheet (mass flow rate, electric power consumption). The interpolation concerns evaporating pressure, condensing pressure and frequency. The outlet enthalpy is gained by applying a thermal network, which captures the impact of the main thermal inertias. Furthermore, the shell-model includes oil-refrigerant-interaction as well as the heat transfer to ambient air.

2.2 Condenser

A one-dimensional model including two phase flow, maps the functionality of the rear-mounted, wire on tube condenser. The refrigerant flow results from solving mass- and energy conservation equations, correlations for pressure drop (Quiben und Thome, 2007), heat transfer (Thome *et al*., 2003) and void fraction (El Hajal *et al*., 2003). The model is able to consider slip between gaseous and liquid phase. The steel-tube including its thermal inertia is also implemented and strongly influences the dynamic behaviour of the condensing pressure. The solution methodology is a mixture of explicit and implicit schemes which solve the pressure distribution inside each cell for given boundary conditions. For more information, the work of Berger *et al*. (2015) is recommended.

2.3 Evaporator

Very similar to the principle of the condenser, the evaporator model is set up. They differ mainly in the specific correlations for heat transfer (Wojtan *et al*., 2005a, 2005b) and void fraction (Steiner, 2006 - modified Rouhani-Axelsson-model) as well as the geometry and material properties. The pressure drop is calculated in the same way as for the condenser. Of big importance is the heat transfer mechanism at the air-side, which mainly determines the overall thermal resistance of the heat exchanger. In our example, we have two different evaporator setups combined in the investigated cooling/freezing combination: These are a tube on plate evaporator for the 5 °C temperature zone and a wrapped evaporator for the -18 °C temperature zone.

2.4 Capillary tube / suction line heat exchanger

This model results from comprehensive measured and simulated data and has been successfully used in several cycle-simulations by now (Heimel *et al*. 2014). The core is an Artificial Neural Network, which computes mass flow rate and suction line outlet enthalpy depending on 10 different parameters like geometry or thermodynamic boundary conditions. The basis of the ANN was computed by using a validated, one-dimensional model for capillary tube heat exchangers. The advantage of this method is the big range of applicable input parameters which is needed for transient processes.

2.5 Compartment

Close attention has to be paid to the compartment and the housing of the cycle-components. The device for this study consists of 3 independently controllable compartments (fresh food, chill, freezer). Top mounted is the fresh food compartment, which is cooled by fan driven air from the chill compartment beneath. The chill compartment receives a cold airstream which comes directly from the foamed tube on plate evaporator. The freezer is bottom mounted and is cooled by a wrapped evaporator. The challenge is now the modelling of the main heat transfer paths between the temperature zones as well as ambient air. Also, the thermal inertia of the walls contributes to the transient development of refrigerant temperatures and pressures and affects the overall energy consumption. Therefore, the model is divided into 3 single boxes, each of them consists of 6 walls. For these walls, the one-dimensional heat conduction equation is solved. The virtual heat exchangers are in contact with the parts of the walls, which they also contact in reality.

3. CYCLE LAYOUT

3.1 Real device

The investigated design is commercially available and features the latest state of the art. Three temperature zones are realized by two evaporators connected in series and a fan. The injection points of the two capillary tubes are located at the beginning of each of the evaporators. First comes the tube on plate evaporator for the 5 °C and 0 °C compartment, second the evaporator for the -18 °C compartment. An electric valve can select a capillary tube for current operation. A single capillary tube heat exchanger combines suction line and both capillaries at once. The variable speed compressor operates between 1300 and 4500 rpm depending on the previous values of on- and off-times.

Table 1: Device specifications

Energy efficiency class	Net volume	independent temperature zones	Freezing capacity in 24h	compressor type
A+++	359 L	3	14 kg	VSD

3.2 Simulation

Before-mentioned models are assembled according to the schematics of the existing device (Figure 1). The logic as well as the geometry of tubing and the cabinets could be obtained by the manufacturer. Detailed information about the heat exchanger, datasheets of the compressor and the capillary tubes and the sensors/actuators are available and incorporated in the model. Despite very accurate geometry and material property data, the system is not determined perfectly, since precise knowledge of local heat transfer coefficients at various positions is missing. Also, the reduction from real, 3d-geometry to 1d or 0d models, comes along with difficulties and requires simplification and generalization.

The solution methodology of the cycle simulation can be described as module-wise transient calculation, where the refrigeration cycle is evaluated first, afterwards the compartment and the air cycle at every time step. The initial conditions are ambient temperature and equalization pressure. The refrigerant is accumulated in the oil of the compressor model, after a pulldown, the system reaches after several hours its typical cyclic behaviour. The thermodynamic boundary of the system is the ambient temperature. In combination with refrigerant charge and thermostat settings, the system is fully determined.

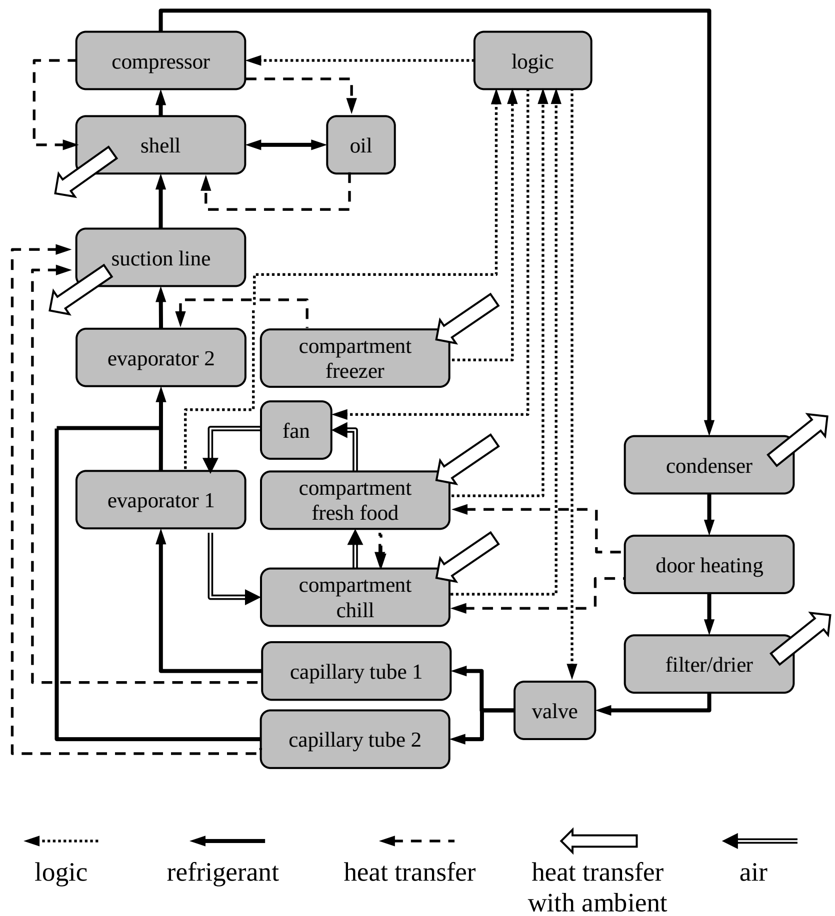

Figure 1: Scheme of the cycle simulation setup

3.3 Consideration of non-standard components

Several components in this cycle deserve special attention with respect to modelling and implementation. First of all, the cooling mechanism of the fresh food- and the chill compartment is realized via air. A fan blows air which comes from the fresh food compartment along an evaporator, mounted on its backside. The cold air is forced to enter the chill compartment, absorbs heat and enters the fresh food-compartment again. This means for the model, a separate hydraulic network has to be set up. The evaporator is only in contact with the airstream and the backside of the cabinet wall. A sensor, attached to the evaporator, measures its temperature. This information is used in the device controller. Therefore, it is important to have this temperature also in the simulation available. The capillary tubes are chosen according to the position of the magnetic valve, either capillary 1, which leads to evaporator 1, or capillary 2, which injects the refrigerant between evaporator 1 and 2. Hard switching elements are a challenge in transient simulation processes, therefore ramps, which continually increase or decrease mass flow rate, are applied. The variable speed compressor is modelled, using the datasheet, where mass flow rate and electric power are given in dependence of frequency, suction- and discharge-pressure. Correlations for outlet temperature are not given and therefore have to be sought.

4. CALIBRATION-CHALLENGES

4.1 Thermodynamic properties of the cabinet

The quality of the cabinet's insulation determines the cooling capacity of the system. This simple relationship points out the importance of realistic modelling of this component. Unfortunately, three things prove this task to be a major challenge. Firstly, the insulation features gaps, cable-feedthroughs or door-sealings. Given a 1d thermal model of each wall, these disturbances cannot be resolved in detail. A holistic method can be used, if these effects should be considered – this would be, heating the compartment with constant heat load until the temperatures reach steady state – now, the temperature difference as well as the heat load is known and the model, which describes the heat transmission, can be calibrated in terms of adaption of the heat conduction coefficient. Secondly, the thermal mass of the device ($V \cdot \rho \cdot c_p$) is probably not known exactly, though it influences the transient temperature during cyclic operation of the device. A good guess for the thermal mass can be obtained by a rapid change in heating load from the previous heat conduction test. The result, a step-response in terms of temperature can be used, to fit the thermal mass of the cabinet walls. The third big issue is related to natural convection or stratification of air. To get an estimate of the air movement, which influences the heat transfer coefficient, rather complex 3d-CFD calculations, or flow visualizations are necessary. For a lumped model of air and the surrounding walls, no such things can be considered. A solution would be the implementation of findings from more detailed simulation in the form of parametric models.

4.2 Influence of load

Load inside a cooling cabinet influences the system in two ways. Firstly, the thermal inertia changes. A fully loaded cabinet equalizes changes in air temperature more rapidly. After a door opening, for instance, the air quickly reaches the temperature of the load again. Secondly, the velocity of air, may it result from natural convection or a fan, depends on the cross sectional areas of the tunnels between the obstacles (load) and the resulting pressure losses. As a result of different air velocity, also the heat transfer coefficients change and have impact on the overall energy consumption. This thesis can be confirmed by two test measurements, where only the storage plan of the test packages is modified. At defined thermostat settings (-21 °C / 4 °C) and ambient temperature as well as humidity, the difference in energy consumption per day is as high as 44 Wh/day (~10 % of daily energy consumption), with a difference in relative duty cycle ratio of 18.6 percentage points. With alternative thermostat settings (-18 °C / 5 °C) the same test gives 38 Wh/day and 4.5 percentage points difference.

4.3 Thermal inertia of sensors

Sensors, which measure the temperature of the air inside the cabinet, are covered by protective layers and mounted at one of the inner walls. The sensors themselves have a certain weight and material properties. In combination with the local heat transfer coefficient, the transient temperature which is measured by the device, is determined and represents not necessarily the real air temperature. The bigger the term $\alpha/(V \cdot \rho \cdot c_p)$ is, the higher is the delay between the measured signal and the air temperature. In Figure 2, the dotted line represents the air temperature, the other lines are simulated temperatures for different sensor configurations. Sensor 1 (dash-dot) lags 11 min behind the air temperature, showing only 35 % of the total amplitude of the temperature signal. The sensor simulates a copper cylinder of 20 mm in diameter where the temperature is measured in the middle.

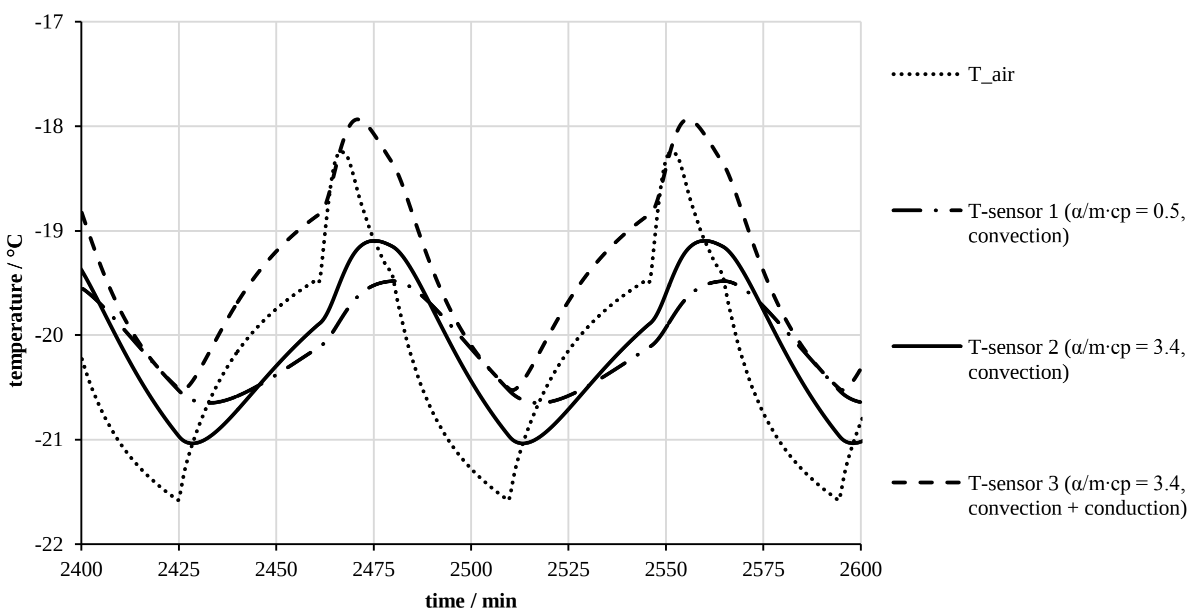

Figure 2: Influence of sensor inertia on temperature reading (simulation)

Sensor 2 (solid) corresponds to a commonly used, plastic coated sensor – the lag is around 3 min compared to the air temperature, measuring 78 % of the amplitude. If additionally heat transfer via the sensor-mount occurs, the signal can be shifted and deformed, like it is the case with sensor 3 (dash). Here, transient heat conduction from one of the naturally warmer walls is considered. This sensor shows the most realistic signal and is used for the cycle simulation setup. The issue now is, that the system's logic doesn't care on the average air temperature, but on the measured signal from various temperature sensors. Based on these information, the compressor, the magnetic valve or the fan are controlled. If the requirements to the cycle simulation are to capture these switching actions, also the sensors have to be modelled as lumped masses. The heat transfer mechanism is obviously not only convection but also conduction plays an important role. This thesis is discussed in the following chapter.

4.4 Sensor –related heat transfer

To investigate the heat transfer mechanism related to the sensor position, mounted directly at the ventilated evaporator, several measurements have been collected and compared. Air is taken from the fresh food compartment (light grey, solid), ventilated along a plate-type evaporator and discharged into the chill compartment. In Figure 3, the sensor (black, solid) mounted on the evaporator, measures the temperature of the plate and sends this information to the system's logic. This sensor is crucial for the cycle simulation, since the compressor and the magnetic valve depend on its reading. So, the aim is now to set up a thermal network to reproduce the measured temperature. Another sensor (dark grey, solid) with very low thermal mass is attached directly at the tube on plate evaporator at a position, where no influence from moving air can distort the reading. This sensor is not used by the logic of the device and represents the average temperature of the tube on plate evaporator. The two unknown heat flows (conduction via sensor mount, convection of air from fresh food compartment) can be determined. Three possible cases of mismatch are shown – convection is underestimated (thin, dash), conduction is underestimated (thin dash-dot) or the sum of both is simply too small (thin, dot). Finally, an optimum can be found, where the both the total amount and also the relative share of heat transfer by convection and conduction represents realistic conditions (thick, dot).

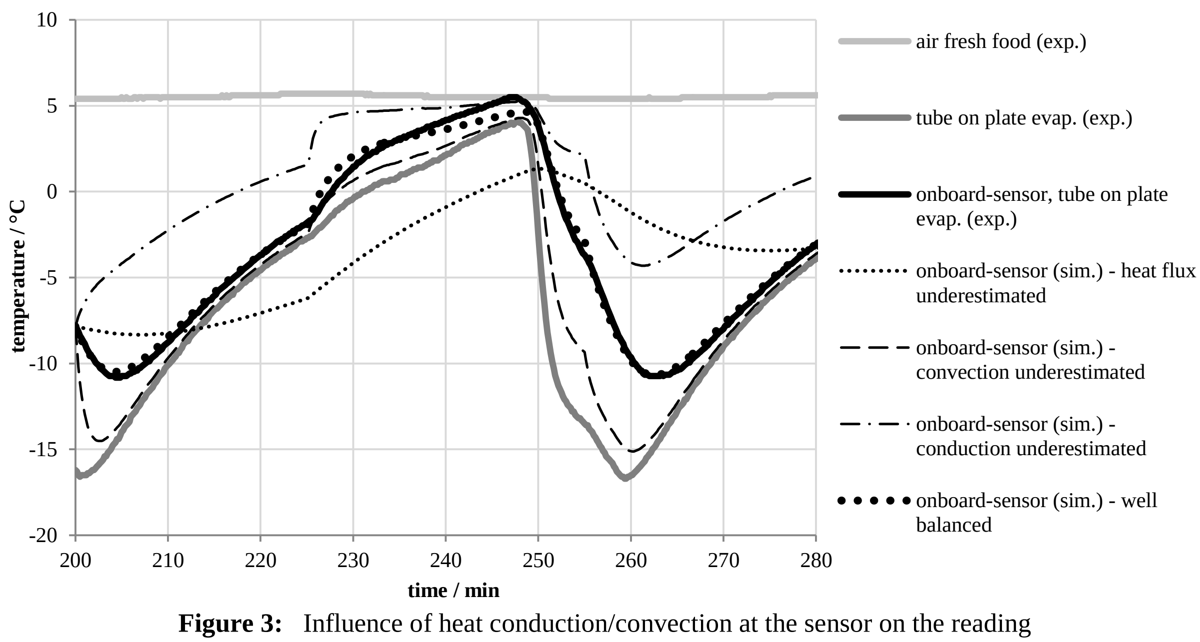

Figure 3: Influence of heat conduction/convection at the sensor on the reading

4.5 Remaining uncertainties

Heat transfer at the condenser has to be considered carefully due to the combination of complex air-side geometry and natural convection of air. The heat transfer has big impact on the condensing pressure, as well as the heat transfer on the evaporator-side influences the evaporating pressure. This knowledge can be used to correct assumptions concerning heat transfer at these components. The door heating is another source of uncertainty, since the heat transfer mechanism is a mixture of convection and conduction. In combination with poorly defined contact surfaces, an estimation of the overall heat loss along this component can lead to different degree of subcooling at the capillary inlet and furthermore influence the whole system. Also the heat flows concerning the compressor and its surroundings have to be treated. The shell-side heat transfer coefficient as well as unknown internal heat flows, contribute to the overall uncertainty of the system.

5. RESULTS

Considering before-mentioned issues, the existing cycle simulation is fed with adjusted sensor and compartment models as well as other similarly estimated heat transfers (door heating, condenser, evaporator, …). A simulation of the energy consumption at 25 °C ambient temperature, -18 °C freezer and 5 °C fresh food compartment temperature is carried out. Therefore, the simulation is stopped, when cycle averaged values of temperatures, mass flow rates and compressor power show no significant trend. Results can be seen in Figure 4, where temperatures measured by onboard sensors from all three cabinets are compared to its simulated values. Beneath, the logical states of compressor, magnetic valve and fan are shown. Besides longer absolute on/off times in simulation (sim: 49/36 min, exp.: 39/24 min) the order of switching commands proves to be the same. This is only possible due to the calibrated sensor models. A slight change in one of the heat transfers would lead to different switching commands, hence energy efficiency. The simulated energy consumption reveals 444 Wh/day at 58.0 % duty cycle ratio, the measured reference case shows 453 Wh/day at 62.4 % duty cycle ratio. At the beginning of each on-cycle, capillary 1, which expands into the fresh-food compartment, is in use. At the same time, the fan in the fresh food and chill compartment is on, to cool the air in primarily in the fresh food compartment (black, solid). The fan is shut off, when this temperature reaches a lower limit. Several minutes after this event, the magnetic valve reacts as a consequence of temperature readings from several sensors. Only the freezer is now in operation, the saturation temperature (black, dash) reaches its lowest point. Meanwhile, due to natural convection, the evaporator for the fresh food and chill compartment gets warmer (black, dot) because of heat input from air of before-mentioned compartments (black solid, black dash-dot). Next, after several minutes, the compressor is stopped, since all temperature-criteria are met and no further cooling is needed. Obviously, the simulation of such a device, based on complex connections of sensors and actors, demands precise knowledge of the logic as well as of the sensors. A difference between simulated and experimental sensor output of 1 °C may result in a switching delay of several minutes. An influence of similar magnitude on the switching times has the mass flow rate, hence cooling capacity, and the heat input into the compartments. Due to the fact, that the uncertainties for models like compressor, capillary or heat exchanger is in the order of several percent for most accurate ones and beyond 10 % for more common models, the chance of a mismatch between observed and simulated behaviour is not inexplicable. This is also probably the explanation of the difference in absolute switching times mentioned before. An independent measurement at another temperature controller setting is carried out and compared with the simulation.

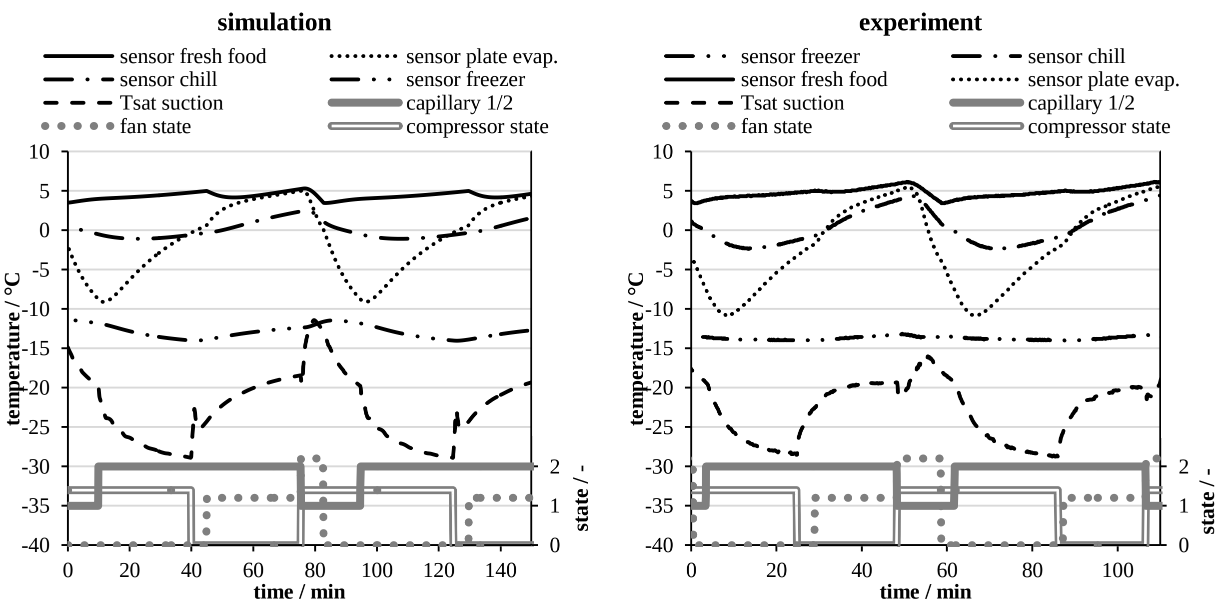

Figure 4: Comparison of simulation and experiment at 25 °C ambient temperature

The simulated result shows 594 Wh/day at 47.7 % duty cycle ratio, the measured reference case shows 569 Wh/day at 65.5 % duty cycle ratio. Although, the relative duty cycle ratio shows a significant discrepancy, the electric energy consumption is within 5 %. In the simulation, the device switches due to longer absolute on-times earlier to a higher frequency, hence the discrepancy in run time ratio.

To demonstrate the sensitivity of the system behaviour with respect to external influences, a variation over the ambient temperature is carried out. Figure 5 shows the state of the magnetic valve, the fan and the compressor. "1" stands for the lowest rotational speed of the compressor, succeeding numbers denote higher rotational speeds.

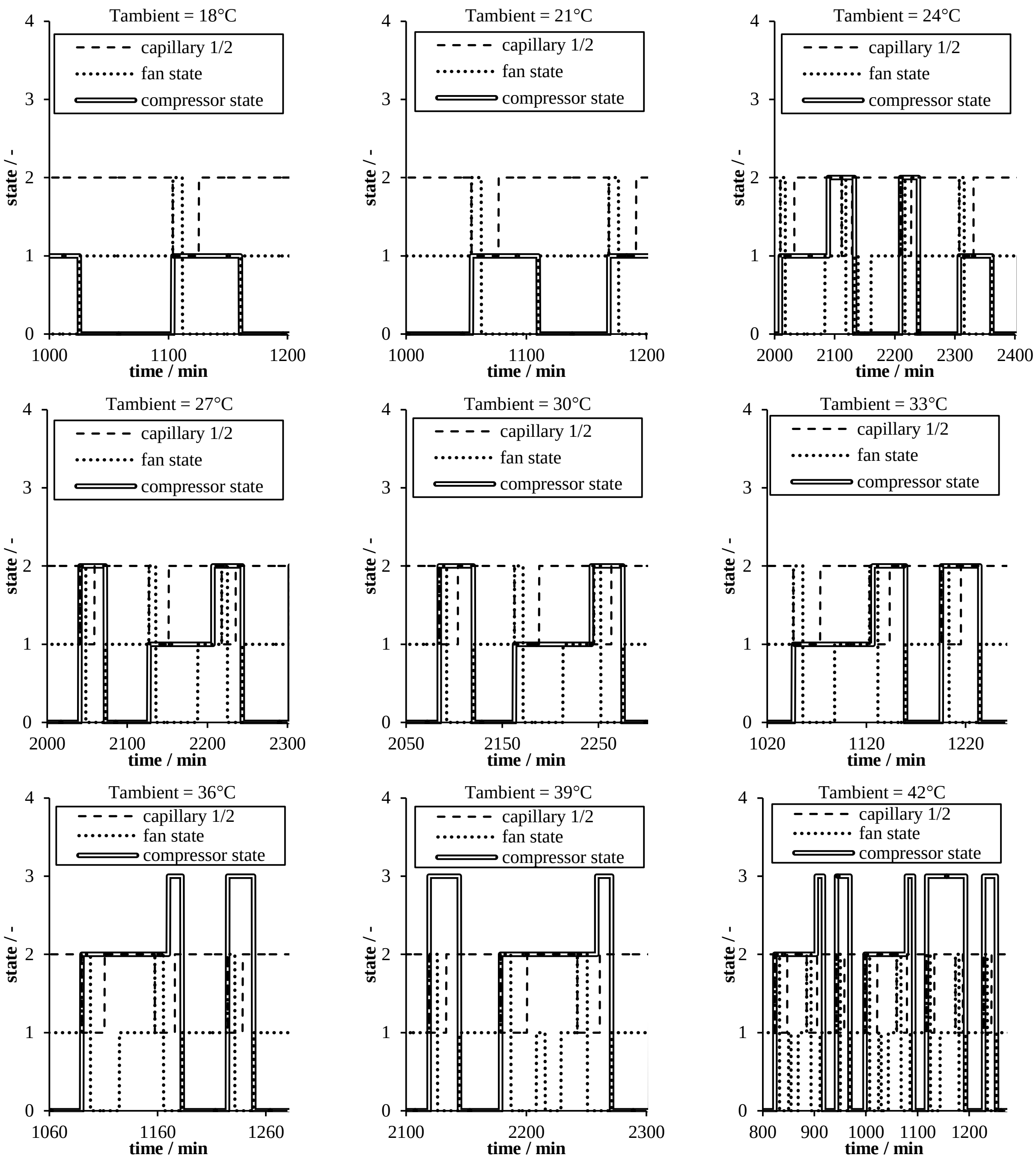

Figure 5: Simulated cyclic behaviour at ambient temperatures between 18 and 42 °C

"capillary 1" injects into evaporator 1 and then evaporator 2, "capillary 2" injects directly into evaporator 2, which is located in the freezer. The fan runs at two different speeds, 0 means, like for the compressor, "off". The temperature settings are not the same as used for validation in Figure 4 but are maintained constant throughout this variation. It can be seen, that with growing heat input into the compartments, the compressor runs for longer periods until it uses higher rotational frequencies. The fan runs intermittently and stops in general shortly after the compressor has started. The magnetic valve is at position "1" and switches to "2" during compressor runtime. Starting from 24 °C ambient temperature, the magnetic valve starts to switch back to "1" during the compressor runtime. The compressor itself rises its rotational speed after a certain period of runtime and lowers it, if the duration of a cycle is below a certain time-limit. Hence, at low ambient temperatures, a typical, periodically recurring cycle consists of a single compressor operation. For higher temperatures, up to five on/off operations can be observed during a periodic cycle. Highly probable is, that these more complex patterns also occur in reality. Since their strong dependence on heat flows inside and into the compartments, it won't be possible to measure congruent behaviour since the impact of 3-dimensional effects, which are not covered by the 1d model, may increase with increasing temperature gradient.

6. CONCLUSIONS

6.1 Core findings

Summing up, a complex cooling/freezing combination is modelled by means of a 1d cycle simulation. The sensitivity of the real system poses a limit on the simulation, since the compressor, magnetic valve and fan are controlled by temperature measurements.

- These measured temperatures rely on sensors, which have themselves influence on the measured signal. The settings of the logic are chosen in such a way, that a possible systematic measurement error, offset or delay due to thermal inertia are compensated for. This poses an immense obstacle to simulation, where correct switching behaviour is valued. In such a case, all physical sensors have to be modelled and calibrated in simulation to existing measurements.

- The cooling load is determined by the properties of the compartment. Thermal bridges, feedthroughs or other distortions of the insulation shall be accounted for by separate experiments. This works well by artificially heating the compartment and measuring heating power and resulting temperatures.

- Influences like 3-dimensional fluid motion and transient heat transfer coefficients cannot be properly integrated in a cycle simulation. Based on experiments, this assumption can be justified. In order to calculate in real time with a standard computer, 1-dimensional simulation of multiphase flow already needs the main amount of the allowed calculation time.

- Remaining uncertainties, especially heat transfer at certain positions (e.g. condenser, door heating, compressor, …) can be only partly eliminated by single experiments and critical analysis of the real geometry.

6.2 Recommendations

The purpose of a cycle simulation defines the detailedness of the simulation model. If the transient power consumption at certain ambient temperatures is the aim of the investigation and the logic of the device sufficiently complex, detailed examination of all sensor positions and heat transfer paths are still no guarantee for success. Due to natural uncertainties in all of the sub-models the switching pattern, hence the duty cycle ratio and moreover the electric energy consumption are influenced. For optimum calibration of the model, single experiments of subsystems, like heat resistance of cabinets, responsiveness of sensors or heat transfer coefficients should be carried out. Finally, an independent experiment has to be used in order to validate the functionality of the simulation model.

NOMENCLATURE

c_p	specific heat	(J/kg K)
m	mass	(kg)
T	temperature	(°C)
V	volume	(m^3)

Subscripts

sat	saturation

Greek symbols

α	heat transfer coefficient	(W/m^2 K)
ρ	density	(kg/m^3)

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