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ABSTRACT

The development process of energy-efficient household refrigerators and freezers is characterized by time-consuming and expensive experiments. In order to make this development process more efficient, dynamic (transient) models of the individual components of the entire refrigeration cycle were developed at Graz University of Technology. To ensure operability beyond the research project and to enable industrial application, these models have been implemented into a commercial process modelling software. The advantage of using such a simulation tool is the simplicity of conducting parameter studies as well as performing design modification to the device.

This work includes modelling approaches of each refrigeration cycle component and the freezer model. Since the model parameter values are unknown and in order to calibrate the freezer model with measurement data, a semi-automatic method for model calibration was developed. Due to interactions and nonlinear dependencies between the model parameters, a manual parameter optimization would be very time consuming and inefficient. The semi-automatic calibration method is capable to handle a large parameter set in a fast and user-friendly way. Furthermore, results of the validated freezer model are presented, followed by an application demonstration of the simulation model.

1 INTRODUCTION

In order to counteract the trend of increasing global energy consumption, energy consumption labelling have been introduced in many areas. These easy-to-compare labels and the increased environmental awareness of consumers lead to major technical challenges for manufacturers of household refrigerators and freezers. Due to steady increasing unit numbers the share of the total energy consumption is substantial. Studies of IEA (2009) showed that in the European Union, about 15 % of the electrical energy consumption of households is used for chilling and freezing food. The average energy consumption of a household refrigerator is given as 1 kWh / day according to Hermes and Melo (2008). According to BDEW (2014) this results in an annual energy consumption of about 22 TWh for Germany (as of 2011). In the development of modern cooling units, more and more computer-aided simulation tools are used in addition to experimental approaches. The aim is to reduce the need for costly and time consuming experimental measurements, thus accelerating the development process. Therefore, within the research project ECO-COOL, a simulation program was developed to calculate the dynamic operation of household refrigerators. For this purpose, transient models for all relevant components of the refrigeration cycle were developed and implemented in the scripting language Visual Basic for Applications (VBA) by Heimel et al. (2015)

and Heimel et al. (2016). The individual components and models of this simulation program are linked together at code level. The equations of the models and the solution method are interlaced. As a result, modifications to the refrigeration circuit are associated with considerable effort and hardly possible for an inexperienced user. In order to ensure the usability of the developed models for unexperienced users, the models were transferred to the commercial process modelling software IPSEpro, see Rohrhofer et al. (2016). A semi-automatic method has been developed and implemented in order to accelerate the model calibration process.

2 MODELLING

This chapter describes the developed models and their implementation into the modelling software. The software is divided into several submodules. The two main modules are the Model Development Kit (MDK) and the Process Simulation Environment (PSE). In the MDK, the models for the individual components (such as capillary, housing wall, compressor) are created. It includes the model equations, parameters, variables and constraints. Subsequently, the individual components are interconnected in the PSE, where the parameters and boundary conditions are set and the simulation is carried out. The research project ECO-COOL included the development of a new and independent model library for the simulation of household refrigerators. In the following section, the models of the most important components are briefly discussed.

2.1 Condenser/Evaporator

Both heat exchangers were basically modelled very similar and differ only in sub-models. The heat exchangers were discretized along the refrigerant pipe into a finite number of control volumes for which the mass and energy conservation equations were solved. The refrigerant flow is assumed to flow through a horizontal pipe. In the single-phase flow region, the pressure loss was calculated according to the Darcy-Weisbach equation and the heat transfer to the pipe wall according to the Dittus-Bölter-Kraussold correlation, see VDI (2013). In the two-phase region, heat transfer and pressure loss are calculated by more complex models based on flow pattern maps. The calculation of the flow form and the volumetric vapor content was modelled according to El Hajal et al. (2003) for the condenser and according to Wojtan et al. (2005) for the evaporator. The heat transfer coefficients of the condenser were calculated according to Thome et al. (2003) and of the evaporator according to Wojtan et al. (2005). The pressure loss was modelled with correlations according to Quiben and Thome (2007). A detailed description of the heat exchanger modelling can be found in Berger et al. (2014a) and Berger et al. (2014b). Different flow patterns lead to jumps in the heat transfer and pressure loss functions of the models. As a result, the implicit solution process does not converge. Therefore, additional linear transition functions had to be implemented in order to get continuous functions.

2.2 Compressor

The compressor model is divided into three modules, housing, oil sump and compressor. The compressor was mainly modelled according to Li (2012) and the fit parameters were calibrated with calorimeter measurements. The housing model includes thermal masses and the refrigerant of the suction and discharge side between which heat is transferred. The heat transfer coefficients were determined experimentally and numerically. The oil sump was modeled after Philipp (2002) and Neto and Barbosa (2010). The absorption and desorption of the refrigerant by the oil is modelled after Equation (1) . Where $\mu = \mathrm{m_{ref}}/m_{oil}$ is the mass ratio between oil and dissolved refrigerant and μ_{sat} is the mass ratio at saturation. The time constant τ is an empirical model parameter determined for absorption and desorption when the compressor is on and off, respectively.

$$\mu = \frac{1}{\tau}(\mu_{sat} - \mu) \tag{1}$$

2.3 Capillary

In order to simulate the capillary and the internal heat exchanger (IHX) accurately, a high computational effort is necessary. Hence a calculation within the simulation program after each calculation step is not practical. Therefore an Artificial Neuronal Network (ANN) was trained with data from a validated one-dimensional homogeneous model of the capillary and the IHX. Around 40,000 simulations with randomly generated boundary conditions were carried out to train the ANN. This makes it possible to calculate the state of the capillary for a wide range of boundary conditions without performing the computationally intensive capillary simulation, see Heimel et al. (2014) and Heimel (2015).

2.4 Further models and overall system

Figure 1 shows the structure of the freezer in the graphical simulation environment. The evaporator and the condenser were modelled by 20 cells respectively. One evaporator cell was designed as collector. The substance data for the refrigerant R600a have been taken from the REFPROP mass spectrometry program, see Lemmon et al. (2010). The walls of the cooling device are modelled by multiple layers. Another important system component is the temperature sensor, which is used to control the system. The graphical user interface makes it very easy to adapt or expand the overall-cycle.

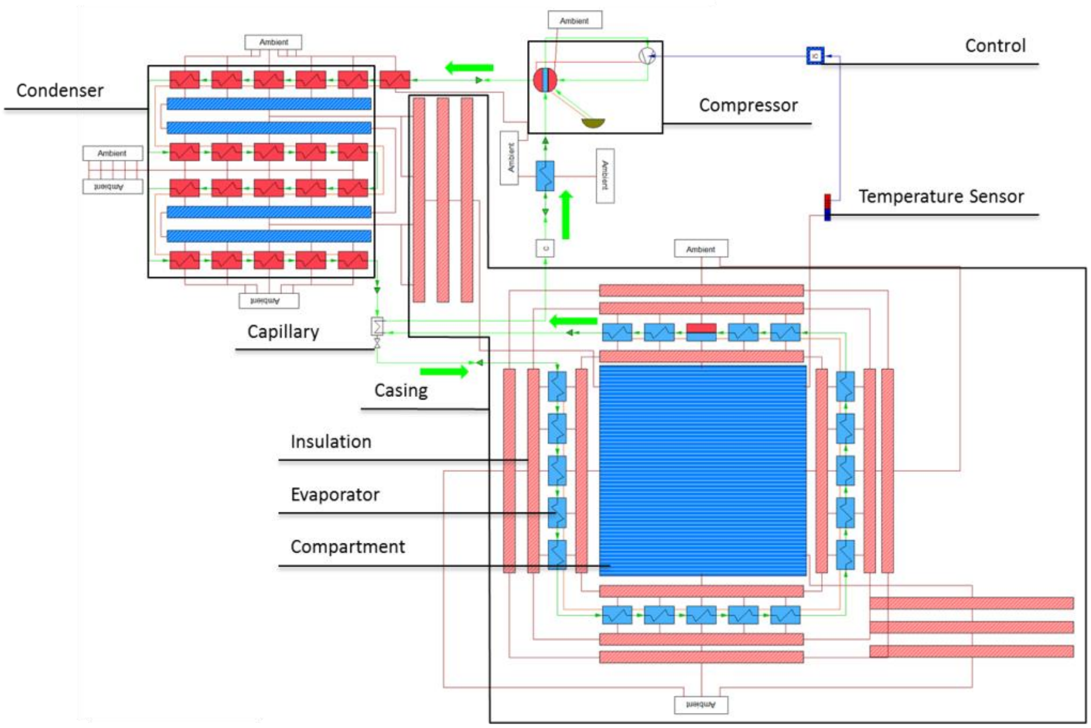


Figure 1: Flow sheet model of the household freezer in the modelling software

3 SEMI-AUTOMATIC CALIBRATION

The overall model of the freezer was built in the simulation software and the model parameters derived from geometry, product specifications and characteristic diagrams of the individual components. In order to validate the model and, if necessary, to compensate incorrect model assumptions, it must be calibrated with measurement results. Since the compressor is operated in cyclic on / off mode, the model must also be validated for dynamic time-dependent operation.

Already during the development of the research code in VBA it turned out, that the calibration of the simulation model is very difficult and time-consuming. Due to interactions and nonlinear dependencies between the model parameters, a manual parameter optimization would be very time consuming and inefficient. However, to be able to use the software for development purposes in industry, a fast and easy calibration of the simulation models is of great importance. For these reasons, a semi-automatic calibration method has been developed. This method is basically divided into two steps: automatic calibration for steady-state and manual calibration for transient operation of the system. Therefore, in addition to the usual transient energy consumption measurement, a further measurement for steady state operation is required. For this purpose, the control system of the compressor was bypassed and switched to continuous operation in order to reach a steady state condition.

Table 1 contains the fifteen measured quantities which were obtained in steady-state and transient tests and used to calibrate the simulation model. In addition to the pipe temperatures along the refrigerant circuit, the high and low pressure at the compressor, the compressor capacity, the ambient temperature and the sensor temperature in the interior were measured. In order to calibrate the simulation model, suitable model parameters (e.g. parameters with high uncertainty) must be selected. In this case, the heat transfer coefficients on the surfaces are associated with high uncertainties as they cannot be fully mapped in the abstracted 0d / 1d model. Table 2 includes ten parameters which were selected for calibration. The parameters are divided into two main groups. The parameter set x_s are determined in the first step of the automatic steady-state calibration. x_s includes heat transfer rates and calibration factors for the pipe pressure losses. The second parameter set x_d includes parameters which have no influence on the result of the steady-state simulation. They are used to calibrate the transient simulation. For example the thermal inertia of the temperature sensor influences the transient behaviour of the system significantly as it is related to the running time of the compressor.

Table 1: Measured quantities

Measured data	Unit	Description
P_{Comp}	W	Electric power of compressor
t_{Comp_surf}	°C	Surface temperature of compressor
t_{Comp_out}	°C	Compressor outlet pipe temperature
t_{Cond_in}	°C	Condenser inlet pipe temperature
t_{Cond_25}	°C	Condenser pipe temperature after 25% length
t_{Cond_50}	°C	Condenser pipe temperature after 50% length
t_{Cond_75}	°C	Condenser pipe temperature after 75% length
t_{Cond_out}	°C	Condenser outlet pipe temperature
t_{Cap_out}	°C	Capillary outlet pipe temperature
t_{Evap_out}	°C	Evaporator outlet pipe temperature
t_{Comp_in}	°C	Compressor inlet pipe temperature
p_{Comp_in}	bar	Compressor inlet pressure
p_{Comp_out}	bar	Compressor outlet pressure
t_{Sensor}	°C	Compartment temperature
$t_{Ambient}$	°C	Ambient temperature

Table 2: Uncertain parameters.

	Parameter	Unit	Description
	α_{Iso_out}	W/(m²K)	HTC between insulation and ambient
	α_{Comp_in}	W/(m²K)	HTC between compartment and insulation
	α_{Dis_Shell}	W/(m²K)	HTC between discharge side refrigerant and shell wall
x_s	α_{Amb_Shell}	W/(m²K)	HTC between ambient and shell wall
	$\eta_{\Delta p\_evap}$	-	Pressure loss calibration factor of the evaporator
	$\eta_{\Delta p\_Cond}$	-	Pressure loss calibration factor of the condenser
	η_{α_Cond}	-	Heat flux calibration factor (condenser pipe and ambient)
	$(\rho cV)_s$	J/K	Thermal inertia of the temperature sensor
x_d	C_{Acc_1}	-	Calibration factor of the accumulator model
	C_{Acc_2}	s/bar	Calibration factor of the accumulator model

3.1 Steady state stage

Calibrating simulation models with measurement results is a task that is required in almost every application of simulation tools. For this reason, an attempt was made to develop an efficient and universally applicable calibration method. This method is based on the Gauss-Newton method, which is a method for solving nonlinear minimization problems, see Deuflhard and Hohmann (2002). The aim of the method is to determine the uncertain parameters $x \in \mathbb{R}^n$ in order to get the best possible fit between $\tilde{y}_j, b \in \mathbb{R}^m$ the simulation results and the measured values, respectively. Where $m > n$, which means that more measured values are required than parameters to be optimized. The deviations of the simulation results $\tilde{y}_j$ from the measured values b_j are called residuals $r_j = w_j(\tilde{y}_j - b_j)$. With the weights w_j, the measurement accuracies and the relevance of the measured quantities can be taken into account. The residuals are summarized with the vector $R(x) = (r_1, r_2, \dots r_n) \in \mathbb{R}^n$. The cost function $\phi(x)$ corresponds to the sum of the squares and has to be minimized in order to increase the fit between simulation and measurement.

$$\min \phi(x) = \min \frac{1}{2} R(x)^{\mathrm{T}} R(x) \tag{2}$$

Therefore, we need to set the derivative $\nabla\phi(x)$ to zero. The derivative of the cost function can be calculated using the Jacobian matrix $R'(x) \in \mathbb{R}^{n\times m}$.

$$0 = \nabla\phi(x) = R'(x)^{\mathrm{T}} R(x) \tag{3}$$

Furthermore, the second derivative of $\phi(x)$ is required.

$$\nabla^2\phi(x) = R'(x)^{\mathrm{T}} R'(x) + R''(x)^{\mathrm{T}} R(x) \tag{4}$$

Since the second derivative $R''(x)$ is associated with considerable computational effort in most applications, the second term of equation (4) is neglected, leading to the Gauss-Newton iteration as given in equation (5).

$$x^{i+1} = x^i - \left(R'(x^i)^{\mathrm{T}} R'(x^i) \right)^{-1} R'(x^i)^{\mathrm{T}} R(x^i) \tag{5}$$

The Jacobian matrix is given in equation (6).

$$R'(x) = WD \frac{\partial y}{\partial x}(x) \tag{6}$$

The diagonal matrix of the weights $W = diag\left(w_j\right) \in \mathbb{R}^{n\times n}$ is known. The matrix D relates the observed variables to the vector of all variables $y \in \mathbb{R}^N$ by the equation $\tilde{y} = Dy$. The matrix $\partial y/\partial x\ (x) \in \mathbb{R}^{N\times m}$ is also referred to as sensitivity matrix and can be calculated with the implicit function theorem as follows:

$$\frac{\partial y}{\partial x}(x) = -\left(\frac{\partial F}{\partial y}(x,y)\right)^{-1} \frac{\partial F}{\partial x}(x,y) \tag{7}$$

The partial derivatives $\partial F/\partial y\ (x,y)$ of the implicit functions $F(x,y)$ are already calculated in the modelling software when solving the system of equations using the Newton method and thus only need to be read out. To calculate the derivatives $\partial F/\partial x\ (x,y)$, the solver has been extended. Thus, all values needed for the Gauss-Newton method can be determined during the solving of the equation system.

The Gauss-Newton method does not pay attention to parameter constraints. This could lead to physically senseless parameter values, for example negative heat transfer coefficients. Therefore, so-called penalty terms have been introduced, see Luenberger and Ye (2008). These terms add a positive value to the cost function when a parameter leaves its defined limits. Thus, the parameter set is no longer optimal. The penalty term is determined by the limits $x_{\min}$, $x_{\max}$ and the two parameters $\epsilon_{\min}, \epsilon_{\max} > 0$.

$$\psi(x) = \begin{cases} \dfrac{x - (x_{\max} - \epsilon_{\max})}{\epsilon_{\max}} & if \;\; x > x_{\max} - \epsilon_{\max} \\ \dfrac{x - (x_{\min} + \epsilon_{\min})}{\epsilon_{\min}} & if \;\; x < x_{\min} + \epsilon_{\min} \\ 0 \;\; else \end{cases} \tag{8}$$

Finally the penalty term $p(x)$ reads as

$$p(x) = a\,\psi(x)^4 \tag{9}$$

with the parameter $a > 0$ to scale the penalty term.

The Gauss-Newton method does not guarantee an absolute minimum. Furthermore, the solution can get stuck in a local minimum. In order to exclude a local minimum, several optimizations with different starting values of the parameters x were carried out. In all simulations, the system converged to the same final state, so that it was very likely that a global minimum was found. Overall, the implemented method leads to a fast and robust optimization of the uncertain parameters.

3.2 Transient Stage

After static calibration, the three remaining parameters x_d need to be optimized. In principle, automatic optimization using the Gauss-Newton method would also be possible for transient simulation, but the computational effort for a single evaluation would be significantly higher.

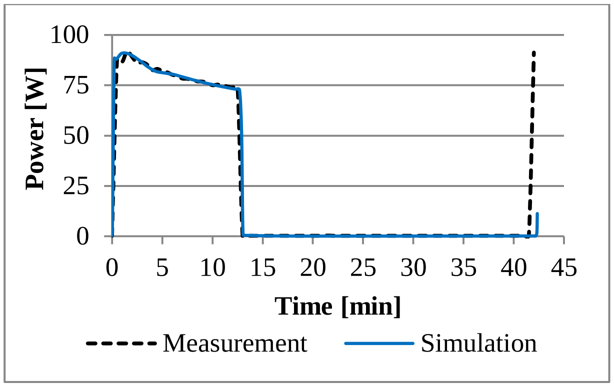
Describing the dynamic behaviour of the temperature sensor is a crucial part, as it is related to the operation time of the compressor. A precise modelling of the sensor is not possible in the 0d / 1d model, since the compartment is modelled as a point mass the temperature stratification is neglected. Additionally, the sensor is in contact with the compartment wall which leads to further heat fluxes. As a result, the behaviour of the sensor can hardly be modelled. One calibration parameter of the transient model is the thermal inertia $(\rho c V)_{\mathrm{S}}$. Therefore, $(\rho c V)_{\mathrm{S}}$ is optimized such that the operation time of the compressor coincides with the measurement data. The other two optimization parameters of the transient model concern the accumulator. Although no accumulator is installed in the examined freezer, it is needed for the simulation model. The reason for this is that the gravity and the alignment of the evaporator pipe in the real device cause a collector effect and thus liquid refrigerant can accumulate. In the simulation model, which was developed for horizontal pipes, this effect would be lost. Energy consumption measurement at 25 °C ambient temperature and temperature sensor thresholds at -20 °C and -20.5 °C are the basis for the calibration. During the measurement, neither additional cooling packages were added nor the appliance door was opened, resulting in a cyclic state of the device.

4 RESULTS

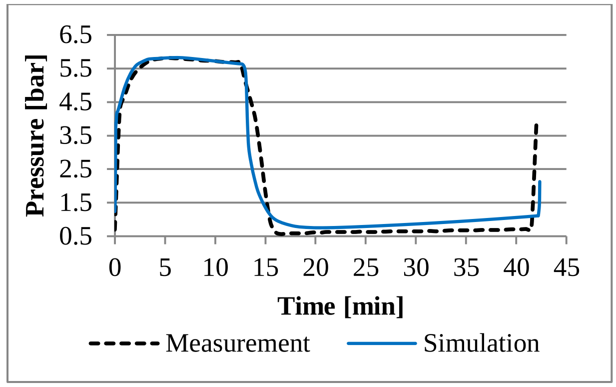
Table 3 shows the results of steady-state simulations compared to measurement data. Although the simulation results with non-optimized parameters were far away from the target values, the automatic calibration converged quickly. The temperature at the inlet of the condenser deviates significantly from the measurement results. This is due to the discretization of the condenser pipe. In the simulation, the temperature is averaged for the first condenser cell, which is around 0.5 m long. At the beginning of the condenser, the temperature gradient along the length of the pipe is relatively high due to the single-phase flow. This high temperature gradient cannot be captured very accurately by the coarse discretisation. All other simulation results agree well with the measurement data after the calibration.

Table 3: Steady-state simulated and measured values at 25°C ambient temperatures

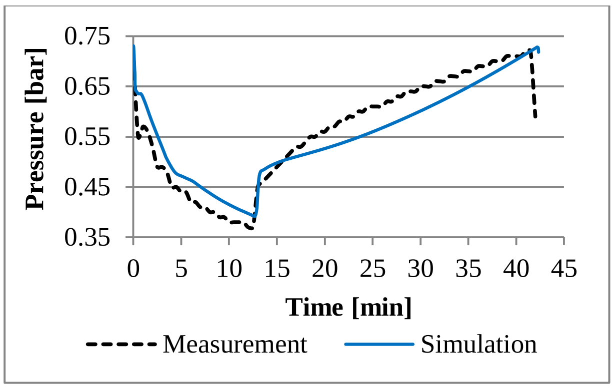
Measuring data	Unit	Simulation before opt.	Simulation after opt.	Measurement
P_{Comp}	W	142.10	58.78	57.87
t_{Comp_surf}	°C	104.37	63.99	62.25
t_{Comp_out}	°C	156.02	66.94	67.28
t_{Cond_in}	°C	150.54	52.90	58.36
t_{Cond_25}	°C	123.68	35.78	37.42
t_{Cond_50}	°C	106.03	36.61	36.57
t_{Cond_75}	°C	102.6	36.56	35.99
t_{Cond_out}	°C	102.5	36.54	35.59
t_{Cap_out}	°C	-10.8	-35.18	-35.73
t_{Evap_out}	°C	-13.08	-40.59	-40.78
t_{Comp_in}	°C	64.03	26.31	27.31
p_{Comp_in}	bar	0.961	0.278	0.238
p_{Comp_out}	bar	20.989	4.894	4.879
t_{Sensor}	°C	-7.74	-34.26	-34.85



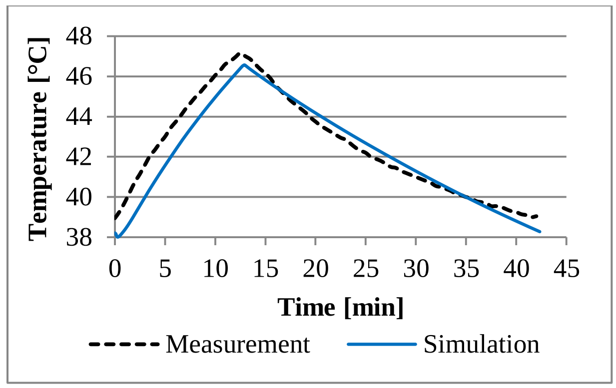
(a) Compressor electric power



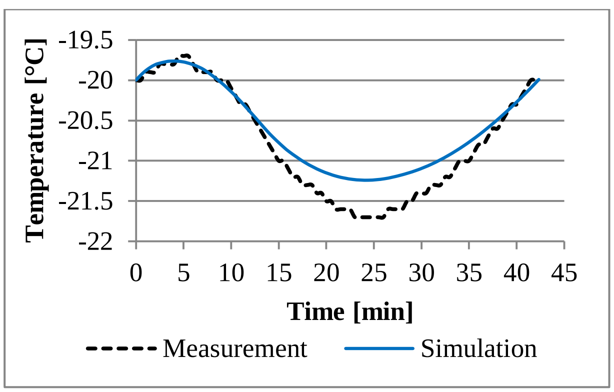
(b) Pressure at compressor outlet



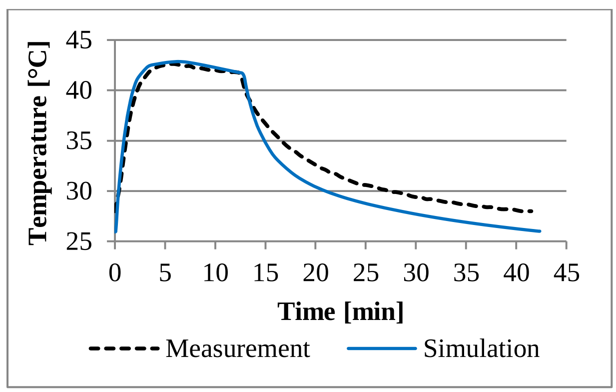
(c) Pressure at compressor inlet



(d) Surface temperature of compressor



(e) Sensor temperature



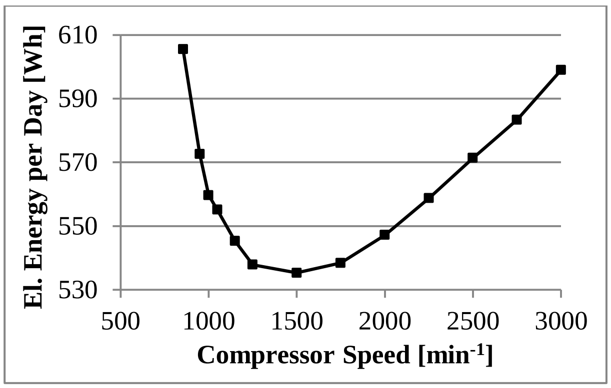
(f) Condenser pipe temperature after 50% length

Figure 2: Transient simulation results compared to measurement data of a periodic cycle

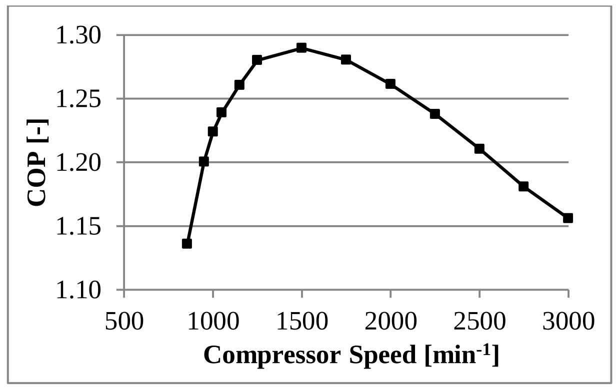
Transient simulation results compared to measurement data can be seen in Figure 2. The simulation result of the electric power consumption of the compressor (Figure 2a) agrees very well with the measured values. Similarly, the duration of the on- and off-cycle could be mapped very well. During operation of the compressor, the discharge pressure correspondence is very good (Figure 2b). The suction pressure shows a similar trend but the simulation results are around 0.05 bar higher than the measurements (Figure 2c). When the compressor is switched off, the suction and discharge pressure are aligning, which can be seen in the measurement data. In the simulation, the compressor speed during an off-cycle was set to 20 rpm as due to numerical reasons, the mass flow in the simulation cannot be set to zero. Therefore, there is no complete pressure alignment between evaporator and condenser. The pipe temperature of the condenser also deviates during the off-cycle (Figure 2f). One of the reasons is the temperature dependence of the natural convection, which was not considered in the model so far. The measured temperature of the sensor decreases after switching off the compressor (Figure 2e) which is less intensive in the simulation. However, within the threshold (-20 °C and -20.5 °C), which is important for the simulation, the measurement and simulation are in good agreement.

5 PARAMETER STUDIES COMPRESSOR SPEED

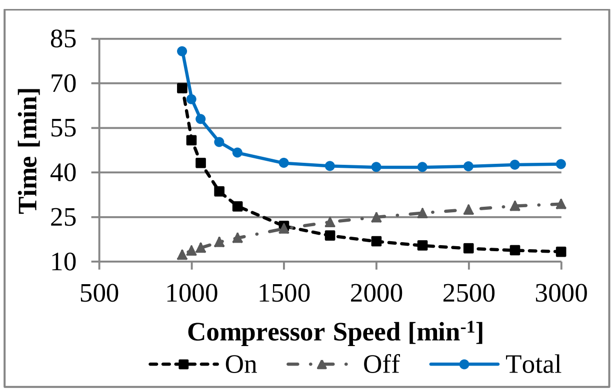
A numerical model of the entire cooling unit is particularly suitable for carrying out parameter studies as well as the estimation of a potential, quickly and inexpensively. At this point, a parameter study is presented as an example in which the speed of the compressor was varied. For this purpose thirteen simulations with compressor speeds between 800 rpm and 3000 rpm were carried out. During one simulation the compressor speed remained constant. It should be noted that the model parameters of the compressor were determined at the highest speed of 3000 rpm. It can be assumed that for the lower speed levels, the compressor model is not accurate anymore. However, it is sufficient to estimate a trend.



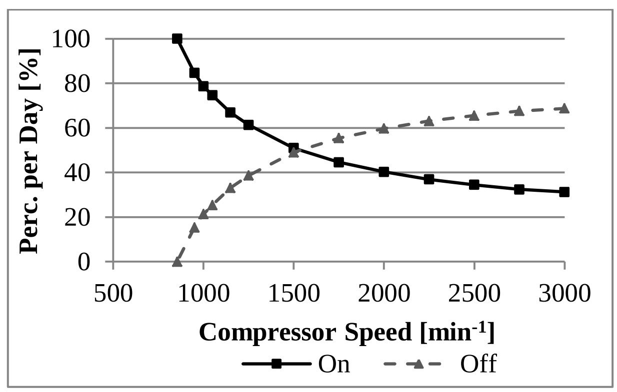
(a) Electric work per day



(b) Coefficient of Performance



(c) Time per cycle



(d) On / Off-Percentage per day

Figure 3: Simulation results at different compressor speeds

All simulations were performed until a periodically constant cycle was reached. Figure 3 shows the results of the parameter study. The reduction of the compressor speed from 3000 rpm to 1500 rpm reduces the electrical energy consumption by approximately 10.6 %. Figure 3c shows the duration of the on- and off-cycle of a periodically constant cycle at different compressor speeds. As expected, the on-cycle duration of the compressor decreases as the compressor speed increases, but the total cycle duration is almost constant for compressor speeds between 1500 rpm and 3000 rpm. The reason lies in the thermal inertia of the temperature sensor. Due to the delayed reaction of the temperature sensor, the averaged compartment air temperature is lower at higher compressor speeds. As a result, it takes longer until the switch-on temperature of the compressor is reached again. As compressor speed decreases below 1500 rpm the on-cycle duration increases very strong. This leads to a higher total cycle duration. At about

900 rpm the compressor must not switch off in order to keep the compartment air temperature within the threshold. At these low speeds, the system becomes very inefficient.

6 CONCLUSION AND OUTLOOK

A model package for the modelling of household refrigerators was developed and implemented into a commercial process modelling software. Thus, a simulation model of a freezer was built and calibrated with a newly developed semi-automatic method. It is based on the Gauss-Newton method and can easily be adapted to other devices. Subsequently, the model will be applied to other devices and the underlying models will be further enhanced to allow efficient and meaningful use during the development process of new refrigerators and freezers.

NOMENCLATURE

a	Scale factor of the penalty function	(–)	W	Weight vector	(–)
c	Specific heat capacity	(J/kgK)	x	Uncertain parameters	(–)
C	Calibration factor	(–)	y	Simulation results	(–)
F	Function	(–)	α	Heat transfer coefficient	(W/m^{2}K)
HTC	Heat transfer coefficient	(W/m^{2}K)	Δ	Difference	(–)
IHX	Internal heat exchanger	(–)	ϵ	Parameter of penalty function	(–)
p	Pressure	(bar)	η	Calibration factor	(–)
p(x)	Penalty function	(–)	μ	Mass-ratio	(kg/kg)
P	Electric power	(W)	ρ	Density	(kg/m^3)
r	Residual	(–)	τ	Time constante	(s)
R	Residual vector	(–)	ϕ	Cost function	(–)
t	Temperature	(°C)	ψ	Penalty function	(–)
V	Volume	(m^3)	∇	Nabla operator	(–)
w	Weight factor	(–)			

Subscript

ref	Refrigerant
S	Sensor
i	Iteration
j	Index matrix element
min	Minimum
max	Maximum
sat	Saturation

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