A Modelica Library for Simulation of Household Refrigeration Appliances Features and Experiences

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Abstract

Today the vast majority of household refrigerators and freezers work with on-off-control of the compressor. This control method leads to significant dynamic behaviour of the refrigeration cycle and cabinet. Moreover, some phenomena cause energy loss during the off-state. The development of new appliances based on static design and simulation tools neglecting all those dynamic effects will not yield optimal results.
Therefore, a Modelica library for the simulation of household refrigerators and freezers was developed. The modelling of the vapour compression cycle is based on the Modelica Media and Fluid library. The heat exchangers are modelled according to the finite volume method. A media model based on quadratic equations was developed for the refrigerant R600a (Isobutane) to allow fast calculations. The modelling of the refrigeration cycle includes effects like different void fraction models and refrigerant sorption in the compressor lubricant.
The library was validated for a one temperature zone refrigerator and a refrigeration-freezer-combination with evaporators connected in series. Adopted script functions have been written to allow parameter runs with one or two parameters or with different compressors. The library combines model freatures of recent studies Janssen [2], Philipp [5] and Radermacher [7] with a high degree of flexibility and a parameterization by data which can be obtained easily.
Keywords: household refrigeration appliances; vapour compression cycle; R600a

1 Introduction

In Germany 15% of the total CO_2-emmisions are caused by the domestic sector. According to data of the VDEW, about 30% of the electrical energy consumption of the domestic sector are used by refrigerators and freezers. Achieving a better energy efficiency will contribute to a decrease of the CO_2-emmisions in mid term.
Most of the household refrigerators and freezers operates with on-off-control of the compressor. A variable speed compressor can achieve higher energy efficiency (up to 25% less energy consumption). But still nowadays variable speed compressors requires an higher investment from the consumer side.
Due to the on-off-control, the refrigeration cycle and the cabinet show a distinct dynamic behaviour most of the time. Therefore, the use of static design and simulation tools will not lead into optimal results. Particularly, some effects caused by the on-off-control, e.g. the refrigerant migration, are neglected.
Hence, dynamic simulations represent an essential tool to achieve further improvements. The aim of this project was to develop of a Modelica Library for the optimization of household refrigerators and freezers. Importance was given to model parameterization is possible with data which are easy to obtain for the end-user of the library. To get good results void fraction models for the two phase flow and the effect of the refrigerant sorption in the compressor lubricant were implemented.

2 Components and Structure

A schematic drawing of a household refrigerator with the simplest cycle option is shown in Fig. 1.
The main components of the systems are the hermetic piston compressor, the condenser, the non adiabatic capillary tube with the suction line heat exchanger, the evaporator and the cabinet. More complex cycles are realized, e.g. two or more evaporators in series or parallel with magnetic valve. Optional components like stop valves and frame heating are included in the li-

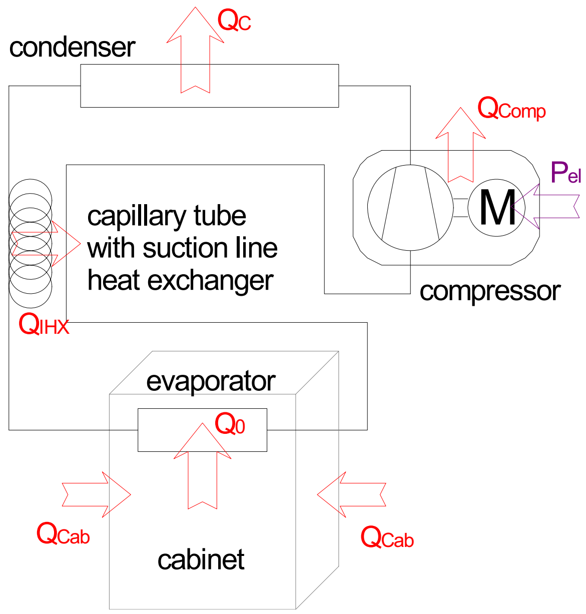

Figure 1: Schematic drawing of a houshold refrigerator with one temperature zone

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2.1 Hermetic Piston Compressor

Hermetic compressors of household appliances have a dynamic behaviour with considerable larger time constants compared to refrigeration systems in the commercial and industrial sector. Since for a decreasing norm capacity the weight only decreases slightly.
The ratio of the heat capacity of compressor to the electrical power consumption

$$\tau_{Therm,Comp} = \frac{C_{p,Comp}}{P_{el}} \qquad (1)$$

can be used to compare the dynamic of the thermal behaviour of compressors, approximately. Using the heat flow of the compressed gas and the rejected heat by the motor to the compressor instead of the electrical power consumption would be more exact, but requires more data of the compressors.
For hermetic compressors of household refrigeration appliances $\tau_{Therm,Comp}$ is in the range of 30 to 60 s. Compressors of the commercial sector, e.g. for supermarket refrigeration applications ($\dot{V}_{Gasflow} = 50m^3/h$) $\tau_{Therm,Comp}$ is about 1 s.
Furthermore the refrigerant charge of the whole system is very small compared to the lubricant charge of the compressor. An average refrigerant charge of an refrigerator with R600a as refrigerant is approximately 30 g. Lubricant charge of hermetic compressors is about 220 g.
Hence, the dynamic behaviour of the compressor should be taken into account. For this purpose a detailed thermal model of the compressor and its subcomponents have been realized as shown in Fig. 3.

Figure 2: Hermetic piston compressor for household refrigerators with opened shell

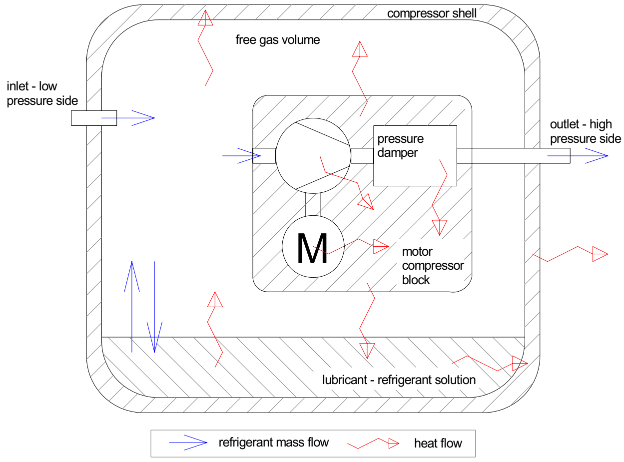

Figure 3: Schematic drawing of a hermetic compressor and existing mass and heat flows

2.1.1 Refrigerant Sorption in the Lubricant

The saturation solubility of the refrigerant in the lubricant ζ_{Sat} depends on the temperature T_{Lub} and the pressure p_{Lub} of the lubricant and the type of the lubricant. Mineral and polyolester oils are applied as lubricants. Fig. 4 shows the saturation solubility as function of temperature for different pressures of a polyolester oil. The implementation in Modelica is done by using an approach

$$\zeta_{Sat} = \frac{(c_1 \cdot p_{Lub})^{c_2}}{T_{Lub} + c_3} \qquad (2)$$

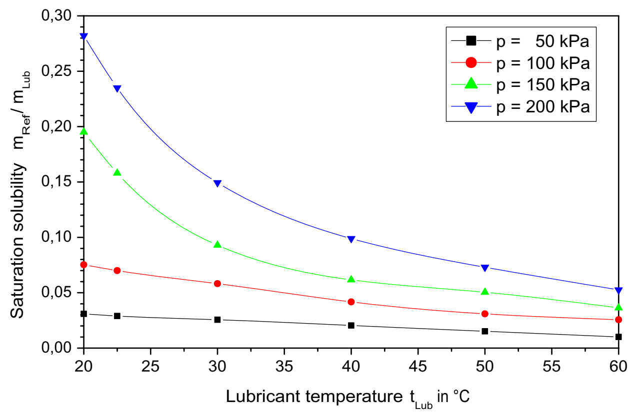

Figure 4: Saturation solubility of a used refrigerant - lubricant mixture

The parameters c_1, c_2, c_3 are fitted for applied lubricant - refrigerant systems based on data of [1] and further measurements.

T_{Lub}, p_{Lub} vary distinctly with the on-off-state of the compressor. The absorption and desorption process depends on the difference of the saturation solubility and the absorbed refrigerant.

$$\frac{d\zeta}{d\tau} = \frac{1}{\tau_{Sorp}} \left(\zeta_{Sat} - \zeta\right) \tag{3}$$

The circulation of the lubricant varies during on and off states of the compressor. Hence different time constants τ_{Sorp} are applied as described in Philipp [5].

2.1.2 Calculation of Mass Flow Rate and Efficiency

For the parameterization of a detailed volumetric efficiency model based on Eq. 4

$$\dot{m} = \rho_{suc} \cdot V_{Dis} \cdot \lambda \left(p_C, p_E\right) \cdot n \tag{4}$$

it is necessary to know about the displacement volume V_{Dis}, the revolution speed n, the exact inlet conditions into the cylinder ρ_{Suc} and the characteristic behaviour of the volumetric efficiency λ. The characteristics which can be obtained for hermetic compressors are usually limited to a data set of one or more points of the condensation temperature t_C, the evaporation temperature t_E, the refrigeration capacity $\dot{Q}_0$ and the electrical power consumption P_{el}.

Hence, for simulation purpose V_{Dis}, λ and n are combined:

$$\Lambda \left(p_C, p_E\right) = V_{Dis} \cdot \lambda \left(p_C, p_E\right) \cdot n \tag{5}$$

Figure 5: Tube-and-wire condenser

Λ is described by an cubic approach of pressure ratio:

$$\Lambda \left(p_C, p_E\right) = \sum_{i=1}^{4} c_i \left(\frac{p_C}{p_E}\right)^{i-1} \tag{6}$$

The coefficients are determined during initialization by least square method using the given data. ρ_{Suc} must be corrected, owing to the higher temperature in the compressor shell.

The electrical power consumption is calculated by the isentropic power P_{is} and the isentropic, mechanical and electrical efficiencies η_{is}, η_{mech}, η_{el}.

$$P_{el} = \frac{P_{is}}{\eta_{is} \cdot \eta_{mech} \cdot \eta_{el}} \tag{7}$$

For the isentropic power consumption, a simplified approach according to Lippold [3] is used: P_{is} depends on p_C, p_E, $\dot{m}$, ρ_{Suc} and the isentropic exponent κ. The efficiencies are combined for the same reason as above mentioned. For the combined efficiency η a cubic approach in respect to P_{is} is applied. Coefficients are determined during the initialization.

2.2 Heat Exchanger

The main heat exchangers are modelled according to the finite volume method (FVM) used in the ThermoFluid Library [9]. Heat transfer coefficients at the refrigerant side are calculated according to Shah [8]. Heat transfer at air side is separated in convection and radiation. Replaceable components are used to consider different types of evaporator and condenser.

2.2.1 Modelling of Two Phase Flow

The void fraction models of the two phase flow were paid attention due to the great influence on the system behaviour. Reasons for the high influence are the low mass fluxes, the geometry of the heat exchangers (particularly the evaporator) and the capillary tube as fixed vessel in household refrigeration appliances. Void fraction models according to Hughmark, Lockhardt-Martinelli and Premoli were applied.

The application of void fraction models increases the computation effort significantly. The void fraction model according to Premoli [6] with correction factors of the liquid volume fraction according to Janssen [2] provided the most stable simulations with good results. The liquid fraction depends on the pressure p, the saturation densities ρ', ρ'', the dynamic saturation viscosities η', η'', the vapour quality x, the hydraulic diameter d_H, the mass flux G and a correction factor depending on geometry c_G.

Based on the vapour volume fraction of the gas phase $\alpha = V''/V$ of homogenous two phase flow a slip factor s is introduced.

$$\alpha = \left(1 + \frac{1-x}{x}\frac{\rho''}{\rho'} \cdot s\right)^{-1} \tag{8}$$

The slip s is calculated by

$$s = 1 + K\sqrt{\frac{y}{1+Cy} - Cy} \tag{9}$$

$$y = \frac{x}{1-x} \tag{10}$$

The parameters K and C correlates to the flow conditions and are calculated by the Reynolds Number Re and the Weber Number We.

$$Re = \frac{G \cdot d_H}{\eta' + x(\eta'' - \eta')} \tag{11}$$

$$We = \frac{G^2 \cdot d_H}{\sigma} \tag{12}$$

$$K = 1{,}578 \cdot Re^{-0{,}19}\left(\frac{\rho'}{\rho''}\right)^{0{,}22} \tag{13}$$

$$C = 0{,}0273 \cdot WeRe^{-0{,}51}\left(\frac{\rho'}{\rho''}\right)^{-0{,}08} \tag{14}$$

To introduce the correction factor for the liquid volume fraction c_{lvf} proposed by Janssen [2] equation (8) has to be rewritten as

$$\alpha = 1 - c_{lvf}\left[1 - \left(1 + \frac{1-x}{x}\frac{\rho''}{\rho'} \cdot s\right)^{-1}\right] \tag{15}$$

The top of Fig. 6 shows the refrigerant charge in the evaporator and suction line calculated by homogenous two phase flow and by void fraction model acc. to Premoli with a correction factor for the liquid fraction for a total refrigerant charge of 36 g. Refrigerant charge calculated by homogenous two phase flow model is only in a range of 10 to 20% of the charge calculated by the void fraction model. A further difference can be identified in the evaporator outlet conditions (Fig. 6-bottom). To get reliable results, a void fraction model has to be applied.

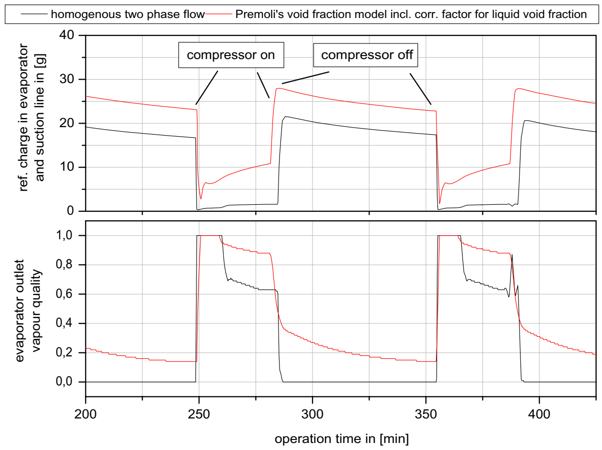

Figure 6: Refrigerant charge in the evaporator and suction line (top) and evaporator outlet conditions (bottom) for different two phase flow models

2.3 Capillary Tube with Suction Line HX

The main part of capillary tube is inside the suction line tube and acts as an internal heat exchanger. The internal heat exchanger ensures superheated inlet conditions at the compressor and has a positive effect on the cycle efficiency.

The mass flow rate of the capillary tube $\dot{m}_{Cap}$ is calculated by a function of p_C, p_E, the inlet vapour quality x_{in} or the degree of subcooling Δt_{sc}, the capillary length and the diameter L_{Cap}, d_{Cap} and the heat flow rate between capillary tube and suction line $\dot{Q}_{IHX}$.

$$\dot{m}_{Cap} = K_{Geo} \cdot K_{sc} \cdot K_{IHX} \cdot \left[K_1(x_{in})\Delta p + K_2(x_{in})\Delta p^2\right] \tag{16}$$

To calculate the heat transfer $\dot{Q}_{IHX}$, the capillary tube is modelled according to the finite volume method. As the refrigerant state is in the two phase region, the temperature can be calculated from the pressure. The pressure in each volume is determined by dimensionless pressure distribution correlation according to Philipp [5].

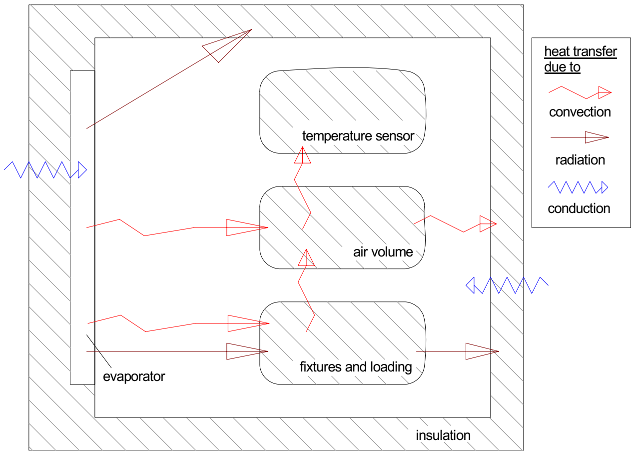

Figure 7: Heat capacities and heat flows in a one temperature zone cabinet

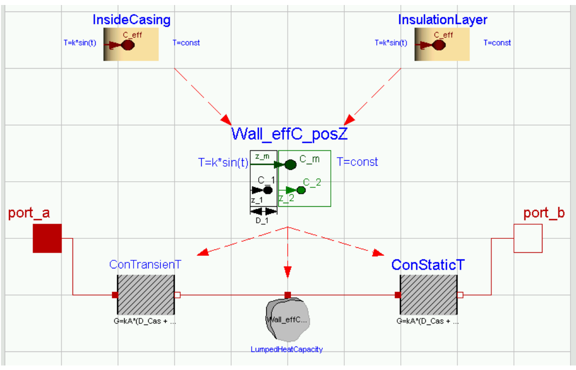

Figure 8: Wall model with calculation of the effective heat capacity

2.4 Cabinet

The cabinet model contains different heat capacity elements which interact via heat transfer due to convection, radiation and conduction. Fig. 7 shows the main heat capacities and heat flows.

For the cabinet wall two models are implemented. In the first model the wall is discretized into a series of heat capacity and conduction elements. The distribution of the parameters has to take the structure of the wall into account, e.g. the inner casing and the insulation.

If the cycling frequency is approximatively known, a simplified model approach can be used. Thus, an effective heat capacity and its position between two heat conduction model is calculated (Fig. 8). By this way the number of state variables can be reduced.

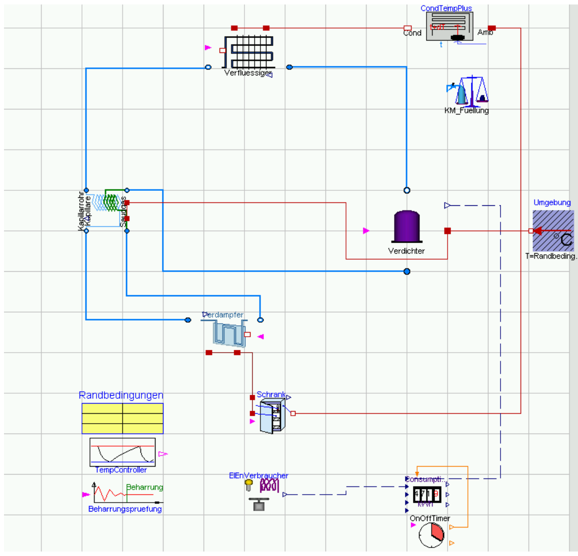

Figure 9: Complete refrigerator model in Dymola

2.5 Refrigerator

All components are connected via fluid ports of the Modelica Fluid Library and heat ports of the Modelica library. Heat ports are separated into convection and radiation heat transfer, if necessary. The cabinet model provides an output signal of the cabinet temperature. Temperature control is done by cabinet and evaporator surface temperature. A complete model is shown in Fig. 9.

2.6 Further Features

To reach a high flexibility, the data inputs for the end-user of each component is put in a replaceable record. The data records are stored as a modification of the basic data record in a top level package outside the main package. This allows fundamental modifications of the components without an impact to the database of the end-user. A data import routine for MS Excel and other databases is under construction.

The component models make high use of the replaceable model feature to allow easy justification of the modelling detail level.

An additional component was developed which allows checking significant variables for steady state. After a selectable number of on-off-cycles in steady state, the component stops the simulation run automatically. Particularly for parameter studies, the computation effort can be decreased efficiently.

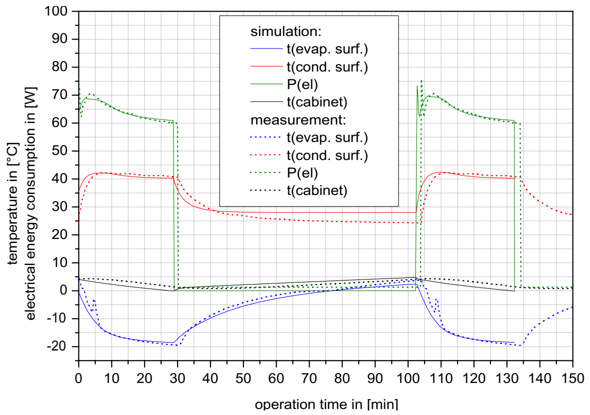

Figure 10: Comparison of a refrigerator system behaviour based on simulation and measurement results

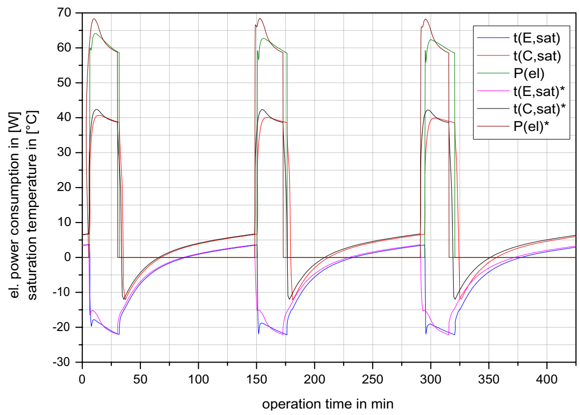

Figure 11: Comparision of simulation including and neglecting the refrigerant sorption

3 Validation

Before the library was used for theoretical investigations, a validation with a one temperature zone refrigerator and a two temperature zone refrigerator - freezer - combination had been performed. Therefore, a series of simulations and measurements with variation of the compressor, the ambient temperatures and the cabinet temperature setting had been done.

Therefore, the ambient temperature where set as a parameter in simulation. The compressor control where done by the temperature output of the cabinet sensor. The parameters of the control hysteresis function of the real refrigerator where applied in simulation to include the influence of thermal inertia of the temperature sensor. For comparison of absolute values (energy consumption, relative compressor operation time, cycling frequency ...) simulation was checked for steady state. Steady state was achieved if all of mentioned values differ less than 0,1 % compared to the previous cycle. For the verification with the measured refrigerator, the average values of three cycles under steady state conditions were applied.

The simulation was in very good agreement in respect of the behaviour of the system and the absolute values like energy consumption and relative compressor operation time. Fig. 10 shows a comparison of simulation and measurement results. Tab. 1 shows the measured and simulated values of energy consumption and relative operation time of the compressor.

The influence of refrigerant soprtion in the compressor lubricant increases with decrasing of ambient temperature and of the relative compressor operation time. Fig. 11 shows the dynamic behaviour of saturation temperature in the evaporator and condenser $t_{E,sat}$, $t_{C,sat}$ and the power consumption $P_{E,sat}$ including the refrigerant sorption and $t_{E,sat}*$, $t_{C,sat}*$ and $P_{E,sat}*$ neglection refrigerant sorption.

4 Experiences

To reproduce the behaviour in the compressor on-state as well as in the compressor off-state adequately precisely the models have to be very detailed. Some of the physical principles only show an influence on one of the both states and can be neglected in the contrary state.

Usually Dassl and Radau IIa were used as integrators. The Dassl integrator shows the best performance with a relative tolerance between $5 \cdot 10^{-6}$ and $1 \cdot 10^{-5}$. Radau IIa, with a relative tolerance between $1 \cdot 10^{-6}$ and $2 \cdot 10^{-6}$, often provides a faster calculation but simulations lead more often into a stiff system. This is generally a common problem. Further decrease of the relative tolerance does not protect against stiff systems. A high computation effort always happens during the change of the compressor state. Fig. 12 shows the calculation time of Dassl and Radau IIa with Dymola.

The proposed state-depending model structure presented by Nytsch-Geusen [4] can be used to have different models depending of the compressor on and off-state. In the compressor on-state, mass flow depends generally on the compressor and the capillary tube characteristic. In the compressor off-state mass flow rate is caused by thermal influences.

As one example: The heat transfer in the capillary tube - suction line heat exchanger can be neglected in the compressor off-state. The finite volume structure, which is necessary in the on-state, can be simplified to only one mass flow - pressure difference correla-

Table 1: Comparision of absolute values of measurement and simulation results

Compressor	tAmbient	tCabinet	Energy Consumption in kWh/d			Relative Compressor Operation Time		
	in °C	in °C	Simulation	Measurement	Δ (rel.)	Simulation	Measurement	Δ (rel.)
Type 1	39	1,0	0,927	1,002	-7,5%	0,536	0,565	-5,1%
Type 1	25	1,0	0,446	0,480	-7,1%	0,288	0,291	-1,0%
Type 1	25	3,5	0,385	0,407	-5,4%	0,244	0,233	4,7%
Type 2	32	1,0	0,643	0,619	3,9%	0,434	0,409	6,1%
Type 2	25	1,0	0,437	0,413	5,8%	0,314	0,293	7,2%
Type 2	25	3,5	0,349	0,333	4,8%	0,251	0,238	5,5%

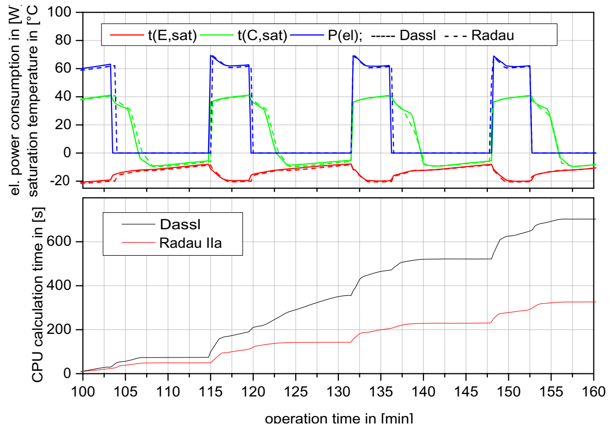

Figure 12: Electrical power consumption, evaporation and condensation temperature and computation time with Dassl and Radau IIa as integrators

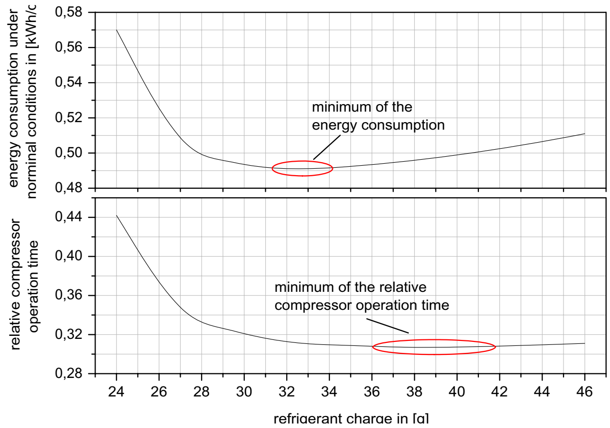

Figure 13: Influence of the refrigerant charge on the energy consumption and the relative compressor operation time

tion with one volume. Hence, calculation effort can be decreased and simulation stability can be increased significantly in the off-state.

5 Issues of Optimization

Effects on different cycle variants, alternative expansion devices and the utilization of stop valves have been investigated with the validated model of the library. Further on, there are some optimization problems which are useful for all refrigerator and freezer models. Getting these results by experimental investigations, means a high effort. Simulation can help to limit that effort efficiently. As examples two optimization problems are presented here. The calculations have been performed on the validated one temperature zone refrigerator.

5.1 Optimal Refrigerant Charge

For a given configuration the energy efficiency varies distinctly from the refrigerant charge. By using a parametric study with Dymola, the optimal refrigerant charge can be found for known ambient and cabinet setting conditions. Fig. 13 shows a significant minimum of the energy consumption for the optimal refrigerant charge in the range of 33 g. The relative compressor operation time shows a wide minimum which can be found for a higher refrigerant charge. Neglecting the refrigerant sorption leads to a calculation results of optimal charge which is between 3 and 10 g lower, depending mainly on ambient temperature and relative compressor operation time.

5.2 Optimal Compressor

There is a wide choice of hermetic compressors for household refrigerators on the market. The compressor with the best COP under nominal conditions is not necessarily the best under real operation conditions. A specially written script function allows the user to compare a set of different compressors. Fig. 14 shows the energy consumption and the relative compressor operation time for a set of different compressors simulated in one refrigerator. The graph identifies one compressor as the optimal solution for the system. The best

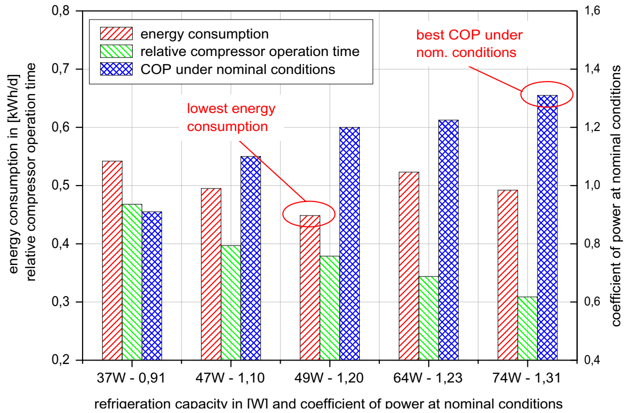

Figure 14: Comparison of energy consumption and relative compressor operation time for a set of compressors

one in the system (COP=1,20) is not the one with the highest COP (1,31).

6 Conclusion and Future Work

A Modelica library for the dynamic simulation of household refrigerators and freezers has been developed. Effects as refrigerant sorption in the compressor lubricant and different void fraction models have been implemented, due to the high influence on the system. The models have been successfully validated and are used for general investigations of the cycle structure as well as for the basic optimization process.
Future work is to be concentrated on the simulation stability, particularly for systems with more temperature zones, as well as for further heat exchanger designs. To obtain a higher stability, the state depended model structure is one issue to work on.

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