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ABSTRACT

To further increase the energy efficiency of domestic refrigeration systems, it is not sufficient anymore to optimize the single cycle components on their own. On the contrary, due to the strong system interactions, an integral consideration of the whole cooling cycle is necessary to capture the system dynamics in detail and to understand them. Just by understanding the occurring physical effects, loss mechanisms can be identified and efficient optimizing measures could be taken.
In this work a transient model for the simulation of the evaporator and the condenser of domestic refrigeration appliances, based on the Finite Volume Method (FVM) is presented. The requirements for cycle simulations differ from design or optimization problems because the models have to be as fast as possible so that the computational results can be used as boundary conditions for the simulation of the other components. Thus, it is challenging to find a satisfying trade-off between the computation speed and the accuracy of the results. The developed algorithm solves the governing equations for mass and energy in combination with some simplifying assumptions (e.g. 1-dimensional fluid flow, negligence of the pressure drop, homogeneous two-phase flow) which have been necessary to keep the computation time within limits.

1. INTRODUCTION & REVIEW

Refrigeration devices – no matter whether they are used for industrial or domestic purposes – are essential for the comfortable daily life we are used to. They are urgently needed for the production, distribution and storage of our food, for various industrial processes or for the air conditioning inside buildings in climate zones where the ambient air is unbearable.
There is no doubt that in future the need for cooling devices will still be increasing due to the worldwide growth of population and prosperity.
With the increase of cooling needs and offered cooling possibilities some serious problems occur because a considerable amount of energy is necessary to power the systems which enable the artificial decreasing of matter's temperature.
Depending on the considered geographic region, between 5% and 29% of the electrical energy end use is used for the operation of refrigerators and freezers (IEA, 2009).

The fundamental design of a typical, nowadays available refrigerator for domestic purposes is based on the vapor compression cycle and it has hardly changed since its invention in the 20th Century. In a thermodynamic cycle, a volatile refrigerant (in modern devices mainly R134a (America) and R600a (Europe)) is continuously compressed and expanded. Primarily, an on/off controlled compressor forces the refrigerant to flow through two heat exchangers

and a capillary tube, which operates as expansion device and so heat can be transferred from an isolated cabinet to the ambient air.
Unfortunately, irreversible losses occur in all subsystems of a refrigerator and thus the actual energy consumption of a refrigerator is higher, than it could be in theory. Although the daily energy consumption of a refrigerator is just about 1kWh/day which is equivalent to the energy consumption of a continuously running 40W light-bulb (Hermes, 2008a), the enormous energy saving potential becomes obvious when looking at the number of appliances which are currently used worldwide. Today, it has been estimated that there is one refrigerator for every six people on earth (Coulomb, 2006).

In the medium term, there will be no alternative cooling technology substituting the vapor compression cycle the standard cooling technology for domestic purposes (Radermacher and Kim, 1996).
Hence, energy saving measures have to be realized by optimizing the present domestic refrigerators. Radermacher and Kim (1996) suggested four general areas to improve the system efficiency of conventional refrigerators: (1) Improving the refrigeration cycle efficiency; (2) decreasing the cabinet heat load; (3) reducing parasitic electrical loads; and (4) reducing the on/off cycling losses.
Furthermore it has to be noted, that each improvement exacts a penalty in terms of increased cost or system complexity and reliability (Radermacher and Kim, 1996).

An efficient approach to improve the refrigerant cycle efficiency is the application of cycle simulation tools because they enable a visualization of the ongoing physical effects. In this way, it is possible to understand loss mechanisms which can be reduced or avoided completely. The challenge in developing a cycle simulation tool is to arrive at a compromise between computation accuracy and computation time because the results of one component provide the boundary condition for the adjacent components. So the single parts of the cycle have to be modeled as simply as possible but as accurately as necessary.

For the reproduction of the complex working process within a heat exchanger in a numeric way various approaches regarding design and optimization of heat exchanger for domestic refrigeration can be found in literature.
Generally, the simulation approaches can be classified into single components (component level analysis) and cycle simulations (system level analysis). Concerning the modeling strategy, three approaches can be found in literature, which differ in modeling complexity and thus in the accuracy of the results. These three are the lumped parameter approach, the moving boundary approach and the distributed parameter approach.

Concerning the component level analysis, McKinley et al. (2008) and Cuevas et al. (2009) used the moving boundary approach in their heat exchanger investigations.
A distributed parameter approach can be found in Jia et al. (1995), Quadir et al. (2002), Bansal et al. (2003) and Hermes et al. (2008b). Jia et al. used a finite volume approach to predict the transient response of an evaporator. The finite element approach was successfully used by Quadir et al. (2002) and Bansal et al. (2003) to model wire-and-tube heat exchangers. The development of a numerical simulation model for plate type roll-bond evaporators using a finite volume approach has been presented by Hermes et al (2008b).
All the mentioned works focus on the optimization of the heat exchanger design. The models are very detailed and thus complex. Their computation is time-consuming but their experimental validations have shown a high degree of accuracy.
For the use in steady state or transient cycle simulation, these kinds of models are far too complex and their computation takes too long. Hence, simplifying assumptions which reduce the accuracy of the simulation results are necessary to keep the computation time within limits.

In 1995, Jakobsen investigated the energy optimization of refrigeration systems. He chose a lumped parameter approach considering the whole heat exchanger as one control volume. Furthermore, Jakobsen neglected the pressure drop within the heat exchangers and he assumed the wall temperatures to be constant over the length.
A moving boundary approach was chosen for example by Janssen et al. (1998) and Negrao et al. (2011). The former developed a dynamic model for small refrigerating systems. They assumed the flow field in the heat exchangers to be one dimensional while the occurrence of the pressure drop and the axial heat conduction were neglected. Negrao et al. (2011) presented energy and cost savings in household refrigerating appliances by using a steady-state simulation-based design approach.
The successful application of a distributed parameter model can be found in Hermes et al. (2008a) in their first-principles simulation model for the start-up and cycling transients of household refrigerators. The authors chose a

finite volume approach to model the heat exchangers. The refrigerant was modeled on the assumptions that the flow is 1-dimensional, the tubes are straight and horizontal and the pressure drop as well as the diffusion effects can be neglected.

The aim of this work was the development of a first and simple heat exchanger model, which can be used for transient cycle investigations of a domestic refrigerator/freezer working with R600a.
By considering the difficulties and advices mentioned in literature, a stable and extendable computation algorithm was developed, which was consciously kept quite simple.
Based on the experiences gained with the first simulation model, physically more detailed models are going to be developed and implemented in a cycle simulation program.
Although, the invented model is simple, the trends of the results seem quite plausible and they help to already understand the physical mechanisms of the evaporator and the condenser in a better way.

2. THE HEAT EXCHANGER MODEL

2.1 Heat Exchanger Basics

The evaporators and condensers which are integrated in currently purchasable domestic refrigerators primarily consist of single tubes with a circular cross-section to guide the refrigerant. Generally, these tubes are bent into parallel serpentines, whose effective surface is extended by adding additional plates (e.g. roll-bond evaporators, hot-wall condensers) or various pairs of wire (e.g. wire-and-tube evaporators or wire-and-tube condensers). Typical materials for domestic refrigeration heat exchangers are aluminum, copper or steel.
The complex heat transfer in the heat exchangers of a domestic refrigerator can be described as follows: Inside the isolated refrigerator cabinet, heat is transferred to the evaporator walls by natural convection. Afterwards, the heat is transferred through the walls by conduction and to the refrigerant by forced convection. In the condenser, due to forced convection, heat flows from the refrigerant to the condenser walls. In the walls the heat is transported by conduction. On the condenser surfaces the ambient air is heated because of natural convection and radiative heat transfer.
All physical effects inside the heat exchangers are complex 3-dimensional processes. Besides being a combination of convective, conductive and radiative heat transfer, the working process is additionally complicated by the phase change of the refrigerant. In reality this is not a pure refrigerant, but actually a refrigerant/compressor oil mixture, whose composition is unknown and hardly detectable.
To sum up, the physics of a heat exchanger is extremely complex for simulation models, simplifying assumptions are necessary.

2.2 Simplifying Assumptions

The following simplifying assumptions are implemented in the condenser model:

- The refrigerant flow was assumed to be 1-dimensional.
- The two phase flow was assumed to be homogenous.
- The refrigerant channels were considered as straight, horizontal tubes with a constant cross-section and without any wires.
- The axial heat conduction in the tubes was neglected.
- The axial heat conduction in the refrigerant was neglected.
- The temperature of the ambient air was assumed to be constant.
- The pressure drop within the pipes was neglected.
- The forced convection heat transfer coefficients were computed by a polynomial function, just depending on the vapor quality.
- The natural convection heat transfer coefficients were assumed to be constant.
- The terms for kinetic and potential energy in the energy conservation equation were neglected.
- The liquefied refrigerant and the vaporized refrigerant were in thermal equilibrium.

The following additional assumption was made for the evaporator model:

- The presence of moisture was neglected.

2.3 Properties of R600a

The working media, which was used in this study, is the refrigerant R600a (isobutane). The properties for the refrigerant were implemented as a set of functions based on the formulations for the thermodynamic properties by Bücker and Wagner (2006).

2.4 Mathematical model

For the mathematical description of the physical processes, which occur in the heat exchanger, a distributed approach was chosen. The heat exchangers were divided in 30 equally sized finite volumes. In each of these cells, the first law of thermodynamics (1), which governs the conservation of energy, is solved for the refrigerant and the heat exchanger walls. With the help of the equation of continuity (2), the mass distribution and the mass fluxes have been calculated.

$$\dot{Q} + \sum_{j=1}^{k} \dot{m}_j \cdot h_j = \frac{dU}{dt} \tag{1}$$

$$\dot{m}_{in} - \dot{m}_{out} = \frac{dm}{dt} \tag{2}$$

With the assumptions mentioned above and by discretizing the first law of thermodynamics (1), an explicit formulation for the local enthalpy can be derived (3). A first order upwind scheme was used to estimate the enthalpy at the inlet of the cells.

$$\frac{\left(\dot{m}_{in} \cdot h_{in} + \dot{Q}_i\right) \cdot \Delta t + U_i^n + p_i^{n+1} \cdot V_{cell}}{\dot{m}_{in} \cdot \Delta t + m_i^n} = h_i^{n+1} \tag{3}$$

By knowing the specific enthalpy and the pressure, the corresponding density can be evaluated by the usage of functions concerning the thermo physical properties (4).

$$\rho_i^{n+1} = \rho(p_i^{n+1}, h_i^{n+1}) \tag{4}$$

With the size of the finite volume cell and the density, the mass in the cell can be computed (5).

$$m_i^{n+1} = V_{cell} \cdot \rho_i^{n+1} \tag{5}$$

The mass leaving each cell is evaluated by applying the equation of continuity (6).

$$\dot{m}_{out} = \dot{m}_{in} + \frac{m_i^n - m_i^{n+1}}{\Delta t} \tag{6}$$

2.5 Computation Algorithm (see Figure 1)

The pressure in the heat exchanger is calculated iteratively, starting with a first guess. Afterwards, the whole heat exchanger is calculated and the mass flux leaving the heat exchanger is compared with the boundary condition. If the mass flux at the heat exchanger outlet is higher than the boundary condition, the pressure is

increased for the next calculation. If the mass flux is lower than the boundary condition, the pressure is decreased.

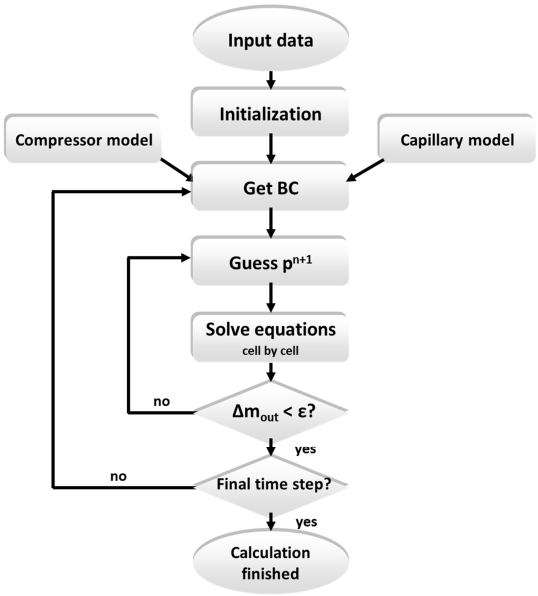

Figure 1: Information flow chart of the computation algorithm

3. CONDENSER RESULTS

3.1 Initial and boundary conditions

In order to assess the potentialities of the developed model, it has been tested under varying conditions.
Below, some results of a condenser simulation are going to be shown. These results are based on the following assumptions:
Since the other refrigerator components are not modeled, fictitious initial and boundary conditions are required
As initial condition thermodynamic equilibrium between the refrigerant and the condenser walls was assumed.
The mass flux, which enters the condenser, is assumed to be constant over the working period. The inlet enthalpy is computed by considering the current condenser pressure.
At the condenser outlet the mass flux follows a polynomial function and after a certain time it equalizes the inlet mass flux.

3.1 Simulation results

Figure 2 shows the mass fluxes at the inlet and the outlet of the condenser, which are given as boundary conditions. The total refrigerant mass within the condenser changes in dependence of the difference between the mass, which enters and leaves the condenser.

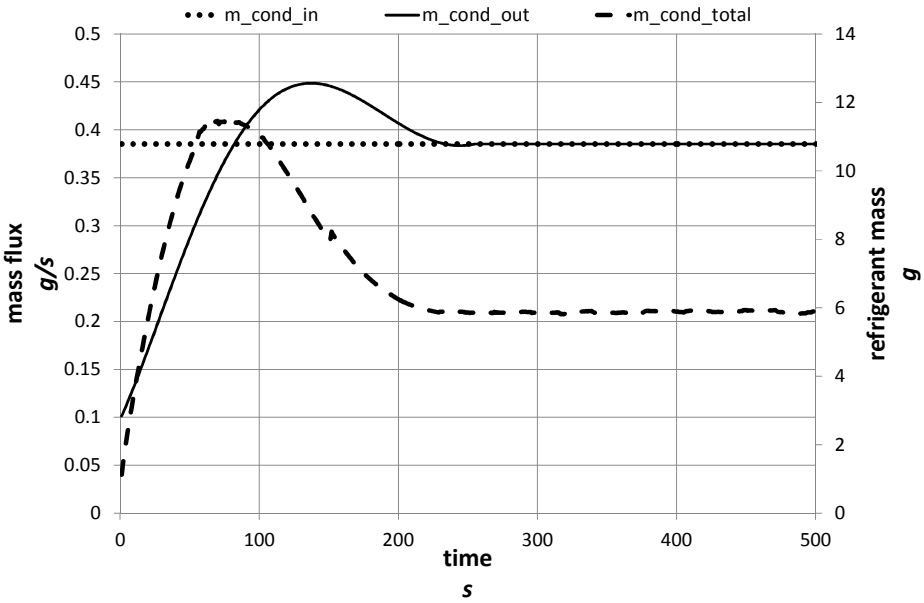

Figure 2: Mass fluxes entering and leaving the condenser and total refrigerant mass within the condenser

The resulting condenser pressure is depicted in Figure 3. In the first few seconds, the pressure increases quickly up to 4.0 bars. Afterwards, the pressure increases slowly and the steady-state pressure is reached after 350 seconds.

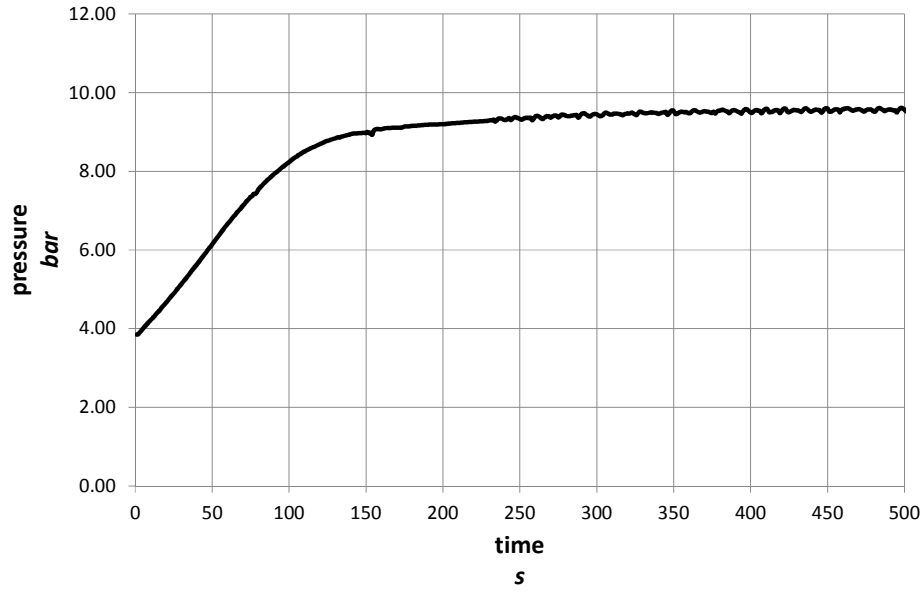

Figure 3: Condenser pressure trend

The refrigerant temperatures at three different condenser points (monitoring points in cell 1, cell 15 and cell 30) are depicted in Figure 4. The refrigerant is superheated in cell 1 and sub-cooled in cell 30. In cell 15 the refrigerant temperature is equal to the saturation temperature and thus it contains liquefied and vaporized refrigerant.

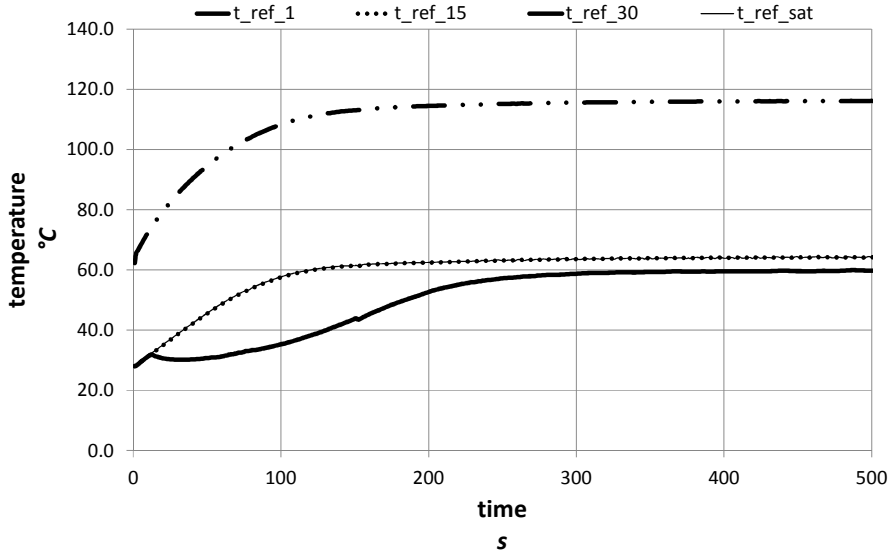

Figure 4: Temperatures of the refrigerant at three different cells

In Figure 5, the temperature of the condenser walls and the temperature of the refrigerant after 500 seconds are compared. When the refrigerant is superheated, there is a higher temperature difference between the refrigerant and the walls because of a low heat transfer coefficient. In the two-phase region, the heat transfer coefficient increases significantly and the temperature difference decreases. At the end of the condenser more and more refrigerant is liquefied. Thus the heat transfer coefficient decreases and the temperature difference increases again.

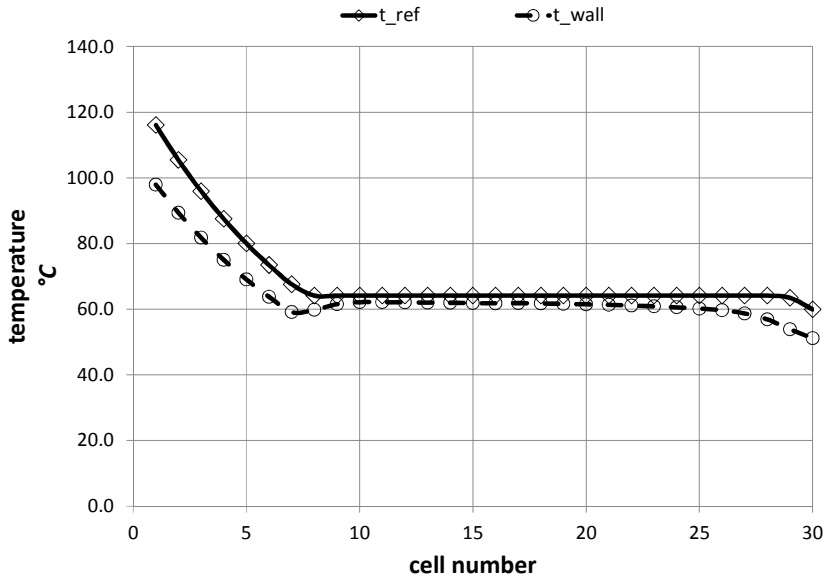

Figure 5: Temperature comparison: Condenser walls vs. refrigerant

4. CONCLUSION

In this work a simple but extendable computation algorithm for transient heat exchanger investigations was presented. The model is consciously kept quite simple and thus the results are inaccurate but some physical trends can already be reproduced in a plausible way.
In future, a more detailed model is going to be developed which is based on the experiences of the presented model. Furthermore, experiments are going to be carried out to validate the developed model. Finally, the validated model is going to be implemented in a transient refrigeration cycle simulation program.

NOMENCLATURE

$\dot{m}$	mass flow rate	(kg/s)	**Subscripts**	
h	enthalpy	(J/kg)	in	cell inlet
$\dot{Q}$	heat transfer rate	(W)	out	cell outlet
$\frac{dU}{dt}$	change of internal energy with respect to time	(J/s)	$n+1$	new time step
$\frac{dm}{dt}$	change of mass with respect to time	(kg/s)	n	former time step
Δt	time step	(s)	i	cell index
U	internal energy	(J)		
V_{cell}	cell volume	(m^3)		
p	pressure	(N/m^2)		
m	mass in cell	(kg)		
ρ	density	(kg/m^3)		

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