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CYCLING LOSSES IN DOMESTIC APPLIANCES; AN EXPERIMENTAL AND THEORETICAL ANALYSIS

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Nomenclature

A	area	(m²)	α	local void fraction	(-)
C	thermal capacity	(W/K)	$\bar{\alpha}$	mean void fraction	(-)
c_p	specific heat	(J/(kgK))	η	length	(m)
G	mass flux	(kg/(sm²))	ρ	density	(kg/m³)
h	enthalpy	(J/kg)	τ	time constant	(s)
k	heat transfer coeff.	(W/(m² K))			
M	mass	(kg)	subscripts		
m	mass flow	(kg/s)	c	condensation	
N_c	cycling frequency	(1/h)	c	cross-section	
p	pressure	(N/m²)	comp	compressor	
P	perimeter	(m)	e	evaporation	
R_p	running time perc.	(%)	i	inside	
T	temperature	(K)	l	liquid	
t	time	(s)	o	outside	
V	volume	(m³)	out	outlet	
W	electrical power	(W)	v	vapour	
x	quality	(-)	w	wall	
z	coordinate	(m)	∞	surroundings	

1. INTRODUCTION

The on/off mode is the way of control usually applied in domestic refrigeration equipment. This way of operation introduces cycling losses, the magnitude of which depends on a large number of parameters such as the on/off cycle length, the dimensions and heat capacities of the heat exchangers etc. These cycling losses can be split in two parts:

- During the on-periods the thermal load of the heat exchangers is higher than in a continuously controlled system (assuming an adapted, lower, refrigeration capacity). This results in a lower thermodynamic efficiency.
- "Start/stop"-losses can be defined since during the off-period refrigerant vapour flows through the capillary tube into the evaporator where it condenses.

The aim of the study described in this paper is to analyze these cycling losses in detail, both numerically and experimentally:

- The cycling losses can be determined in a numerical way using a dynamic simulation program for cooling circuits (DYSIRE) developed at Philips Research Laboratories. A summary of this model has already been presented at the IIR/Purdue conference in 1988 /1/. A comparison with other models is given there as well. Some basic principles will again be described in this paper (section 2).
- The experimental verification is performed on a freezer cooled by a forced convection evaporator. By measuring the pressures, energy flows and a large number of temperatures the system characteristics are determined. The experimental set-up is explained in section 3.

In section 4 a detailed comparison between a measured and a simulated on/off cycle is given. This is done for two different operating strategies: the first with the condenser closed during the off-period (using a valve between condenser and capillary tube; the so called "micloss system") and secondly, the system with the condenser open during the off-period (in this study referred to as normal system).

Thereafter the cycling losses are investigated in section 5. The disadvantage of a cycling system compared to a continuously running system and the effects of closing the condenser during the off-period are discussed. Finally some conclusions are drawn in section 6.

2. DESCRIPTION OF THE MODEL

A summary of the theoretical model is given here (restricted to the components of the refrigeration system which are of interest). The refrigerating system is subdivided into four components. Each of these components are discussed below.

2.1 Heat exchangers

In the model most attention is paid to the heat exchangers, both evaporator and condenser. In general the conservation equations for mass, momentum and energy describe the transient response of the fluid inside the heat exchangers. Assuming one dimensional flow, no pressure loss, neither gravity effects nor axial conduction (high Peclet number) and neglecting the kinetic energy of the fluid and viscous dissipation, one can derive the following conservation equation for mass and energy for a heat exchanger element with length Δz :

$$\frac{\partial M}{\partial t} = - \frac{\partial \dot{m}}{\partial z} \Delta z \tag{1}$$

$$\frac{\partial Mh}{\partial t} - V \frac{dp}{dt} = - \frac{\partial \dot{m}h}{\partial z} \Delta z + k_i A_i (T_w - T) \tag{2}$$

In principle, it is necessary to solve also the momentum equation. For reasons explained in /1,2,3/ this equation can be omitted here. To complete the system of equations it is necessary to add the following relationships:

$$M = \rho V \ , \ \rho = \Phi_1 \ (p \, , h) \ , \ T = \Phi_2 \ (p \, , h) \tag{3,4,5}$$

Here functions Φ_1 and Φ_2 are representations of the same refrigerant equation of state. Equations (1) to (5) form a set of non linear partial differential equations (PDE) which are solved iteratively. The response of the heat exchanger wall can be calculated using the energy equation:

$$C_w \frac{\partial T_w}{\partial t} = k_o A_o (T_\infty - T_w) - k_i A_i (T_w - T) \tag{6}$$

The above set of equations gives a good description of the single phase parts in the heat exchangers. However, in the two-phase region the thermodynamic properties of the fluid can only be correlated if certain assumptions on the flow field are made. The average density and enthalpy of the fluid can be calculated if a relation is given between the void fraction α and the quality x (defined as the ratio of vapour to total massflow). For homogeneous flow this relation can be derived from conservation principles. However, in practice a non-homogeneous flow in the heat exchangers is observed and a so called Void Fraction Model (VFM) has to be used.

Equal sets of equations can now be used for all three parts in the heat exchangers: the subcooled liquid, the two-phase and the superheated part. The heat exchangers are divided into a number of elements and a distributed (multiple node) model is thus created.

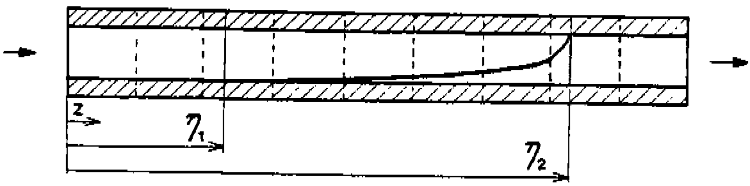

Figure 1. Schematic lay-out of the condenser heat exchanger

In Figure 1 the "numerical" lay-out of the heat exchanger is given in a schematic way. For reasons explained in /1/ the distributed model is not applied for the two phase area in the form presented. For this two phase area an integral approach is applied which takes into account the movement of the boundaries of the two-phase area /4,5/. For the complete two-phase region one continuity and one energy equation are derived assuming thermal equilibrium between the vapour and the liquid /1/. These conservation equations are obtained by an integration over the heat exchanger length. In this way ordinary differential equations (ODE's) are found. This integration can only be performed if the local liquid/void distribution is known. This distribution is given by the Void Fraction Model in general form:

$$\alpha = \alpha\,(x\,,\,T\,,\,m\,,\,...) \tag{7}$$

To find the average liquid/void distribution in the two phase area an integration over the local void fraction has to be performed which yields the mean void fraction:

$$\bar{\alpha} = \frac{1}{x_{\eta 2} - x_{\eta 1}} \int_{x = x_{\eta 1}}^{x_{\eta 2}} \alpha\,(x\,,\,...)\,dx \tag{8}$$

However, the latter integration is performed over a quality range in stead of over the length of the heat exchanger. Therefore the integral has to be transformed to an integral over the length of the heat exchanger which is possible by assuming a linear relationship between quality of the flow and position. In this way equation (8) is transformed to an integral over the length which on its turn is used to derive the continuity and energy equation for the complete two phase area.

The VFM published by Premoli /6/ is used in this investigation. Its selection is based on separate investigations /7,8/ on steady state functioning of appliances. It must be emphasized that especially in small refrigeration systems the prediction of the charge distribution is important since a capillary tube is applied and no accumulators are used.

The continuity and energy equation for the two phase area, together with the VFM and thermodynamic relations for the saturated conditions of the fluid, form a complete set of equations. This set of equations is solved iteratively for the entire two-phase area.
The complete set of equations for the heat exchangers has now been derived. The necessary boundary conditions for the condenser are the fluid massflow at inlet and outlet and the inlet temperature. For the evaporator the boundary conditions to be supplied are the inlet quality and the massflow at inlet and outlet.

The heat exchanger model as described above functions properly with the constraint that a reasonable flow of refrigerant exists. However, the refrigerant flow becomes too small if the compressor is switched off. Therefore different models need to be applied for the heat exchangers in case the compressor is not running:

- For the condenser no clear distinction between the different phases can be observed in the compressor off-phase. Therefore a one-node model ("stirred tank") is used with a homogeneous liquid/void distribution.
- For the evaporator model different boundary conditions exist during the compressor off-period. The two boundary conditions, massflow at inlet and outlet, remain. Normally the inlet quality is the third boundary condition. However, this value has no meaning anymore when the inlet massflow drops to zero. To be able to determine the two-phase region in the evaporator another boundary condition has to be given: in the model the position of the liquid dry-out point (end of the two phase region) is kept at the same place during the compressor off situation. Of course this is a rather artificial constraint. In practice the liquid redistributes through the evaporator mainly by gravity effects during the compressor off-period. This effect is thus not taken into account.

For the forced convection evaporator the thermal response of the wall is calculated using equation (6) as a basis. However, the term for the outside heat exchange is different. Heat is transported to the forced air flow passing the evaporator. For the air side the following balance is yielded:

$$\dot{m}_{air} c_p \Delta T_{air} = k_o A_o (T_w - T_{air}) \tag{9}$$

This equation is only valid for a constant air flow. Both variations in the air flow and accumulation effects of the air are neglected (low heat capacity, residence time of the air inside the heat exchanger very short) and therefore the simple stationary energy equation (9) can be used.

2.2 Compressor and compressor shell

The compressor outlet quantities and the electrical input can in general be defined with:

$$m = \dot{m}\,(p_c\,, p_e)\,, \qquad T_{comp,\,out} = T\,(p_c\,, p_e\,, T_{suction}), \qquad W_{comp} = W\,(p_c\,, p_e) \qquad (10, 11, 12)$$

The algebraic form of the equations expresses that these quantities are defined in quasi-stationary form. The equations themselves form approximations to experimentally obtained data. In the simulation the compressor shell is included.

2.3 Capillary

The massflow through the capillary tube can generally be expressed by:

$$m = \dot{m}\,(p_c\,, p_e\,, h_{c_{out}}) \qquad (13)$$

where the enthalpy denotes the condenser outlet enthalpy. In the model polynomial functions are used derived from calculations with a separate model /9/.

2.4 Solution method

The PDE's and the ODE's for the heat exchangers are coupled to the equations for the compressor, capillary and the compressor shell. In this way the necessary boundary conditions for the heat exchangers are obtained. All equations are discretized in a fully implicit form following an upwind scheme in order to ensure a stable calculation. The advantage is that rather large time steps can be used.

The disadvantage of the implicit scheme is that at every time step an iteration is necessary. The convergence can only be guaranteed for a limited time step size and depends also on the form of the non-linearities incorporated (equations of state, VFM, etc.). More details on the iteration process are given in /1/.

A timestep is completed after all iterations have ended succesfully. Then a next step is made using estimates from the previous step. Cycling is simply simulated by switching the compressor on and off during certain time periods. At the switch point special provisions are made in the model for switching from one algorithm for the heat exchangers to the other.

3. EXPERIMENTAL SET-UP

Figure 2 presents a schematic drawing of the experimental set-up. The cooling circuit under consideration contains two compressors, a condenser (natural convection, tube and wire construction), a capillary tube (7 l N_2/min) and an evaporator (forced convection, frostless type, refrigerant flow "bottom-up").

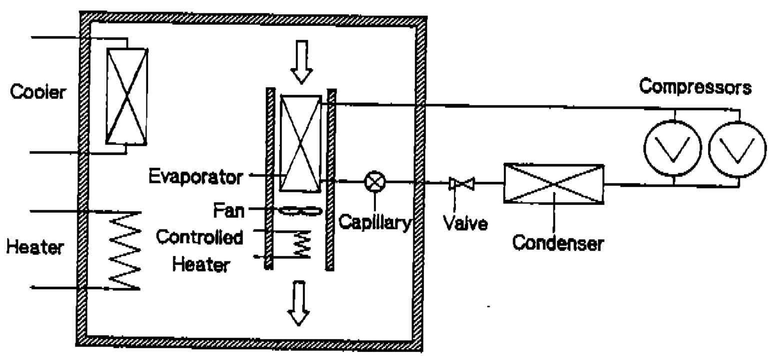

Figure 2. Schematic lay-out of the experimental set-up

The evaporator is mounted inside a box which is cooled by an external cooler which is continuously in operation. The temperature inside the box can be adjusted by means of an electrical heater which generates a constant heat load. If the cooling system being measured is switched on, the temperature in the box will decrease. However, this decrease is prevented by means of the controlled heater mounted directly after the fan. The cooling load of the evaporator has to be exactly balanced by the controlled heater in order to maintain the same constant temperature in the box. The electrical input of this controlled heater is measured and so the time dependent cooling capacity of the evaporator is obtained.

Furthermore the cooling system contains two identical compressors. This is done for the comparison of the cycling mode (with two compressors in parallel operation) with the continuous running mode (one compressor). In this way the comparison is not influenced by differences in the efficiency characteristics of the compressors.

An electric valve is placed between condenser and capillary tube. This valve can be closed during the compressor off-period (thus obtaining the micloss system).
A large number of thermocouples is mounted in the cooling system and pressure transducers are mounted in the suction and discharge line of the compressor.

4. EVALUATION OF ONE ON/OFF CYCLE

In Figure 3 the results of measurements and calculations of a typical cycling condition are shown (R_p = 50 % and $N_c = 8$). The figure shows the correlation for both the normal and micloss operation mode. As to the calculations shown some additional remarks must be made.

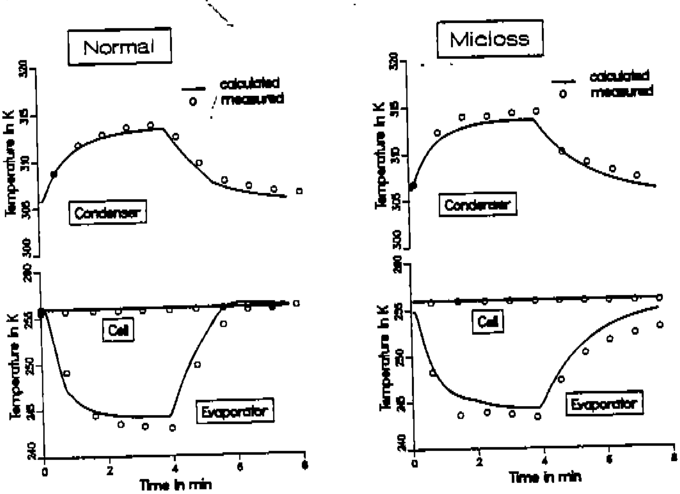

Figure 3. Temperatures during on/off cycling in normal and micloss operating mode

First, the outside heat exchange coefficients of the evaporator and condenser are determined from separate experiments and are fitted to the stationary running condition of this experimental set-up (i.e. two compressors running continuously)

Secondly, the void fraction model as given by Premoli /6/ does not correctly predict the charge used. For the stationary running condition the charge applied in the simulation has to be set to 75 g of refrigerant where the actual charge used in the measurement is 130 g. Only in this way the same area of superheating in the evaporator (app. 50 % for the stationary condition) can be calculated. An earlier study on the same apparatus /10/ yielded evidence that this discrepancy is mainly caused by some effects in the evaporator. The evaporator is of a forced convection type (fin-and-tube): the air flow is vertical. The refrigerant flow is mainly horizontal but is vertical

in the bends where the refrigerant flows from one layer of tubes to the next layer. It is important whether the refrigerant inlet is at the top or at the bottom of the evaporator.
In the study mentioned earlier /10/ the refrigerant inlet was at the top (thus obtaining a "top-down" flow). Measurements were carried out with a charge of 120 g obtaining a superheating area of approximately 20 %. To simulate the effects induced by the charge used in the measurements a multiplication factor of approximately 1.5 had to be applied to the liquid hold-up predicted by the Premoli relation.

However, in the present study the refrigerant flows "bottom-up". It proves to be necessary to use a much higher multiplication factor here (a value of 4.0) in order to obtain the same superheated region in the evaporator. Therefore the Premoli relation is not applicable for this type of flow and for such large diameter of tubes resulting in low mass fluxes. This is in contrast to earlier studies presented by the authors /1,7,8,11/ where in general the Premoli relation predicted the charge reasonably well. Applying the factor of 4 to the liquid hold-up the results presented in Figure 3 were obtained.

Additional investigations need to be performed to derive a correlation between the evaporator geometry (pipe diameter, number of bends etc.) and the liquid hold-up. Such a correlation can than be used in future simulations for a correct charge prediction without additional measurements.

5. CYCLING LOSSES

The cycling losses are defined as the difference between the energy consumption of a system with a continuously running (capacity controlled) compressor and a system with a cycling compressor both having the same operating temperatures and the same cooling load. In general these losses can be ascribed to three different phenomena:

Thermodynamic During the on-period the thermal load of the heat exchangers of the cycling system is higher than in the capacity controlled mode. This leads to a lower evaporation temperature and a higher condensation temperature and therefore to a lower thermodynamic efficiency.

Start/stop In domestic appliances using a capillary tube start/stop losses can be observed. Normally, the capillary remains open during the off-period and fluid flows from the condenser to the evaporator. The condenser pressure drops below the saturation pressure at ambient temperature and refrigerant collected in the condenser starts to evaporate. Furthermore, vapour flowing through the capillary tube condenses in the evaporator. Both effects have a negative influence on the system efficiency.

Compressor The efficiency of a compressor is a function of its capacity. This holds especially for small hermetic compressors. Therefore, if a capacity controlled (small) compressor is compared with a larger cycling compressor the difference in energy consumption will be partly due to differences in the efficiency characteristics of the compressors. In this study two identical compressors are applied to avoid a compressor efficiency influence. Thus only the thermodynamic and start/stop losses are investigated.

The following measurements are performed:

- One compressor in continuous running mode. The cooling capacity and the input power are measured and the ratio is defined as the coefficient of performance (COP).
- Two compressors with various cycling frequencies (N_c) and running time percentages (R_p). The results from these measurements are interpolated to obtain the COP as a function of N_c at the same cooling capacity as obtained by the first measurement (one compressor running continuously).
- Two compressors in continuous running mode.

The results of the tests above are presented in Figure 4. The results of the continuous running tests are given at $N_c = 0$. For the micloss operation the efficiency increases with increasing N_c. The thermodynamic losses decrease with shorter on/off periods due to the decreasing fluctuations in the heat exchanger temperatures. No start/stop losses occur because the condenser is closed during the off-period. On the contrary, for the normal operation start/stop losses do contribute to the cycling losses. The evaporation effect in the condenser and the condensation in the evaporator (during the off-period) are directly related to the number of off-periods per hour. In the measurements a clear reduction of the COP can be observed with increasing N_c. So the positive effect of lower thermodynamic losses is more than counterbalanced by the negative effect of the start/stop losses.

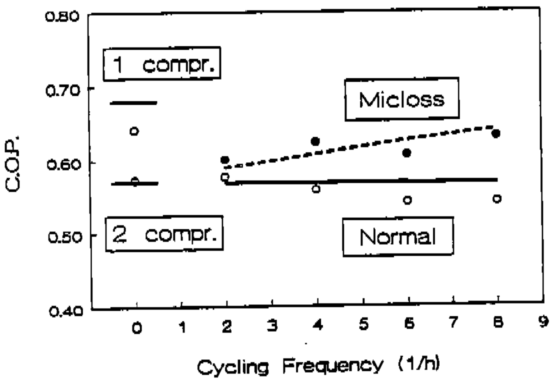

Figure 4. COP as a function of the cycling frequency (the measurements are presented by the circles, the calculations by the lines)

In the same figure the lines represent the results of the simulation runs. The agreement for the micloss operation is reasonable whereas the agreement for the normal operation is not satisfying. In the calculations the COP is almost independent of N_c. From a detailed analysis it follows that the filling degree of the evaporator differs between simulations and experiments. Every calculation yields the same two phase region at the end of the off-period (in this case approximately 50 % of the evaporator surface). However, in the measurements a clear influence can be observed. For the measurement with $R_p = 50$ % and $N_c = 2$ the same two phase region (50 %) is obtained. However, for $N_c = 8$ the two phase region at the end of the on-period is reduced to approximately 35 %. This results in a worse functioning of the evaporator and consequently a lower efficiency.
The efficiency calculated for the one compressor running continuously differs from the experimental value. Also here a discrepancy is found in the superheated area of the evaporator. The filling degree is worse in the experiment, leading to a lower efficiency.

In the measurement the liquid in the evaporator is redistributed by gravity effects during the off-period. After the start of the compressor the liquid is redistributed again resulting ultimately in a situation obtained with continuously running compressors. It is observed that this process takes a considerable amount of time (appr. 400 s). For high cycling frequencies the on-period is too short to get a complete redistribution. This leads to a lower filling degree for higher cycling frequencies and thus to a lower efficiency.

In the simulation the liquid is distributed instantaneously as prescribed by the Void Fraction Model given by Premoli /6/. In Figure 5 the liquid fraction given by Premoli (but multiplied with the factor 4) is given as a function of the quality and the massflux (G). For higher mass fluxes the liquid fraction is not much dependent on the mass flux. At very low mass fluxes which occur during the off-period the liquid fraction increases assymptotically (in fact no solution is found for $G = 0$). This is due to the mathematical formulation of the Premoli relation, which was never designed to be used in this area, of course. However, in the simulation this assymptotical behaviour can be used very well. At zero mass flux the liquid can accumulate up to a maximum level of 100 %. In this way the same VFM has been used in the on- and the off-period.

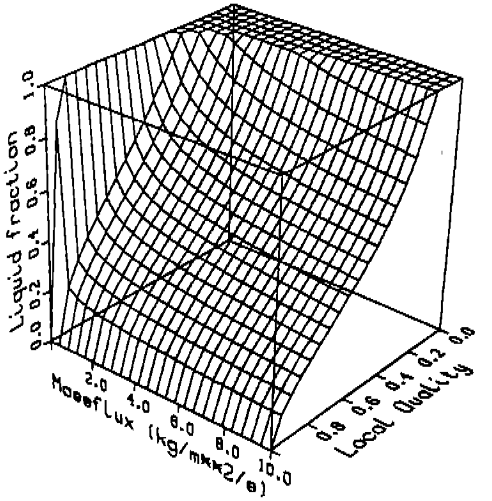

Figure 5. Liquid fraction given by Premoli (multiplied by 4.0 and cut-off at the value 1.0) as used in the simulation.

However, after a start of the compressor the mass flux increases very rapidly and the high liquid fraction is drastically reduced, back to a "normal" level. This process only takes some seconds in the simulation. Therefore the same filling degree of the evaporator is obtained independent of the cycling frequency which is in contrast with the measurements.

To obtain a better agreement between the measurements and the calculations an artificial time delay is built in the simulation by not using the liquid fraction directly derived from the actual mass flux (G_{real}) but from a "time delayed" mass flux (G'). To obtain this time delayed mass flux at time t the following formulation is used here:

$$G'_t = G_{real} - (G_{real} - G'_{t-\Delta t})\ e^{\left(\frac{-\Delta t}{\tau}\right)} \tag{14}$$

In Figure 6 the effect of this modification is shown using a time constant $\tau = 600$ s. It can now be shown that the COP is a function of the cycling frequency for the normal system. The spread in evaporator filling degree between cycling frequencies of 2 and 8 is still lower than observed in practice. For the micloss operation only a small influence can be observed.

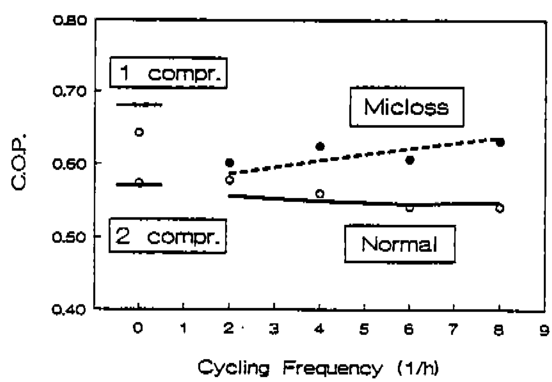

Figure 6. COP as a function of the cycling frequency; a time delay with time constant 600 s has been applied

5. CONCLUSIONS

In a dynamic model the implementation of a void fraction model (VFM) is necessary. From a previous study it is concluded that the Premoli VFM yields good results for the condenser in steady state conditions. However, for the evaporator a large multiplication factor to the liquid fraction has to be applied here (a factor 4). This is due to the low mass fluxes occuring and the geometry of the evaporator. The Premoli relations were derived for straight tubes where in this investigation an evaporator with a large number of bends is applied. It needs more investigation to obtain a correlation between the liquid hold-up and the evaporator geometry which can be used for future charge predictions.

The VFM applied in the evaporator has to be made time dependent in order to describe the transient behaviour in an acceptable way. Large amounts of liquid can be stored in a relatively small area of the evaporator. After start of the compressor the redistribution takes time and the stationary VFM cannot be applied directly. A more acceptable agreement is shown if a time delay is applied to the VFM.

In the tests performed an efficiency increase of 10 to 18 % for a continuously running system (one compressor) compared to an on/off system (normal operation) is possible (dependent on N_c). If a micloss system is applied the gain will be lower (from 2 to 6 %) and the micloss system will theoretically reach the maximum efficiency obtained with the one compressor system for a very high cycling frequency (the auxiliary energy for a shut-off valve has not been taken into account).

The above figures are only valid for the operation conditions selected. As a next step the simulations are a useful means in the determination whether a continuously running process (with a capacity controlled compressor) is feasible for a certain cooling circuit or not. The simulations will also be an appropriate tool in the determination of the optimal on/off operation.

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ABSTRACT

The on/off mode is the way of control usually applied in domestic refrigeration equipment. Due to this operation cycling losses will occur which depend on the on/off cycle length, the dimensions of the heat exchangers etc. It is the aim of this study to analyze these cycling losses in detail, both numerically and experimentally. Two operation modes for cycling are selected: the condenser closed during the off-period and the condenser open during the off-period. Both cycling operation modes are compared with a continuously running system.