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Abstract

In recent years many models for a simulation of the transients in a refrigeration cycle have been presented; most of them were aimed at an efficiency optimization and concern heat pump applications. These models assume either a three zone division in the heat exchangers and/or thermodynamic equilibrium in the two phase parts.

A different dynamic model has been designed for simulating a small refrigerating cycle as e.g. applied in domestic equipment. A multiple node model is used here for the description of the heat exchangers. This multiple node character is necessary for an accurate calculation of superheated and subcooled areas. In the model a mean void fraction approach is used in order to be able to describe the distribution of liquid and vapour in the two phase areas. This distribution is of influence in the calculation of the consequences of a certain fixed refrigerant charge applied.

Various assumptions for the basis for calculations using coefficients which take into account heat transfer and flow processes on the "outside" of the cycle; these coefficients are determined from steady state experiments or calculations. The emphasis in this paper is laid on a study of the influence of this approach on the overall accuracy.

Experiments and calculations for a start-up/ pull-down phase of an upright freezer with evaporator shelves are given. A sensitivity analysis is carried out in order to investigate the accuracy of the dynamic model itself and the importance of the accuracy of the steady state coefficients used.

Keywords: Dynamic Model, Small Refrigeration Systems, Mean Void Fraction, Start-up Phase.

RECHERCHE EXPERIMENTALE ET THEORIQUE D'UN MODELE DYNAMIQUE D'UN PETIT SYSTEME FRIGORIFIQUE

RESUME : Au cours des dernières années on a présenté de nombreux modèles de simulation en régime transitoire d'un cycle frigorifique, la plupart destinés à optimiser le rendement et concernant des applications de pompes à chaleur. Ces modèles consistent à diviser en trois zones les échangeurs de chaleur et/ou rechercher l'équilibre thermodynamique dans les parties en régime biphasique.

Un modèle dynamique différent a été conçu pour simuler un petit système frigorifique, tel un équipement ménager. On utilise un modèle à noeuds multiples pour décrire les échangeurs de chaleur. Cette caractéristique des noeuds multiples est nécessaire à un calcul précis des zones surchauffées et sous-refroidies. Dans le modèle on utilise

une étude de la fraction moyenne de vide pour décrire la répartition du liquide et de la vapeur dans les zones biphasiques. Cette répartition a une influence sur le calcul des conséquences pour une charge fixée de frigorigène.

Plusieurs hypothèses ont été formulées pour la base des calculs en utilisant des coefficients tenant compte des processus de transfert de chaleur et d'écoulement à "l'extérieur" du cycle ; ces coefficients sont déterminés à partir d'expériences ou calculs en régime permanent. Cette communication insiste sur l'étude de l'influence de ce mode d'étude sur la précision générale.

On présente les expériences et les calculs de la phase de démarrage/mise en régime d'une armoire-congélateur avec étagères évaporatrices. Une analyse de la sensibilité est effectuée pour apprécier la précision du modèle dynamique lui-même et l'importance de la précision des coefficients de régime permanent utilisés.

Mots-clés : modèle dynamique, petits systèmes frigorifiques, fraction de vide moyenne, phase de démarrage.

THEORETICAL AND EXPERIMENTAL INVESTIGATION OF A DYNAMIC MODEL FOR SMALL REFRIGERATING SYSTEMS

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Nomenclature

A	area	(m^2)	α	mean void fraction	(-)
C	thermal capacity	(W/K)	η	length	(m)
f	multiplication factor	(-)	ρ	density	(kg/m^3)
h	enthalpy	(J/kg)	τ	time constant	(s)
k	heat transfer coeff.	(W/m^2K)			
M	mass	(kg)	subscripts		
m	mass flow	(kg/s)	c	condensation	
p	pressure	(N/m^2)	c	cross-section	
P	perimeter	(m)	e	evaporation	
T	temperature	(K)	f	fluid	
t	time	(s)	l, liq	liquid	
V	volume	(m^3)	o,out	outlet	
x	quality	(-)	v	vapour	
z	coordinate	(-)	w	wall	
α	local void fraction	(-)	∞	surroundings	

1. INTRODUCTION

A mathematical analysis of refrigeration systems is often made by the use of steady-state models. For the investigation of a number of phenomena in small refrigeration systems this kind of modelling proved to be appropriate /1,2,3/. The literature mentioned concentrates on the effect of a fixed amount of refrigerant in capillary tube equipped systems. Void fraction models (VFM) proved to be an important part in order to obtain valid conclusions.
In normal refrigeration systems which are on/off controlled, steady state conditions are rarely obtained. Investigation of the influence of the dimensioning of components and of the control strategy require the application of models taking into account the transient behaviour.

The aim of this paper is to present a model which can be applied for the design and optimization of small refrigeration systems as e.g. used in domestic equipment. In practice it is important to study this kind of systems during several hours; therefore a fast model has to be used in the development of small refrigeration systems. The model also has to be flexible, since in development a large number of component modifications has to be studied.

In the transient situations, occurring during start-ups and on/off operation, the heat exchangers play a dominating role; their behaviour also depends on the total amount of refrigerant charge in the system. As far as the refrigerant flow and heat transfer is concerned, these components should therefore be modelled in a detailed way. Other components and effects may be implemented using steady state characteristics. E.g. this holds for the heat transfer on the air side of the evaporator or the condenser. The consequences of this kind of implementation are the main subject of this contribution.

It is useful to present a comparison of the models published up to now. Most of them are only suitable for use by the specialists who designed and tested them. It is therefore impossible to compare phenomena occurring in the practical use of these models. Only possiblities and shortcomings in modelling itself can be analyzed. Comparisons and recommendations for what a model should comprise if it has to be used for small systems are given in section 2. The next section describes the principles of a new model proposed. In section 4 experiments on a small appliance and the sensitivity of a number of approximations in the simulation model are presented. Conclusions can be found in section 5.

2. COMPARISON OF MODELS

A review of the literature shows that there is an increasing number of publications in the area of dynamic refrigeration modelling. Many of the studies refer to start-ups of heat pump installations. Most of the literature gives a short review on the models published so far and concludes that the development of new model should be preferred. A comparison of five recent models was e.g. given by Rajendran and Pate at Purdue, 1986 /4/. They, in their turn, also stated that a new model would be useful which should form a balance between comprehensiveness and simplicity.

From two reviewing articles on dynamic modelling some different points of view should be quoted. In an article on optimum conditions for dynamic functioning, Kruse and Upmeier /5/ conclude that, in spite of all the work, "there is a lack of applying the performance of computer simulation for a better understanding of the parameters affecting the energy losses during cycling operation." This is exactly one of the main fields of applications for the optimization of small refrigerating systems. In 1986 James and James published two technical memos /6, 7/ which present an analysis and a survey of dynamic mathematical models for refrigeration systems. For a more extended review the latter survey /7/ should be referred to where the authors conclude that, since 1973, the mathematical models derived "have been variations on a theme with only a few improvements being made."

Within the framework of this study a comparison of various topics found in the literature is considered to be necessary as well.
Most of the models consider the different parts in the heat exchangers as stirred tanks, to be seen as so called single nodes, and apply Euler methods for the solution of the equations. This holds for Dhar and Soedel's model published in 1979 /8/, and also for more recent models as e.g. published by James and James /9/. These models often concentrate on start-ups with large transients, so the calculational time steps have to be small, of the order of 0.005 s -as mentioned by Chi and Didion /10/- to 0.025 s /9/. Murphy and Goldschmidt consider both start-ups and shut-downs of a system /11, 12/. They are mainly interested in the condenser and evaporator pressure behaviour during the first seconds; compared to the influence of the heat exchangers, capillary tube characteristics are more important here. Simulation of long operation times would require time steps of the order of seconds. For this kind of time steps Euler methods are unstable.

Time steps in the order of seconds require the use of implicit solution schemes, which, in turn, are characterized by a considerable amount of computer CPU time per step. This solution method has been published recently by Sami /13/, who uses a lumped parameter, single node model. As an extra detail the author mentions the use of a new scheme, developed by his own, for the calculation of the void fraction for two-phase flow inside pipes.
The use of the momentum equation to calculate pressure loss is seldom considered; Chi and Didion did take this effect into account and therefore had to choose a very short time step /10/. Brasz and Koenig /14/ recommend neglecting the acceleration terms and calculating, if necessary, the pressure loss afterwards. They concentrate their studies on heat transfer and fluid flow in the heat exchangers, in which they use an implicit solution method with a step size of 0.1 s.
In a study of the flow in condensers and evaporators Wedekind and collaborators /15, 16/ use a moving boundary model for the two-phase part. Using this model they investigate the application of a mean void fraction, determined via an integration of a separate VFM.

Since 1984 the use of multiple node models has got more and more emphasis /12, 17, 18/. MacArthur /17/ uses this grid spacing and studies local effects in the heat exchangers. He mentions the use of equilibrium conditions in the heat exchanger but does not give a slip relationship in an explicit manner. In a subsequent contribution in 1987, MacArthur /19/ concludes that the lack of a VFM or slip model -which was not used in the aforementioned investigation- results in inaccurate refrigerant mass distributions. At the same time he presents a sophisticated, new model and mentions time steps of the order of 10 s obtained with a fully implicit method; this when applying the method to the heat exchangers in a separate way. Yet the use of his VFM requires further study and the use of fixed boundaries -equal to the grid spacing- probably yields difficulties; this will be the reason that he states a so called reduced-order moving boundary heat exchanger model being therefore under development /19/.

Extensive models for expansion valves and compressors are used in a number of the aforementioned dynamic models. It has not been investigated whether simple models for these components would yield equally good results. However, it can be assumed that the time dependency of the compression and expansion process is not important and that only heat transfer processes in these components should be taken into consideration.

3. DESCRIPTION OF THE MODEL

For the model described in the present paper the refrigerating system is subdivided into six components. Each of these components will be discussed separately below.

3.1 Heat exchangers

In the present 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 :

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

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

This somewhat unusual representation of the conservation equations (in M and h) is convenient for coupling with other type of equations which will be described later.

For calculating the transient response in the heat exchangers the momentum equation with its transient terms must also be solved. Simultaneously a solution for the pressure waves is obtained; these pressure waves are stated to be of minor importance for the large transients, which are mainly determined by heat transfer processes /7, 14/. Moreover, solving the momentum equation will lead to a large computer CPU time increase because of the small time scale effects of the acoustic waves. Due to the fact that the non-transient terms in the momentum equation, i.e. the pressure loss, are neglected in this investigation, the complete momentum equation is omitted. To complete the system of equations it is necessary to add the following relationships:

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

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 (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 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. It is emphasised by the authors that a distributed model is necessary for a correct simulation of small refrigeration systems. These systems are usually not provided with liquid accumulators and are charged with a certain fixed amount of refrigerant. Due to these facts and the use of a capillary tube, the heat exchangers are not always properly filled with evaporating or condensing refrigerant. Considerable parts of the heat exchangers contain subcooled or superheated refrigerant. The temperature of these parts is a function of the position in the heat exchanger. One way to account for these temperature profiles is the use of a distributed model. The subcooled and superheated parts have a severe influence on the transient behaviour of the heat exchanger as a whole.

The model described above leads to problems if the solution procedure selected by the authors for solving the complete system is applied. It is necessary to use especially this solution procedure as will be discussed below. The problems arise when the transition point between a single phase and the two-phase area passes from one element to another. At that stage it is necessary to switch from one thermodynamic property relation valid for the single phase to another relation valid for the two

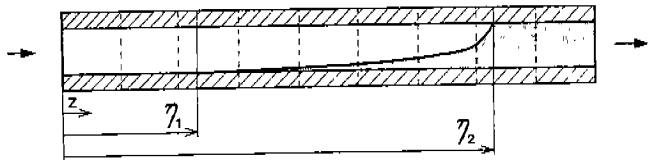

Fig. 1. Schematic lay out of the condenser heat exchanger with the two-phase boundary length coordinates

phase region. This yields convergence problems since the solution method uses an iterative approach. Therefore another description is used for the two-phase part which takes into account the movement of the boundaries of the two-phase area /15, 16/. In Fig. 1 the "numerical" lay-out of the heat exchanger is given in a schematic way.
For the two-phase part the following conservation equations for mass and energy can be derived assuming thermal equilibrium between the vapour and the liquid:

$$\frac{d}{dt}\{(\rho_l(1-\bar{\alpha})+\rho_v\bar{\alpha})\ A_c(\eta_2-\eta_1)\} = m_{\eta 1}-\dot{m}_{\eta 2}$$

$$-\{\rho_l(1-\alpha)+\rho_v\alpha\}_{\eta 1}A_c\frac{d\eta_1}{dt}+\{\rho_l(1-\alpha)+\rho_v\alpha\}_{\eta 2}A_c\frac{d\eta_2}{dt} \tag{7}$$

$$\frac{d}{dt}\{(\rho_l h_l(1-\bar{\alpha})+\rho_v h_v\bar{\alpha})\ A_c(\eta_2-\eta_1)\}-A_c(\eta_2-\eta_1)\frac{dp}{dt} =$$

$$\int_{z=\eta_1}^{\eta_2} k_i P(T_w-T)\,dz+\{\dot{m}[h_l(1-x)+h_v x]\}_{\eta 1}-\{\dot{m}[h_l(1-x)+h_v x]\}_{\eta 2}$$

$$-\{\rho_l h_l(1-\alpha)+\rho_v h_v\alpha\}_{\eta 1}A_c\frac{d\eta_1}{dt}+\{\rho_l h_l(1-\alpha)+\rho_v h_v\alpha\}_{\eta 2}A_c\frac{d\eta_2}{dt} \tag{8}$$

The response of the heat exchanger wall is again calculated by means of equation (6). In general the void fraction models are supplied in local void fraction form:

$$\alpha=\alpha(x,T,\dot{m},\ \ldots) \tag{9}$$

This local void fraction model is needed in case the boundaries of the two-phase areas coincide with the physical boundaries of the heat exchanger, i.e. in the case of not complete condensation or evaporation. Nevertheless, in all cases a mean void fraction is required which is defined as:

$$\bar{\alpha}=\frac{1}{x_{\eta 2}-x_{\eta 1}}\int_{x=x_{\eta 1}}^{x_{\eta 2}}\alpha(x,\ \ldots)\,dx \tag{10}$$

This integral over the quality range is transformed to an integral over the length of the heat exchanger by assuming a linear relationship between quality of the flow and position. In this way equation (10) is transformed to an integral over the length which equation is used to derive the equations (7) and (8) from the general conservation form.
The VFM published by Premoli /20/ is used in this investigation. Its selection is based on separate investigations /1,2/ 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 two ordinary differential equations (ODE's), given as equations (7) and (8), form a complete set of equations, together with the VFM and thermodynamic relations for the saturated conditions of the fluid; this set of equations is solved iteratively for the entire two-phase area.
The ODE's for the two-phase part must be coupled with the PDE's for the single phases. Very special attention has to be paid to the description of the single phase elements bounded by the two-phase region. Equations (1) to (5) can be used for these elements, but a term for the moving boundary has to be included.
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.

3.2 Compressor and compressor shell

The compressor outlet quantities can in general be defined with:

$$m = F_c\,(p_c\,,\,p_e)\,, \qquad T_{comp_{in}} = F_T\,(p_c\,,\,p_e\,,\,T_{suction}) \tag{11, 12}$$

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. The interaction between the refrigerant and the oil in the shell is accounted for by means of equilibrium- state relations defining the refrigerant dissolution in oil:

$$M_{f_{oil}} = F_{oil}\,(p_c\,,\,T_{oil}) \tag{13}$$

By calculating the heat transfer processes within the shell and from the shell to the ambient, the oil temperature is calculated. The response of the fluid inside the shell is calculated using the equations (1) to (5) with extra terms for the refrigerant fluid flow into or out of the oil and extra terms for the heat transfer processes.

3.3 Capillary

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

$$m = F_{cap}\,(p_c\,,\,p_e\,,\,h_{c_{out}}) \tag{14}$$

where the enthalpy denotes the condenser outlet enthalpy. The inclusion of advanced mathematical models in the dynamic simulation in order to obtain the massflow will increase the calculation time considerably. Therefore large numbers of calculations were performed with a separate model /21/ the results of which were approximated by polynomial functions. These functions could be derived due to the limited number of influencing parameters.

3.4 Cabinet

The appliance itself is modelled by dividing the walls into several elements. The elements in their turn are divided into a number of layers which form the basis for the discretisation of the Fourier heat conduction equation. The heat flow is considered to be one dimensional, i.e. only perpendicular to the walls of the appliance. At the wall surfaces appropriate heat exchange coefficients are introduced. The internal heat capacity of the appliance is modelled in the form of one node. The response of this node is calculated according to:

$$C_{cell}\frac{\partial T_{cell}}{\partial t} = \sum_i Q_{wall_i} - \sum_i Q_{evap_i} \tag{15}$$

were Q_{evap_i} is the heat absorbed by wall element i of the evaporator.

3.5 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.
Rather large time steps, of the order of 1 to 2 seconds, can be used but they depend greatly of course on the number of elements chosen for the heat exchangers and on the absolute values of the capacities in the system. The fully implicit form is also used for the compressor and capillary relations which implies that these relations are supplied in terms of the variables at the new time level.

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.). The iteration in the heat exchangers is performed by searching for the correct pressure for which the outlet massflow is equal to the massflow through the next component. For instance, for the condenser pressure, iterations are performed via a Newton process until the outlet massflow equals the capillary massflow within a specified tolerance. Due to this tolerance a mass inconsistency in the system originates and at every time step a very small inaccuracy in the total mass balance in the system is created. In case of large simulations, however, the error may increase to unacceptable levels. Therefore the total mass in the system is continuously adjusted by adding a very small mass as a correction of the mass balance in the compressor shell.

3.6 Use of the model

With the solution method discussed it proved possible to simulate 10 on/off periods (2 periods per hour) in approximately 15 minutes CPU time on an IBM3081 mainframe.
The model has been extended with data banks containing the specifications of a number of components and several input/output facilities in order to obtain a "user friendly" program as a whole. Selection of components can easily be made and new ones can be added. It is also possible to use more condensers or evaporators in series. In this way it is possible to calculate a broad range of appliance types, upright freezers, freezer/ refrigerator combinations etc. The aim of the dynamic model research is to obtain an appropriate and practical tool for the development department. That this aim has been achieved is demonstrated by the extensive use of the program at present.

4. COMPARISON WITH MEASUREMENTS

Temperature measurements were performed on a 290 l domestic, static upright freezer appliance. The appliance is equipped with seven evaporator shelves with a total surface area of 2.35 m^2 ; the condenser is of the louvre type and has a total surface area of 1.0 m^2
Two measurements were carried out as follows. In the equilibrium "off" situation -refrigerant vapour in all components at ambient temperature- the compressor is started and is operated until the equilibrium "on", steady state situation is reached after 8 to 10 hours; the total is defined as a "pull-down" measurement. The differences between the two measurements can be found in the compressor and capillary capacity and the refrigerant charge. In the second measurement ("B", Fig. 3) an extra thermal capacity (5.2 kg of copper material) is put inside the appliance.
Both measurements are simulated in the "standard" configuration and the results are also presented in Figs. 2 and 3. In the simulation physical data are used as e.g. for the air side heat exchange coefficients or the compressor mass flow which are obtained from separate, steady state measurements or from literature. It can be observed that the simulations and measurements correlate very well in the steady state situation. This in itself is not surprising since steady state physical parameters are applied here. The start-up phase, i.e. the first two hours of operation, represents the most severe condition for calculational predictions. It can be deduced from Figs. 2 and 3 that the "standard" simulation and measurement do indeed differ considerably during this transient part.
The aim of this investigation is to estimate the influence of the steady state character of the physical relations used in the transient mode. The eleven most important relations hold for:

1. compressor massflow
2. compressor outlet temperature
3. condenser internal heat exch. coefficient
4. condenser external heat exch. coefficient
5. condenser VFM
6. capillary massflow
7. evaporator internal heat exch. coefficient
8. evaporator external heat exch. coefficient
9. evaporator VFM
10. refrigerant-oil dissolution
11. the use of an "one-node" model to account for all the capacity inside the appliance

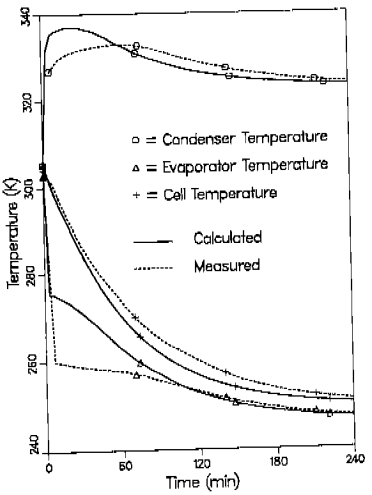

Fig. 2. Comparison between pull-down measurement and "standard" simulation (pull-down "A").

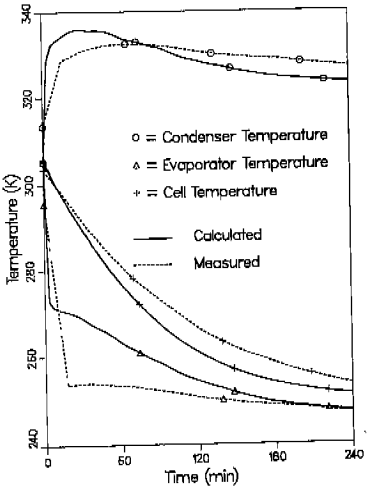

Fig. 3. Comparison between pull-down measurement and "standard" simulation (pull-down "B").

The effect of introducing variations (order of magnitude, time dependency) to the above relations has been studied for one of the tests ("A", Fig. 2). Comparison is made between a number of data which are considered representative:

- the evaporation temperature at 30 min;
- the evaporator wall temperature at 30 min;
- the cell "one-node" temperature at 90 min;
- the time at which the condenser temperature "peaks";
- the maximum condenser wall temperature (see point 3);
- the condenser subcooling degree at 30 min.

The evaporator or condenser wall temperature is defined as that wall temperature of the heat exchanger where evaporation or condensation takes place at the inside.

The steady state relations are varied following

$$\phi = \phi_{st} (1 + f \ \exp \ (-t / \tau)) \tag{16}$$

where ϕ_{st} represents one of the eleven parameters given above. Values for τ used range from zero to two hours, so that the steady state value ϕ_{st} is always used when simulating equilibrium "on" conditions.

The sensitivity analysis is performed by a separate variation of each of the parameters. The aim of the investigation is to study which variation yields a better agreement between measurement and calculation, i.e. a lowering of both the condenser and evaporator temperatures and a time shift in the condenser temperature peak (see Figs. 2 and 3). It should be underlined that there is a strong link between both temperatures; i.e. a lower evaporation temperature directly influences the condensation pressure or temperature, which is mainly caused by the capillary and compressor characteristics. Each effect of a parameter variation is described separately below and is shown in Table I.

1. **Compressor Mass Flow.** Only an unrealistic increase of the volumetric capacity yields a decrease in the evaporator temperature, however, at the same time to an increase in the condenser temperature.
2. **Compressor Outlet Temperature.** In practice a lower outlet temperature will appear compared to the steady state values, since it depends on warming-up of the thermal capacities present within the compressor shell. A substantial decrease of the outlet temperature results in a small decrease of the condenser and an equal value for the evaporator temperature.
3. **Condenser Inside Heat Exchange.** This effect has not been investigated since the outside -natural convection- heat exchange, with its much lower order of magnitude, is predominant.
4. **Condenser Outside Heat Exchange.** Only by an unrealistic increase of the convection coefficient the temperature peak can be somewhat decreased and delayed.
5. **Condenser VFM.** Large influences on temperatures are observed when the mean void fraction, derived from the VFM, is substantially decreased. This parameter proves to be of paramount influence and investigations are more extensively described below.
6. **Capillary Mass Flow.** Due to the coupling of condenser and evaporator temperature via the capillary characteristics, contrary effects are obtained, e.g. a decrease of the evaporator and an increase of the condenser temperature.

Number	Parameter ϕ	f (-)	τ (h)	....Evaporator... Fluid Temp 30 min (K)	Wall Temp 30 min (K)	Cell Aver. Temp 90 min (K)	.....Condenser..... Time T' at Peak (min)	Max. Temp at T' (K)	Subc. Temp 30 min (K)
	Measurement				259.0	265.0	60.	333.0	
	Standard			265.2	270.7	261.0	22.	337.0	9.5
1	m_{comp}	0.5	2.	260.6	266.5	257.4	20.	340.9	13.4
2	$T_{comp_0} - T_{ein}$	3.0	2.	264.2	269.8	260.6	20.	334.8	10.8
4	$k_{c,outside}$	1.0	2.	261.8	267.9	259.3	30.	328.1	6.0
5	$1 - \overline{\alpha}_c$	3.0	2.	256.4	267.9	266.8	55.	332.7	0.6
6	$m_{capillary}$	-0.2	∞	262.1	269.0	261.8	25.	336.9	15.8
7	$k_{c,inside}$	9.0	2.	266.5	267.2	260.2	25.	338.5	10.3
8	$k_{c,outside}$	-1.0	2.	259.6	262.5	265.4	40.	332.5	3.7
9	$1 - \overline{\alpha}_c$	1.0	2.	262.8	269.5	263.0	25.	335.1	7.0
10	$M_{f,col}$			260.1	269.1	264.3	40.	333.7	5.2
11	ΔT_{cell}		2.	264.2	269.0	261.2	25.	336.3	8.4

Table I. Effect of parameter variations on different characteristic data obtained in the simulation of two pull-down tests (compare Figs. 2 and 3).

7. **Evaporator Inside Heat Exchange.** Without influencing the condenser temperature the internal heat exchange mainly influences the evaporator wall temperature. Better agreement between measurement and calculation is obtained by increasing the heat exchange coefficient to a level of 1000 W/m^2K . The wall temperature is decreased in the order of 5K in this way, at the same time resulting in a 1K increase of the evaporation temperature. There is some theoretical evidence for this increase because the evaporation heat exchange will be of the boiling type during the start-up phase, rather than of the convective type as used in the steady state phase.
8. **Evaporator Outside Heat Exchange.** A significant decrease of both the evaporator and condenser wall temperatures is only found for an unrealistic decrease of the heat exchange coefficient
9. **Evaporator VFM.** Clear influences on both evaporator and condenser wall temperatures are observed when the void fraction in the VFM applied is lowered. However, the void fraction has to be substantially decreased, which, automatically, leads to a smaller length of the evaporator filled with evaporating liquid. Some remarks on the evaporating liquid front are made below.
10. **Refrigerant Oil Dissolution.** Effects are found when the release of the refrigerant out of the oil is made time dependent in a numerical way; this implies that the refrigerant is stored in a hypothetic accumulator from which it is released again in a controlled way. Only unrealistically large time delays yield noticeable influences on both heat exchanger temperatures.
11. **Evaporator Air Temperature Profile.** The introduction of a variation of the air temperature as a function of the evaporator length -so that the evaporator element behaviour is calculated using a varying outside heat exchange- yields a slight decrease in the evaporator and condenser wall temperatures. The indications about the air temperature profile used are derived from the two experiments described.

Summarizing, three parameters should be implemented in order to obtain better agreement between measurements and calculations. Although only yielding small influences, a time dependent compressor capacity and evaporator temperature profile should be included since these can be assumed realistic and are partly confirmed in experiments. The third and only parameter to be considered for obtaining better agreement is the condenser VFM.
Results of a modification of the void fraction by application of equation (16) are presented in Figs. 4 and 5. Best agreement is obtained by applying a multiplication factor of two and a time constant of one hour to the liquid fraction, defined as one minus the void fraction.

It might be assumed that the modification applied to the VFM would be necessary due to the fact that a non-correct steady state VFM is applied. However, satisfying agreement could already be

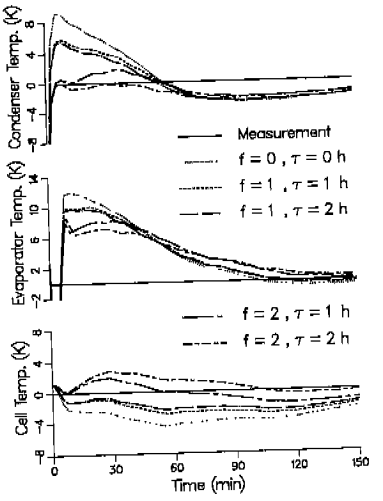

Fig. 4. Effect of a variation of the liquid fraction in the condenser on the time dependent evaporator, condenser and cell temperatures. Data are represented relative to the temperatures measured; it concerns pull-down test "A".

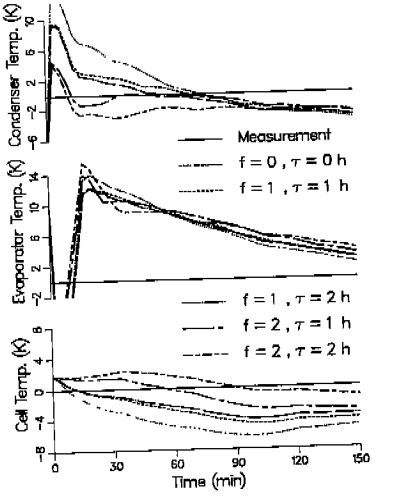

Fig. 5. Effect of a variation of the liquid fraction in the condenser comparable to Fig. 4; it concerns pull-down test "B".

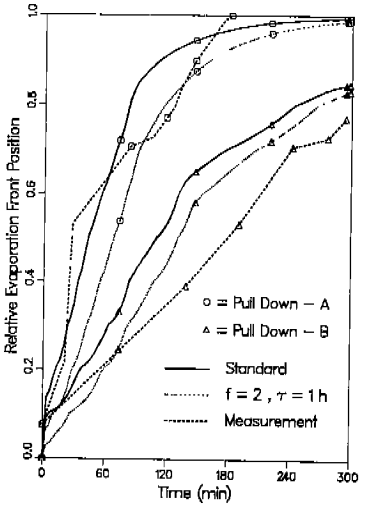

Fig. 6 Time dependent position of the liquid front in the evaporator for both pull-down tests given in Figs. 2 and 3 ("A" and "B"). Results are presented, which are obtained from both the "standard" and the modified simulation -time dependent VFM-; the measurement results are also given.

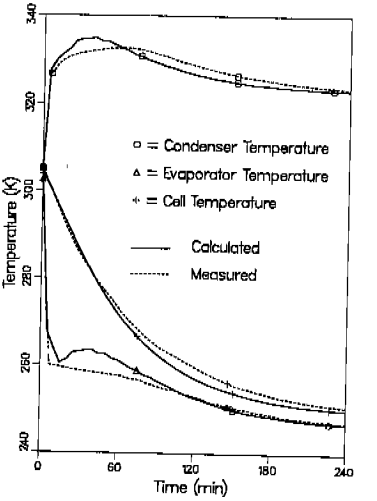

Fig. 7 Comparison of a pull-down measurement ("A") and the results of a simulation applying a time dependent VFM and increased evaporator internal heat exchange.

concluded for the Premoli VFM applied in steady state measurements /2, 3/. Yet time dependent characteristics will be influencing the refrigerating cycle. This can indeed be explained by the fact that a large part of the more or less subcooled liquid is still in the non-equilibrium phase and should be accounted to the two phase part, thus decreasing the mean void fraction assumed.
Confirmation of the above assumption for the decrease of the mean void fraction in the condenser VFM only -and not in the evaporator- can be found in the evaporating liquid front position in the evaporator. The application of the Premoli VFM model -without modification- in the evaporator yields good agreement with the measurement, which can be concluded from the curves represented in Fig. 6.

Additionally, better agreement between the evaporator wall temperatures measured and calculated can be obtained by a substantial increase of the evaporator internal heat exchange. For a choice of a multiplication factor of four (heat exchange coefficient in the order of 1000 W/m^2K) and a time constant of two hours the best agreement is obtained, which is shown in Fig. 7. When comparing the "standard" results (given in Figs. 2 and 3) it can be concluded that calculations, applying both a modified VFM and an increased evaporator heat exchange, correlate very well to the measurements in this way.

5. CONCLUSIONS

A multiple node fully implicit model for the description of a small refrigeration cycle is a must in case time dependent behaviour has to be described sufficiently accurately. This kind of model will also lead to reasonable calculation times.

Implementation of a void fraction model (VFM) is absolutely necessary. For steady state conditions the Premoli VFM yields good results. However, time dependence has to be added to the VFM in order to describe the transient condenser behaviour in an acceptable way. An increase of the mean liquid contents of the condenser by a factor of three in the start-up phase proves to be required. During the start-up phase a great increase of the evaporation heat transfer coefficient yields much better simulation results. Further investigation should be performed whether the increase of both the mean liquid fraction and the heat transfer coefficient should also be applied in the normal on/off functioning.
On the other hand, most steady state coefficients applied for a lot of phenomena occurring in small refrigerating systems, prove to be accurate enough for a good description of all transients.

6. REFERENCES

/1/ L.J.M. KUIJPERS and M.J.P. JANSSEN. 1986. *Proc. IIR Meeting at Purdue,* 1986 - 1, pp. 61-70.

/2/ L. KUIJPERS, M. JANSSEN, J. de WIT. 1987 *Proc. XVIIth Int. Congress Refrigeration,* Vienna, Vol. B, pp. 307-315 (see also pp. 555-563)

/3/ L.J.M. KUIJPERS, M.J.P. JANSSEN and P.J.M. VERBOVEN. 1988. "The Influence of the Refrigerant Charge on the Functioning of Small Refrigerating Appliances", Paper No. 3194 presented at the ASHRAE Meeting, Ottawa, June 1988, to be published in *ASHRAE Transactions 1988, Volume 94, Part 2*

/4/ H. KRUSE and B. UPMEIER. 1984. *Proc. IIR / B2 Meeting Dresden (GDR)* pp. 239-261

/5/ N. RAJENDRAN and M.B. PATE. 1986. *Proc. IIR Meeting at Purdue,* 1986 - 1, pp. 201-213

/6/ K.A. JAMES, R.W. JAMES. 1986. "Dynamic Analysis of a Heat Pump using Established Modelling Techniques", *Technical Memorandum No. 98,* Inst. Environm. Eng., Polytechnic South Bank, London (UK)

/7/ K.A. JAMES, R.W. JAMES and A. DUNN. 1986. "A Critical Survey of Dynamic Mathematical Models of Refrigeration Systems and Heat Pumps and Their Components", *Technical Memorandum No. 97,* Inst. Environm. Eng., Polytechnic South Bank, London (UK)

/8/ M. DHAR and W. SOEDEL. 1979. *Proc. XVth Int. Congress Refrigeration,* Venice, Vol. II, pp. 1035-1067

/9/ K.A. JAMES, R.W. JAMES and A. DUNN. 1987. *Proc. XVIIth Int. Congress Refrigeration,* Vienna, Vol. B, pp. 629-634

/10/ J. CHI and D. DIDION. 1982. *Int. J. Refrigeration,* Vol. 5, No. 4, pp. 176-184

/11/ W.E. MURPHY and V.W. GOLDSCHMIDT. 1985. *ASHRAE Transactions,* 91, part 2, pp. 427-444

/12/ W.E. MURPHY and V.W. GOLDSCHMIDT. 1986. *ASHRAE Transactions,* 92, part 1

/13/ S.M. SAMI, T.N. DUONG, Y. MERCADIER, N. GALANIS. 1987 *ASHRAE Transactions,* Vol. 93, part 2, pp. 471-490

/14/ J.J. BRASZ and K. KOENIG. 1983. "Numerical Methods for the Transient Behavior of Two Phase Flow Heat Transfer in Evaporators and Condensers", *Numerical Properties and Methodologies in Heat Transfer,* ed. T.M. Shih, Springer Verlag, Berlin, pp. 461-476

/15/ G.L. WEDEKIND et al. 1978. *Int. J. Multiphase Flow,* Vol.4, pp. 97-114.

/16/ B.T. BECK and G.L. WEDEKIND. 1981. *ASME J. Heat Transfer,* Vol. 103, pp. 81-85

/17/ J. WARD MACARTHUR. 1984. *Energy Convers. Mgmt.* Vol. 24, No. 1, pp. 49-66.

/18/ T.J. BECKEY. 1986. *Proc. IIR Meeting at Purdue,* 1986 - 1, pp. 175-183

/19/ J.W. MACARTHUR and E.W. GRALD. 1987. *ASHRAE Transactions,* Vol. 93, Part 2, pp. 1159-1178

/20/ A. PREMOLI, D. Di FRANCESCO and A. PRINA. 1971. *La Termotecnica,* Vol. 25, pp. 17-26

/21/ L.J.M. KUIJPERS and M.J.P. JANSSEN. 1983. *Proc. XVIth Int. Congress Refrigeration,* Paris, Vol. II, pp. 689-698