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Continuous Control of Refrigerating Appliances; Limits Due to Capillary Tube Behavior

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CONTINUOUS CONTROL OF REFRIGERATING APPLIANCES;
LIMITS DUE TO CAPILLARY TUBE BEHAVIOUR

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1. INTRODUCTION

The way of control normally selected for domestic refrigeration equipment has to be characterized by aspects of simplicity and reliability; these aspects form the most important reason for the on/off control applied. The use of a capillary tube together with a compressor running at full capacity, provides a stable and acceptable functioning over a wide ambient temperature range. "Acceptable" in this respect means that the evaporator is filled with a sufficient amount of refrigerant at a reasonable energy consumption of the compressor. One important aspect has to be mentioned as well. In contrast with large cooling systems where measures are taken to control the charge in various components, small domestic equipment functions with a fixed total amount of refrigerant charge in the cycle. This amount of charge should yield good cooling conditions for a wide range of ambient temperatures, which conditions result, for a large part, from the capillary tube characteristics.

The on/off control forms the subject of many investigations at present, especially on numerical simulation; the number of papers published on this topic, being already large, is steadily growing. Kruse /1/ gives a review of the phenomena influencing the efficiency of the on/off cycle; he concludes that a certain improvement in energy consumption is possible if a proper stepwise control is applied. Two-step capacity control for small refrigerating equipment has been numerically investigated by Kuijpers /2/. Although considerable improvements -in the order of 10% and more- could be established, the variation in capacity possible with two-step control is concluded to be too small for domestic refrigerating equipment. This means that, outside a certain ambient temperature range, on/off control has to be applied again. Full advantage of capacity control can only be obtained in case the capacity can be substantially reduced compared to the maximum possible; this e.g. is possible by the use of continuous control. However, this way of control may put extra requirements to the refrigerant charge to be used.

In an increasing number of publications, especially from Japanese authors, it is proposed to use a compressor of the continuous control type in a small refrigerating cycle, see e.g. /3/. Together with this type of compressor control the application of an expansion valve, and sometimes of extra capillary tubes, all controlled by microprocessors, is considered to be necessary. The aim of the control consists of obtaining a low superheat in the evaporator under all kinds of conditions. Whether this requires certain provisions in case a fixed refrigerant charge is used or whether adaptations to the charge present in the cycle would be desirable -given certain ambient conditions- is not made clear in literature.

There is a lower limit, as far as the cooling capacity needed for an appliance is concerned, for a stable and reliable operation of an expansion valve. Capacities required for average European appliances at low ambient temperatures, will normally be below this limit; the use of a capillary tube has to be preferred then. A study of the phenomena occurring in case continuous control is applied on a cycle equipped with 'conventional' components (normal heat exchangers and a capillary tube) therefore seems quite logical. It will serve as a reference in the discussion whether refrigerant charge and/or capillary tube have to be adapted as a function of the ambient temperature.

In this paper a steady-state model is proposed, used for the investigation of the aforementioned effects; details of the model can be found in section 2. The steady-state model comprises a part with certain assumptions for the void fraction as a function of the quality; an intercomparison of various models and the selection of a correlation is given in section 3. Results and conclusions are presented in the sections 5 and 6.

2. STEADY-STATE MODEL

A lot of research has been performed on refrigeration cycles, especially for heat pump operation, where steady-state simulation models are used. However, the present day tendency is to use dynamic models for a detailed study of component behaviour, due to the fact that mostly on/off control is applied.

On the other hand, steady-state models are realistic in case a cycle is continuously controlled, assuming the control only depends on variations in the ambient temperature and on the temperature setting of the refrigerating appliance.

The steady-state calculation is performed using mathematical functions for the mass transport through compressor and capillary tube

$$\dot{m}_{compr} = f_{compr} (T_e, T_c, T_{inlet}) \tag{1}$$

$$\dot{m}_{cap} = f_{cap} (T_e, T_c, T_{subc}, d_{cap}, l_{cap}) \tag{2}$$

The polynomial function for the compressor capacity is derived from calorimetric tests; the approximation function for the capillary mass flow from data obtained from a separate model which has been described elsewhere /4/, in which model thermodynamic equilibrium is assumed. Approximating expressions are also used for the heat transfer coefficients of the heat exchangers, derived from measurements

$$k_c = k_c (T_c - T_{amb}) , k_e = k_e (T_{cbt} - T_e) \tag{3}$$

For the heat flow from the condenser can be written

$$Q_c = \dot{m}_c \Delta h_c = \int_{\sigma=0}^{A_{supc}} k(T_\sigma) (T_\sigma - T_{amb}) d\sigma + k(T_c)(T_c - T_{amb})A_c + \int_{\sigma=0}^{A_{subc}} k(T_\sigma)(T_\sigma - Tamb) d\sigma \tag{4}$$

$$A_{cond} = A_{supc} + A_c + A_{subc} \tag{5}$$

where the heat exchanger is divided into regions of superheat, condensation and subcooling; the first and the last are subdivided into a large number of elements.

For the heat flow to the evaporator can be written

$$Q_e = \dot{m}_e \Delta h_e = k(T_e) (T_{cbt} - T_e)A_e + \int_{\sigma=0}^{A_{supe}} k(T_\sigma)(T_{cbt} - T_\sigma) d\sigma \tag{6}$$

$$A_{evap} = A_e + A_{supe} \tag{7}$$

where the heat exchanger is divided into regions of evaporation and superheat; the latter is subdivided into a large number of elements again.

In the heat exchanger calculations local heat transfer coefficients are used, i.e. heat transfer coefficients the value of which depends on the temperature of the element of the surface area considered.

No pressure loss is assumed over both condenser and evaporator.

The transmission loss of the appliance

$$Q_{trans} = f_{trans} (T_{amb}, T_{cbt}) \tag{8}$$

is determined via calculations using the geometric description, heat transfer coefficients for natural convection and radiation from the inner and outer walls, as well as certain material constants.

Calculation of a steady-state cycle consists of solving a set of non-linear algebraic equations. Calculations are performed according to an iterative scheme until convergence has been reached, i.e. the mass flows calculated for each of the components are equal and the heat transported to the evaporator, Q_e, is equal to the transmission loss of the appliance Q_{trans}.

The calculational method still has one degree of freedom, this means that refrigeration cycle conditions can be determined for a preset value of the superheat in the evaporator, the subcooling in the condenser or the total refrigerant charge in the system. This charge influences the superheat in the evaporator and hereby the capacity of the evaporator, i.e. the performance of the system as a whole. However, the effect of the amount of charge in a heat exchanger is directly dependent on assumptions for the void fraction; suitability of certain assumptions is dealt with in the next section.

3. VOID FRACTION

Next to the liquid present in the subcooled part of the condenser, the liquid refrigerant flowing in the evaporating and condensating region in the heat exchangers determines the charge in the cycle for an important part.

From literature it can be taken that, for a wide range of pressure and tempera-

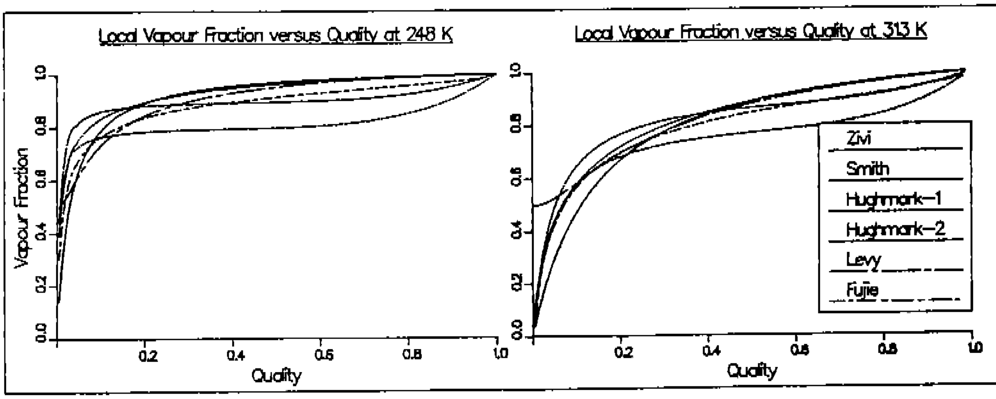


Fig. 1 Local void (vapour) fraction as a function of the quality (for typical evaporation and condensation temperatures of 248K and 313K, respectively), determined by the use of various kinds of correlation functions -for the relationship recommended by Hughmark /9/ two curves are given, representing the minimum and maximum values being realistic for the heat exchangers investigated in this contribution-

ture conditions, annular flow is observed over a large quality range during evaporation and condensation; this in despite of the circuiting being not straight in the heat exchangers normally applied. It therefore is reasonable to assume (semi-) annular flow in both evaporator and condenser and to neglect possible liquid entrainment.

Many investigations give correlations for the void fraction with quality for annular flow; most of these are of semi-empirical or empirical nature. In order to select an appropriate relationship for the description of the flow in the heat exchangers of domestic refrigeration equipment, five relationships have been compared. The comparison takes into account the correlations given by Zivi /5/ and Levy /6/, based on theoretical assumptions, those given by Smith /7/ and Fujie /8/ based on theoretical considerations together with one empirical constant, as well as the correlation published by Hughmark /9/, in which, via one empirical constant, all kinds of effects occurring in practice are incorporated.

In Fig. 1 the void fraction is given as a function of the quality by the use of the above correlations, for both evaporating flow at 248K (0.124 MPa) and for condensating flow at 313K (0.906 MPa). In case it is assumed that a uniform heat load is present over the evaporating region, a mean void fraction can be derived by integrating the local void fraction along the length of the two phase region. This can be done in the same way for both condenser and evaporator, yielding a mean void fraction as a function of the inlet quality (evaporator) -or the outlet quality (condenser)-

$$\alpha^{l} = \frac{1}{1 - x_{inlet}} \cdot \int_{x_{inlet}}^{1} \alpha(x)\ dx \tag{9}$$

In Fig. 2 the mean void fraction is represented as a function of the inlet (outlet) quality for the aforementioned evaporator and condenser conditions.

Taking into account the results given in Fig. 2 the following can be stated. The spread in the curves calculated via the Hughmark relationship is inherent to the method in which guess values should be used, to be determined from assumptions for heat exchanger flow present. The different results obtained in comparison with the other relationships may be due to the fact that the correlation has originally been derived from measurements on large installations; a, maybe too, large extrapolation towards domestic refrigeration equipment is necessary then. Relationships as those of e.g. Zivi and Levy yield a very high mean void fraction for higher inlet qualities. This implies that, in case variations in the liquid amount in a heat exchanger are calculated by the use of the steady-state method, application of this kind of correlations will lead to extremely large changes in evaporating length. This may not be realistic.

For the condensation region, assuming an outlet quality of zero, the difference between the various relationships is marginal (see Fig. 2, b, x=0). For evaporation pressures a lower void fraction than predicted by the majority of authors cited /5,6,7/ is given by Fujie /8/. This tendency is confirmed in practice by preliminary measurements on a refrigerating appliance /11/; also by Wedekind /12/ in a comparative

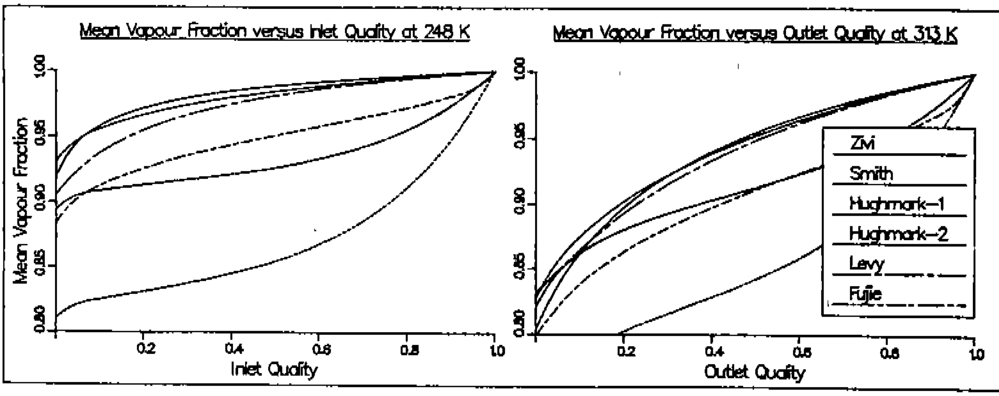


Fig. 2 Mean void fraction as a function of the inlet quality (or outlet quality in the case of the condenser) for the temperatures, equal as in Fig. 1; this mean fraction is determined for a dry evaporator (or a subcooled condenser, i.e. integration has been performed from inlet quality to 1.0. -for the relationship of Hughmark /9/ the same statements as given under Fig. 1 are valid-

analysis together with some experimental results obtained from refrigeration cycles. Correlations taken into account have all been published in the 1960's; improvements may have been found since then. Nevertheless, in a recent publication by Tandon /10/, presenting an intercomparison between the aforementioned correlations and an elaborate and detailed method of his own, no conclusions can be found which point at a superiority of methods for approximating the void fraction at higher (above 0.1) qualities, which have not been considered here. In the steady-state method described in this paper, the correlation as proposed by Fujie has therefore been implemented.

For the total charge in a system can be written

$$M = ((1-\alpha_e^l)\ \rho_{le} + \alpha_e^l\ \rho_{ve})\ V_e + \int_{v=0}^{v=V_{supe}} \rho(T_v)\ dv + \rho(T_{compr})\ V_{compr}$$
$$+((1-\alpha_c^l)\ \rho_{lc} + \alpha_c^l\ \rho_{vc})\ V_c + \int_{v=0}^{v=V_{subc}} \rho_{l,subc}(T_v)\ dv + \int_{v=0}^{v=V_{supc}} \rho(T_v)\ dv \qquad (10)$$

in which the mean void fraction is now calculated using the Fujie correlation. The mass in the superheated part of the evaporator and in the superheated suction pipe, the superheated part of the discharge circuit and the mass present in the compressor are calculated using temperature-pressure dependent relationships as given by Rombusch and Giesen /13/. These are also used for the determination of other thermodynamic data.

For the larger part the total mass in the refrigeration system is determined by the parts of the heat exchangers filled with liquid. A low charge results in a high superheat and a large superheated surface area, implying a low suction pressure and a low thermodynamic efficiency of the cycle. There is direct interaction between the capillary tube characteristics and the refrigerant charge applied, see section 5.

4. APPLIANCE CHARACTERISTICS

The appliance on which the effects of continuous control have been investigated, is a 330 l natural convection based upright freezer, equipped with eight evaporator shelves. The transmission loss of this freezer amounts to 46.5 W in case of ambient and freezer temperatures of 298K and 255K, respectively. It is assumed that the freezer is fully loaded with packages, of which the warmest package temperature is equal to 255K (-18 C) in the continuous control mode. For the air inside an average temperature of 252K (-21 C) can be taken under this condition, which value has been derived from measurements.

Heat transfer relationships for the evaporator (loaded condition, which influences the heat transfer coefficient) and condenser have been determined in separate measurements /11/, results of which are confirmed elsewhere /14/. For the heat transfer coefficients the following approximating functions are used (surface area evaporator $2.56 m^2$, condenser $1.10 m^2$) :

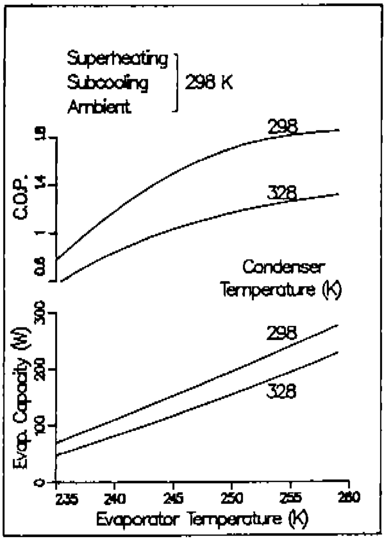


Fig.3 Capacity and COP of the -commercially available- compressor applied in the 330 l freezer investigated. Subcooling, superheat and ambient temperature amount to 298K (25 C)

$$k_{cond}(\Delta T) = 0.0115\,\Delta T + 1.449\sqrt{\Delta T},\quad \Delta T = T - T_{amb} \tag{11}$$

$$k_{evap}(\Delta T) = 0.0536\,\Delta T + 1.893\sqrt{\Delta T},\quad \Delta T = T_{cbt} - T \tag{12}$$

The appliance is normally equipped with a capillary of 0.71mm diameter (3.5m length). Capacity and input power of the compressor are derived from catalogue data of a commercially available compressor of the hermetic type; data are given in Fig. 3. Full capacity is presumed to be required at 316K (43 C) ambient. In order to have a clear presentation of the effects of continuous control it is assumed that lower capacities can be provided by the compressor at equal coefficients of performance (COP). This will not be valid in practice, where the COP will decrease towards lower capacities for all the control methods known. However, next to indications on a possible decrease in energy consumption the main purpose of this study is to show whether good refrigeration conditions can be provided in the evaporator over a wide range of ambients. For the calculation of these conditions only the capacity function, i.e. the resulting mass flow through the compressor, is important.

5. RESULTS

Using the steady-state model, calculations have been made for three control strategies -the results of which are presented in Figs. 4 and 5- :

-1. On/off control applying a compressor with a capacity large enough to keep the load at a temperature level of 255K (-18 C), for an ambient of 316K (43 C) -see Fig. 3-. The capillary capacity amounts to 5.5 l (see Table I) and the charge is kept constant (100g of R12);

-2. Continuous control using the same capillary and refrigerant charge as under 1.;

-3. Continuous control using the optimal capillary capacity and the optimal refrigerant charge (comparable to the use of an expansion valve controlled for low superheat as well as a storage of refrigerant).

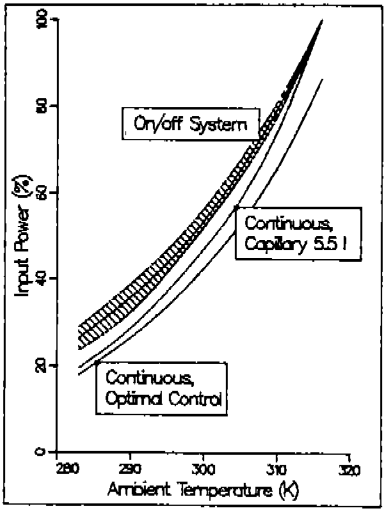
Data for the on/off control strategy are obtained from the steady-state model using full compressor capacity at different ambient temperatures, assuming a running time percentage to get the right cooling capacity. These results have to be considered in an approximative way since

- on/off losses are not taken into account (these losses will be in the range from 10% (298K) to 18% (283K ambient temperature /2/);
- the average COP of the on/off controlled compressor during running will be different from the one assumed in steady-state condition, dependent on the on/off frequency /1/.

Combination of these effects will result in an energy consumption as a function of the ambient temperature, which can be found in the shaded area (Figs. 4 and 5) around the curves calculated; the same holds for the running time percentage. In comparing control strategies it is convenient to consider the values calculated by the steady-state model as a reference.

capillary capacity ($l\ N_2$/ min)	capill. diameter (mm)	capill. length (m)
2.0	0.482	3.5
2.5	0.526	3.5
3.0	0.558	3.5
3.5	0.592	3.5
4.5	0.656	3.5
5.5	0.710	3.5
6.5	0.754	3.5

TABLE I. Capillary dimensions (length diameter) and equivalent nitrogen flow capacity (pressure difference 1 MPa)



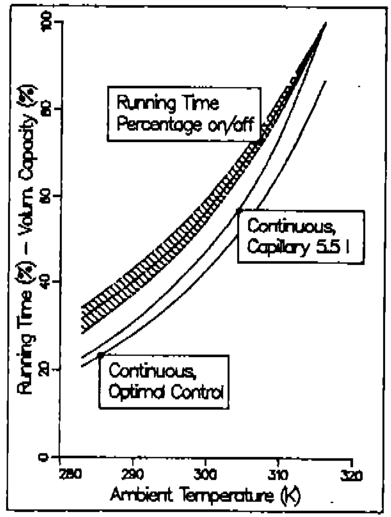


Fig. 4 Energy consumption dependent on the ambient temperature, in relative sense (normalized to 100% at an ambient of 316K (43 C) where continuous running is assumed) for

Fig.5 Running time percentage and volumetric capacity of the on/off and continuously controlled systems (normalized to 100% at an ambient of 316K) for

- an on/off controlled system (capillary 5.5 l);
- a continuously controlled system (capillary 5.5 l, fixed charge);
- a continuously controlled system (optimal expansion and charge).

When applying the capillary with a 5.5 l capacity in the continuous mode, savings in energy consumption between 0 and 25% can be observed, dependent on the ambient temperature. This clearly shows the saving potential of continuous control, which is caused by the fact that higher evaporation and lower condensation temperatures prevail in the heat exchangers. It should be emphasized here that the refrigerant charge is kept constant over the ambient range, which results in a large superheated area in the evaporator at low ambient temperatures (280-290K). In Fig. 5 the variation in the volumetric capacity of the compressor is given. Whilst from 316K down to 283K the transmission loss of the appliance is decreased by a factor of about two, the volumetric capacity of the compressor can be reduced by a factor of four.

The continuously controlled mode can even be made more efficient by an optimization of the capillary tube capacity and the charge as a function of the ambient. In this optimization the important requirement is that the evaporator has to be filled with liquid for a large part, i.e. for more than 90%. This in order to have a proper functioning of the evaporator. The resulting, lowest energy consumption obtainable (as e.g. by application of a continuous adaptation of expansion element and the refrigerant charge) is also given in Fig. 4. However, in case capillary tube(s) are used, this would require a switch in capillary tube capacities. This should be avoided in small refrigerating equipment where it is realistic to use one capillary and a fixed charge over a certain ambient range. Whether it will be possible to approximate this energy consumption with a fixed capacity and charge, is described below.

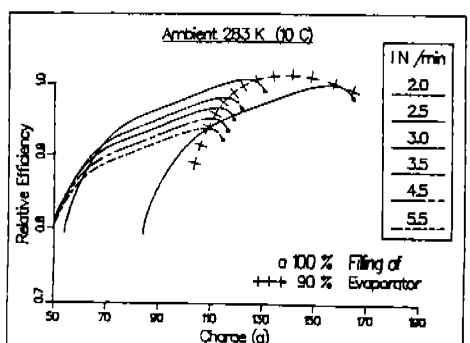


Fig. 6 Influence of the capillary tube capacity on the efficiency of the cycle -in relative sense-, as a function of the refrigerant charge applied. Ambient temperature 283K (10 C).

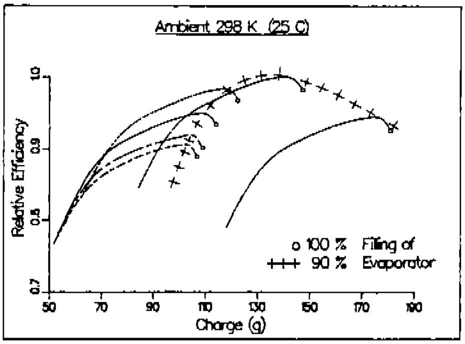


Fig. 7 Idem Fig. 6
Ambient temperature 298K (25 C)

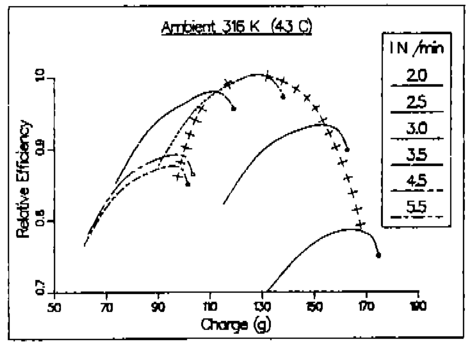


Fig. 8 Idem Fig. 6,
Ambient temperature 316K (43 C)

In Figs. 6, 7 and 8 the influence of the capillary capacity on the efficiency of the cycle is presented, dependent on the refrigerant charge applied. The efficiency is defined as the ratio of the evaporator capacity (transmission loss) and the compressor input power. In the figures the influence is given in relative sense, i.e. the efficiency maximum obtainable (for certain capillary capacity and charge) is taken to be 1.0; this in order to give comparable figures for different ambient temperatures -characterized by different efficiencies-. In Figs. 6 to 8 a curve is given as well which represents the "90% filling grade" of the evaporator.

The highest efficiency at an ambient of 316K (43 C) appears to occur when using a capillary capacity of 3.0 l together with a refrigerant charge of slightly more than 130g. This charge, however, would lead to "overcharging" -low efficiency- at lower ambient temperatures. When using this charge of 130g at a low ambient temperature, the best choice would be a capillary with a somewhat lower capacity (between 2.0 and 2.5 l capacity).

In Fig. 9 the charges of refrigerant are given which result in a "90% filling grade" of the evaporator, dependent on the capillary capacity and the ambient temperature. The curves represent the same points as given in Figs. 6 to 8 by the "90% filling grade" lines. Instead of the capillary capacities of 3.0 l and smaller, for which the necessary charge varies very much with the ambient temperature, application of a capacity of 3.5 l yields better results with respect to the purpose aimed at in this investigation. Together with a refrigerant charge of 113g this capillary capacity will yield a good filling of the evaporator over the whole ambient temperature range. It results in a rough 5% decrease in efficiency, compared to the values maximum obtainable (see Figs. 6 to 8).

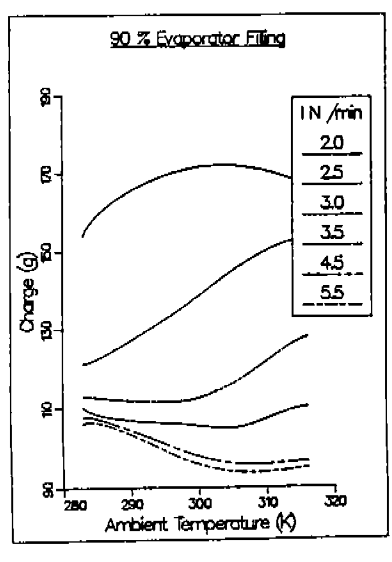


Fig. 9 Charges of refrigerant, the application of which results in a "90% liquid filling grade" (i.e. 90% of the heat exchanger tubing is in contact with evaporating liquid) of the evaporator. The charge necessary is given as a function of the ambient temperature for a number of capillary tube capacities (from 2.0 to 5.5 l).

As far as energy consumption is concerned, the results are summarized in Fig. 10. Here the savings in energy consumption achievable with a capillary capacity of 5.5 l as well as with optimal controlled expansion are given -compare Fig. 4-. A third curve presents the saving possible with the aforementioned capillary tube capacity of 3.5 l. It can be observed that, compared to the highest values possible, a 2-5% lower saving is being realized with this capillary capacity. Results, especially at higher ambient temperatures, can be considered to be more than satisfying.

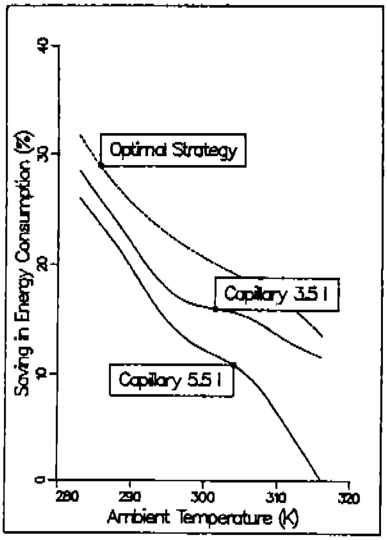


Fig. 10 Saving in energy consumption -in comparison with an on/off controlled cycle- possible with an optimally controlled expansion element and charge, and with a capillary of 5.5 l capacity. Values obtainable by the application of the 3.5 l capillary and a fixed charge (113 g) are given as well.

6. CONCLUSIONS

The use of a steady-state model is an appropriate way for the calculation of the behaviour of a continuously controlled refrigeration cycle. In this model certain assumptions for the liquid layers in the heat exchanger pipes play an important role.

Compared to an approximative calculation of the energy consumption of an on/off cycle, reductions from 15 to 30% are calculated in case of continuous control applying the optimal expansion element and a control of the refrigerant charge. Practical solutions for small refrigerating appliances are negatively influenced by the capillary tube behaviour. On the other hand, it has been shown that an optimization can yield the dimensions of one capillary tube which is best given a fixed charge and certain ambient conditions. Using the steady-state model in case of a practical example the result has been obtained that application of this "best" capillary tube induces a saving in energy consumption comparable to the saving determined for the optimal configuration. A saving of about 20% compared to the consumption calculated for the on/off cycle seems achievable.

Investigations have been performed on a cycle without using a capillary tube-suction gas heat exchanger. Here too high charges yield an "overcharge", which directly results in a sharp decrease in efficiency. By applying a liquid-suction gas heat exchanger this decrease will be smaller, but the effect will always be negative. Optimization of capillary tubing may then be easier, since a certain small overcharge -occurring in a small abient temperature range- can be tolerated then. It is a subject of further study.

In order to get a good overview of the saving potential of a large variety of configurations an "ambient temperature profile" -comparable to the reference heating seasons in heat pump calculations- would be of much use. Via this "profile" the influence of the whole ambient range can be incorporated in a weighed form in a sort of performance factor.

NOMENCLATURE

A	surface area	$/m^2/$	$\dot{m}$	mass flow	/kg/s/
d	diameter	/m/	M	mass	/kg/
h	specific enthalpy	/Nm/kg/	Q	heat flow	/W/
k	heat transfer coefficient	$/W/m^2K/$	T	temperature	/K/
l	length	/m/	V,v	volume	$/m^3/$

x	quality	/-/	compr	compressor
α	void fraction, local	/-/	cond	condenser
α'	void fraction, mean	/-/	e	evaporation
ρ	density	$/kg/m^3/$	evap	evaporator
σ	surface area	$/m^2/$	l	liquid
subscripts			subc	subcooled
			supc	condenser superheated
amb	ambient		supe	evaporator superheated
c	condensation		v	vapour
cbt	cabinet		v	volume element

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SUMMARY

The control mode usually applied in domestic refrigeration apparatus is the on/off control, with application of a capillary and a fixed refrigerant charge. Continuous control of cooling equipment -dependent on ambient and preset cabinet temperature- yields an energy saving which is maximal in case the expansion element and charge are optimally controlled. Given the sort of cooling equipment this is not realistic and also for continuous control application of one capillary and a fixed charge have to be preferred. Implications concerning the energy saving still realizible have been investigated via numerical simulation.

For this simulation a steady-state program is used, appropriate in case of continuous control. The mathematical model included for the 'void fraction' in the heat exchangers through which a two-phase mixture is flowing, is very important; it has been selected from a literature search. A large number of capillary capacities, together with different charges, have been simulated in case of an upright freezer; this kind of appliance puts extra requirements to the evaporator filling. It has been shown that an optimization is possible, i.e. that only one capillary capacity -presuming a fixed charge- leads to a good evaporator behaviour, independent of the ambient. Using this configuration a large part of the energy saving maximum possible can be realized.

RÉGLAGE CONTINU DES APPAREILS FRIGORIFIQUES; DELIMITATIONS CAUSÉES PAR LE COMPORTEMENT DU CAPILLAIRE

RESUME: Normalement, de petits appareils frigorifiques sont réglés d'une façon 'on/off', tandis qu'un seul capillaire et un charge fixe de réfrigérant y sont appliqués. Le réglage continu des appareils frigorifiques en fonction de la température ambiante, mène à une restriction maximale de la consommation de l'énergie, quand un réglage optimal de l'élément de l'expansion et du charge sont supposés. Pourtant, cela n'est pas réaliste pour de petits appareils domestiques et il faut préférer le capillaire unique et le charge fixe, aussi pour ce mode de réglage. Ce que sont les implications pour la restriction eventuelle de la consommation de l'énergie, a été examiné dans cet article par la voie de la simulation numérique. Pour cela, un programme "steady-state" est employé. Les admissions pour le "void fraction" y sont très important. Un bon nombre de capacités du capillaire a été examiné pour un congélateur vertical. Il parait qu'il n'y a qu'un seul capillaire qui mène à un bon comportement du cycle avec un charge fixe du réfrigérant; ceci indépendant de la température ambiante. Ainsi, par cette configuration il est possible que la consommation de l'énergie peut être restreinte presqu'au degré de la restriction maximale.