HEAT PUMPS

Solving Energy and Environmental Challenges

Proceedings of the 3rd International Energy Agency Heat Pump Conference, Tokyo, Japan, 12–15 March 1990

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PERGAMON PRESS

Member of Maxwell Macmillan Pergamon Publishing Corporation

OXFORD · NEW YORK · BEIJING · FRANKFURT

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Interaction of Heat Exchanger Design and Heat Pump Efficiency

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ABSTRACT

A computer programme manageable on a PC was developed to simulate the realistic behaviour and performance of vapour-compression heat pumps. Main emphasis was laid on modelling the thermo- and fluiddynamic behaviour of the heat exchangers. The programme describes the whole thermodynamic cycle of the heat pump, including subcooled conditions at the exit of the condenser and superheated ones at the exit of the evaporator.

The interaction between the heat exchanger behaviour and the coefficient of performance (COP) of the heat pump is complex, and impaired heat transfer can deteriorate the heat pump efficiency sensitively with low temperature differences between the heat source and the heat sink. For a given temperature of the heat source the COP improves with increasing heat transfer surfaces of the condenser or of the evaporator. It is shown that at given values of heat source temperature and heat flux, the condenser design influences not only the thermodynamic circuit but also the evaporator performance.

KEYWORDS

heat exchanger design, coefficient of performance, thermodynamic heat pump cycle, two-phase flow correlations, heat transfer, pressure drop

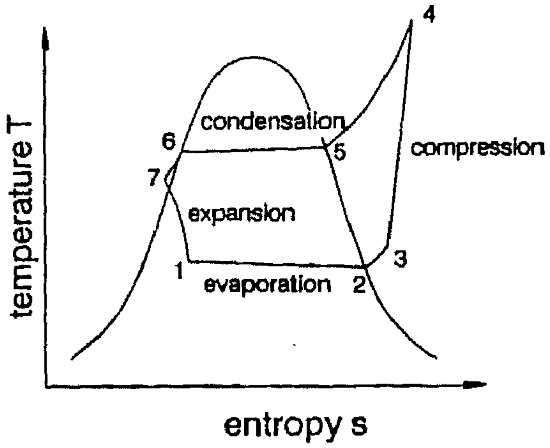
INTRODUCTION

The share of heat pumps in the market for the energy supply to heating systems is strongly depending on fair economical performance. Important criteria for the economy of heat pumps are the COP and the investment costs. To the latter one not only the compressor but also the heat exchangers evaporator and condenser contribute. COP and investment costs interfere with each other and the design of the heat exchangers influences the COP by improved or impaired heat transfer and by smaller or larger pressure drop in these apparatus.

Boiling and condensing are thermo- and fluiddynamic phenomena with high heat transfer coefficients and therefore, a reduction of the heat transfer area and by this of the investment costs should be possible to a certain extent for many plants available in the market without too serious deterioration of the COP. However, from a certain grade of reduction a sudden decrease of the COP has to be expected.

HEAT PUMP CIRCUIT AND SYSTEM OF CONSTITUTIVE EQUATIONS

To investigate the interaction between COP and heat exchanger design a simple heat pump circuit was used, as shown in Figure 1.



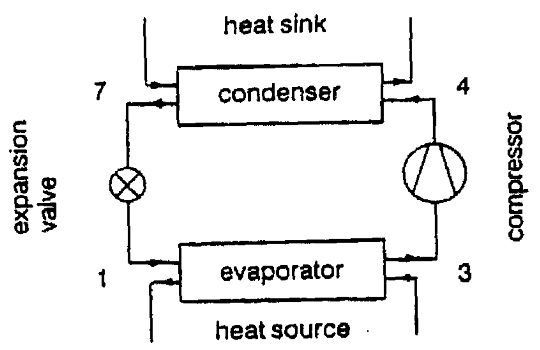


Fig. 1. Thermodynamic cycle and plant scheme of a vapour-compression heat pump

The circuit works with a single stage compressor, taking slightly superheated vapour from the evaporator and compressing it with an efficiency referred to isentropic situation of

$$\eta_{is} = \frac{h_{4is} - h_3}{h_4 - h_3} \tag{1}$$

The compression efficiency depends on the pressure ratio of the compressor, for which Bukau (1983) gives the equation

$$\eta_{is} = 0,849 + 0,0025\frac{p_4}{p_3} - 0,002\left(\frac{p_4}{p_3}\right)^2 \tag{2}$$

The coefficient of performance is defined as usual in the literature

$$COP = \frac{\dot{Q}}{P} = \frac{h_4 - h_7}{h_4 - h_3}\eta_{em} \tag{3}$$

with η_{em} taking into account the electrical and mechanical losses in the electric motor driving the compressor and in the compressor itself.

The refrigerant R12 was used as a working fluid.

The equations of conservation describing the mass, momentum and energy balance in the heat pump system and between its components are well-known in the literature, wherefore it is not necessary to present them here. Information, however, must be given how the fluid

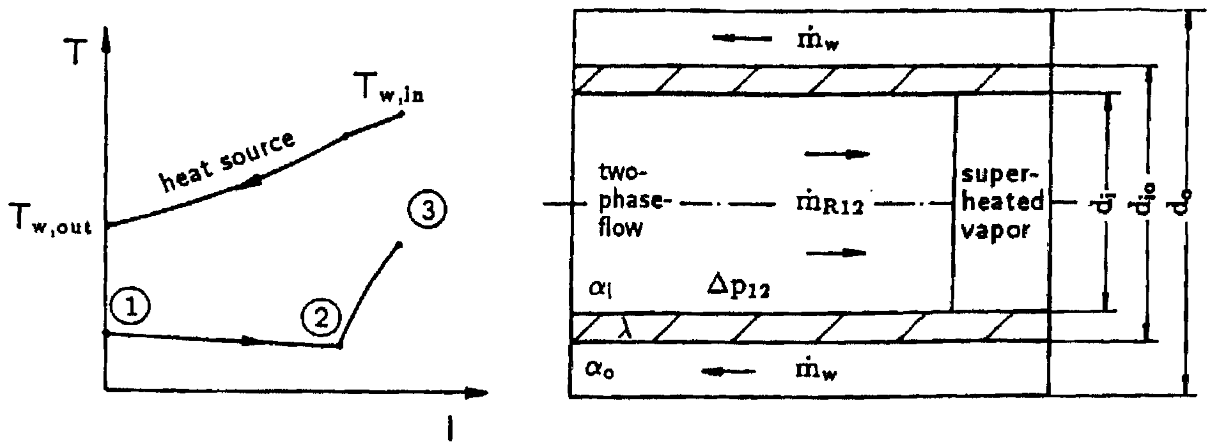


Fig. 2. Axial temperature distribution and modelling of the evaporator

flow and the heat transfer were treated in the heat exchangers. The evaporator was assumed to be of coaxial design as shown in Figure 2.

The refrigerant enters the inner tube as a two-phase mixture after leaving the expansion valve, and is superheated after complete evaporation. The heat is added to the boiling refrigerant R12 from water flowing in the annulus between the inner and outer tube. To keep the logarithmic mean temperature difference

$$\Delta T_M = \frac{\Delta T_o - \Delta T_i}{\ln \frac{\Delta T_o}{\Delta T_i}} \tag{4}$$

to a minimum countercurrent flow was applied. Calculating the pressure drop of the refrigerant, regions of single-phase and of two-phase flow have to be defined carefully and separated. To make the calculation procedure for the PC not too complicated, the point of beginning single-phase vapour flow was simply determined by using the energy-balance and dry-out-, as well as entrainment-correlations were not used in the calculation.

The single-phase pressure drop results from the well-known equation

$$\Delta p = \lambda\, l \frac{\rho}{2} \frac{w^2}{d_{hyd}} \tag{5}$$

where the friction coefficient λ can be determined using the Blasius-equation

$$\lambda = \frac{0,3146}{\sqrt[4]{Re}} \tag{6}$$

because of the limited range of Reynolds numbers to be expected in heat exchangers of heat pumps. In two-phase flow the procedure for predicting pressure drop is more complicated. Several correlations and models exist in the literature. Two of them were selected here: the first one given by Martinelli (1948) because it is widely known in the literature and another, newer one by Friedel (1979) which is based on several thousand experiments with various substances, ranging from water to refrigerants, hydrocarbons, and up to liquid metals.

Martinelli defined a two-phase multiplier which can either be referred to a situation, where the liquid or the gaseous part of the mixture would flow alone in the channel. We used the multiplier referred to the gaseous situation

$$\Phi_V^2 = \frac{\Delta p_{2ph}}{\Delta p_V} \tag{7}$$

which can be predicted according to Martinelli, using the simple equation

$$\Phi_V^2 = 1 + C_{tt}\, X_{tt} + X_{tt}^2, \quad C_{tt} = 20 \tag{8}$$

in which X_{tt} is the so-called Martinelli parameter, defined as

$$X_{tt} = \left(\frac{\rho_V}{\rho_L}\right)^{0,5} \left(\frac{\eta_L}{\eta_V}\right)^{0,1} \left(\frac{1-\dot{x}}{\dot{x}}\right)^{0,9} \tag{9}$$

Friedel (1979) uses a two-phase flow multiplier, too, however, he refers it to a situation, where the whole mixture – liquid and vapour – is assumed to flow in liquid condition in the tube.

$$R = \frac{\Delta p_{2ph}}{\Delta p_{1ph,L}} \tag{10}$$

Friedel gave the following correlation for calculating this two-phase multiplier

$$R = A + 3,34\, \dot{x}^{0,685}(1-\dot{x})^{0,24} \left(\frac{\rho_L}{\rho_V}\right)^{0,8} \left(\frac{\eta_V}{\eta_L}\right)^{0,22} \left(1 - \frac{\eta_V}{\eta_L}\right)^{0,89} Fr_L^{-0,047} We_L^{-0,0334} \tag{11}$$

with the dimensionless parameter A, the Froude and the Weber number.

$$A = (1-\dot{x})^2 + \dot{x}^2 \left(\frac{\rho_L \lambda_V}{\rho_V \lambda_L}\right)$$

$$Fr_L = \frac{\dot{m}^2}{g\, d_{hyd}\, \rho_L^2} \tag{12}$$

$$We_L = \dot{m}^2 \frac{d_{hyd}}{\rho_L\, \sigma}$$

The single-phase friction factors λ_V and λ_L can be calculated with equations given in the literature. Friedel recommends for laminar flow

$$\lambda_{V,L} = \frac{64}{Re_{V,L}} \quad \text{for} Re_{V,L} \leq 1055 \tag{13}$$

and for turbulent situation

$$\lambda_{V,L} = \left(0,86859 \ln \frac{Re_{V,L}}{1,964 \ln Re_{V,L} - 3,8215}\right)^{-2} \quad \text{for} \quad Re_{V,L} > 1055 \tag{14}$$

The quality – vapour content – in eq. (9) is – as usual in the literature – defined as

$$\dot{x} = \frac{\dot{m}_G}{\dot{m}_G + \dot{m}_F} = \frac{h - h'}{h'' - h'} = \frac{s - s'}{s'' - s'} \tag{15}$$

Heat transfer coefficients for single-phase flow are well-known in the literature and we used the correlation presented by Gnielinsky (1976)

$$Nu = \frac{\frac{\xi}{8}(Re - 1000)Pr}{1 + 12,7\sqrt{\frac{\xi}{8}}(Pr^{\frac{2}{3}} - 1)} \left(1 + \left(\frac{d_{hyd}}{l}\right)^{\frac{2}{3}}\right) \tag{16}$$

with

$$\xi = (1,82 \lg Re - 1,64)^{-2} \tag{17}$$

For predicting the heat transfer coefficient with boiling, the equation by Stephan and Preußer (1979)

$$\alpha_R = 0,1 \left(\frac{q \ V_{Bu}}{\lambda_L \vartheta_S}\right)^{0,674} \left(\frac{\rho_V}{\rho_L}\right)^{0,156} \left(\frac{r D_{Bu}^2}{a_F^2}\right)^{0,371} \left(\frac{a_L^2 \rho_L}{\sigma D_{Bu}}\right)^{0,35} \left(\frac{\eta c_p}{\lambda}\right)_L^{-0,162} \frac{\lambda_L}{D_{Bu}} \tag{18}$$

was used, in which for refrigerants the mean diameter of the departing bubbles can be calculated with the following formula

$$D_{Bu} = 0,0146\beta \left[\frac{2\sigma}{g(\rho_L - \rho_V)}\right]^{0,5}$$

$$\beta = 35^\circ \quad \text{refrigerants} \tag{19}$$

The thermal diffusivity

$$a_F = \frac{\lambda_L}{\rho_L c_{pL}} \tag{20}$$

has to be calculated with the liquid properties. For the overall heat transfer coefficient

$$\frac{1}{k} = \frac{1}{\alpha_W} \frac{d_i}{d_{io}} + \frac{1}{\alpha_R} + R_W + R_{fouling} \tag{21}$$

also the heat resistance of the wall material and of a fouling layer has to be taken in account if necessary.

The heat transfer elements of the condenser are also assumed to be designed in the form of concentric cubes. The refrigerant is now flowing in the annulus however, and water serving as heat sink in the centre tube as Figure 3 shows.

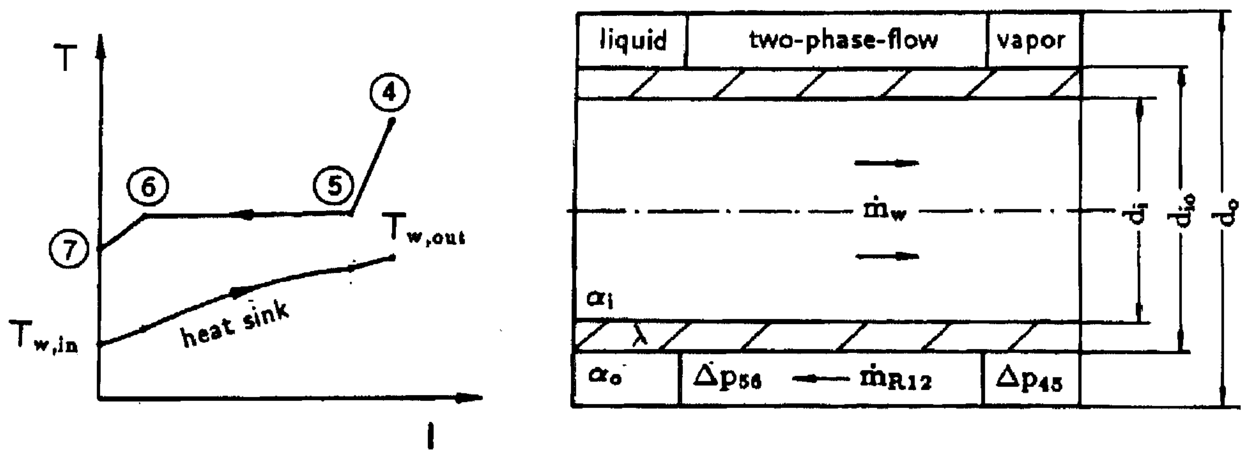


Fig. 3. Axial temperature distribution and modelling of the condenser

The refrigerant R12 which enters the condenser as superheated vapour is condensed and the liquid is slightly subcooled. In the single-phase regions the equation by Gnielinsky (1976) was used to predict the heat transfer coefficient and for the condensing part an equation proposed by Bukau (1983)

$$\alpha_R = \frac{0.725(1560 - 6T_S)}{(d_{io}(T_S - T_W))^{\frac{1}{4}}} \tag{22}$$

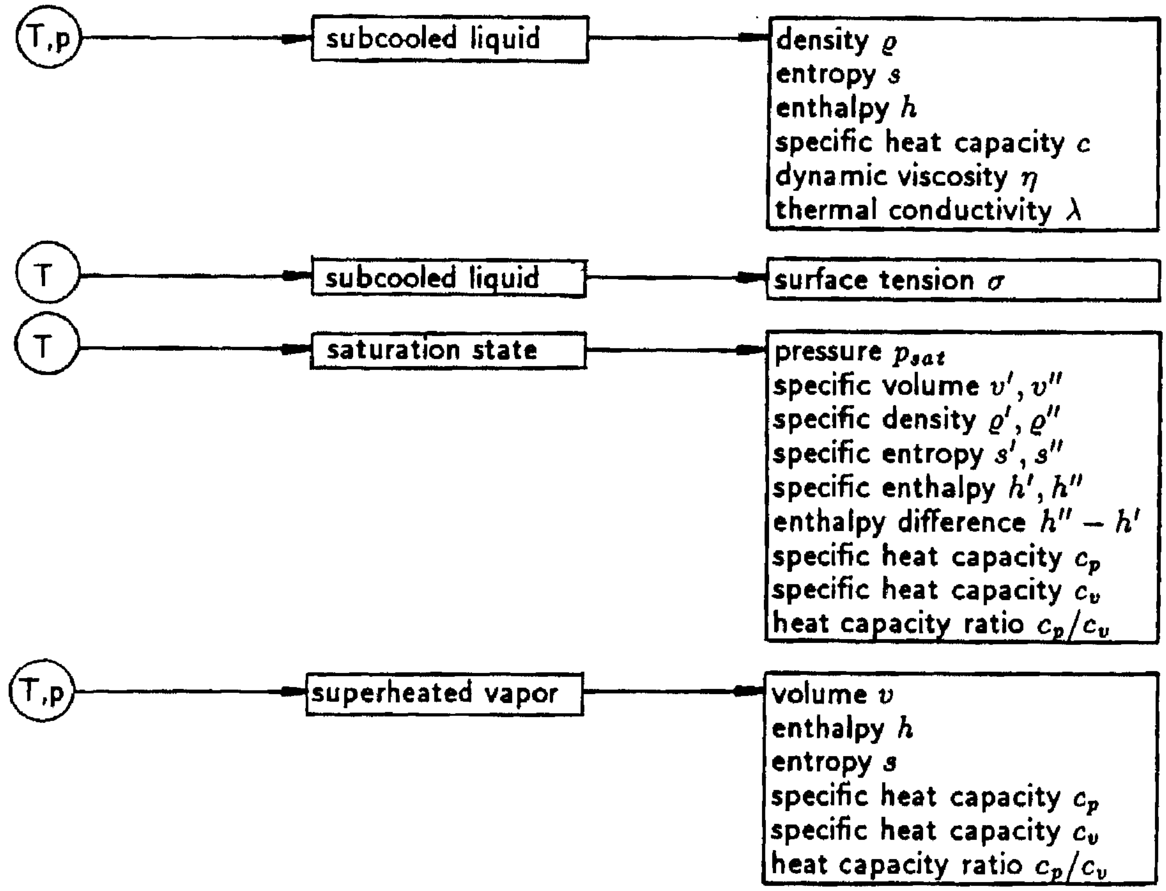
based on Nusselt's liquid film theory was used. The overall heat transport coefficient can again be calculated with eq. (21) and for the heat transport resistance due to fouling, the value

$$R_{fouling} = 0.0001 \frac{m^2K}{W} \tag{23}$$

was assumed.

For calculating the thermal and caloric properties as well as the transport properties of the working fluid R12 a special subroutine was developed and implemented in the PC-computer programme. This subroutine is based on equations of state and correlations proposed by Watson (1975). The properties which can be predicted by this subroutine are listed in Figure 4.

The subroutine works in the liquid subcooled region, at the saturation lines of liquid and vapour, and in the superheated vapour region. The upper limit of the superheated region is given by the isotherm 150°C, and by the critical isobar of R12 p=41,2 bar. In the liquid subcooled region, the subroutine works down to –40°C and the pressure limitation is again the critical isobar.



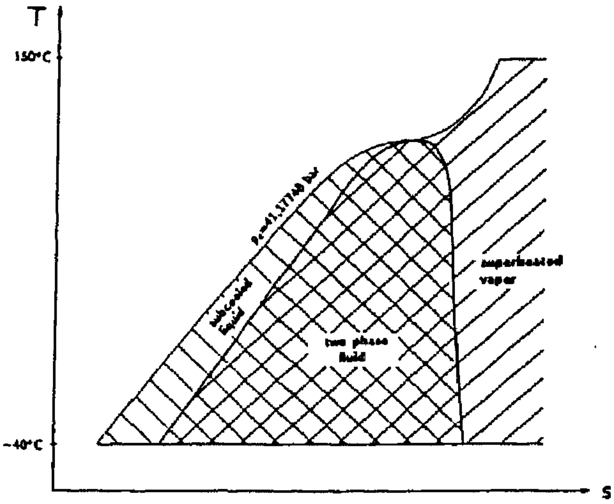


Fig. 4. Computable regions and properties of R12

PARAMETER STUDIES AND DISCUSSION OF RESULTS

With the programmed heat pump model and system of equations an example for a heat pump circuit was studied. As a practical example the heat pump system for heating a one family house with a heating power of 15 kW was chosen. As reference data for normal operation - 100 % power - the following values were assumed:

power consumed in the compressor: 5.7 kW
heating water temperature leaving the heat pump: 60°C
volumetric flow of the heating water: 1.3 m^3/h
water-temperature of the heat source entering the heat pump system: 10°C
water-flow of the heat source: 1.8 m^3

In the parametric studies, the design of the evaporator and of the condenser was varied and by this the mean temperature differences between the primary and secondary side. Also the temperatures of the heat sink and of the heat source were used as parametric variables.

Let us first discuss alterations in the evaporator design. The heat transporting area of a heat exchanger can be varied by changing the length and the diameter of the tubes. Reducing the length, the coefficient of performance responds only weakly at the beginning as Figure 5 shows, which is due to two effects: the high heat transfer coefficient on the boiling side makes the heat exchanger little sensitive to slight reductions in the heat transfer area and shorter tubes produce less friction pressure drop from which the compression rate benefits. Beyond a certain reduction in the heat transfer area, however, the COP starts to become strongly reduced which is mainly due to the strong decreasing of the main temperature difference. In Figure 5 the situation is given for different heat source temperatures and also lines of constant mean temperature difference between the water and the boiling refrigerant are drawn in.

Figure 6 shows the influence of the evaporator design on the COP when the temperature of the heat sink is varied. Comparing with Figure 5 one realizes that the COP is much more sensitive to the heat sink temperature – the temperature of the house heating system – than to the heat source temperature which is well-known from the literature.

A similar effect of the heat exchanger design on the COP as with the evaporator can be experienced when altering the heat transfer surface of the condenser. The condensing heat transfer coefficient, however, is much lower than that with boiling and therefore, a reduction in the heat transfer area of the condenser affects the COP much more than we have seen with the evaporator.

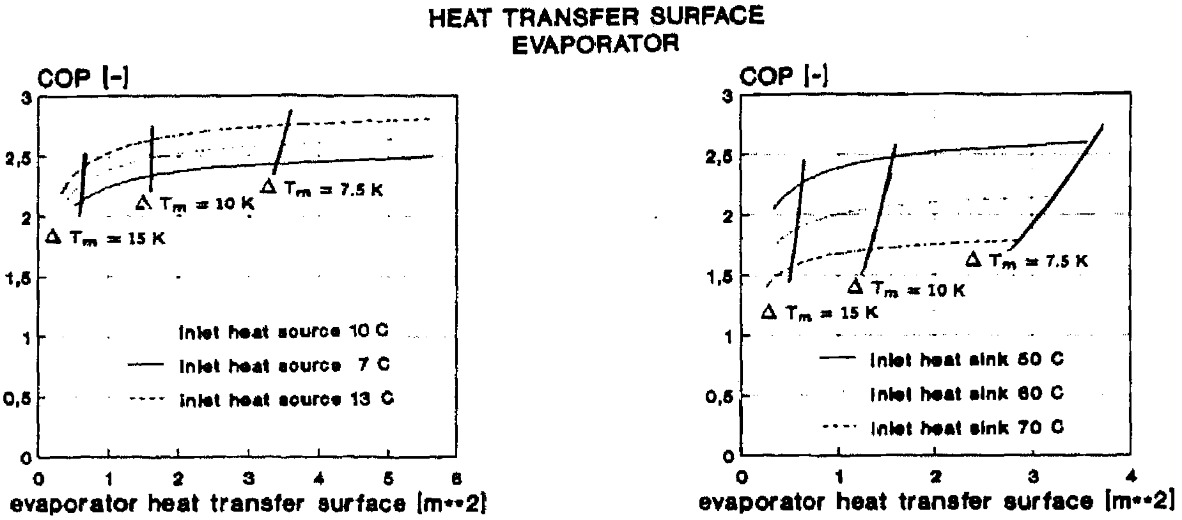


Fig. 5/6. Evaporator: Influence of heat transfer surface

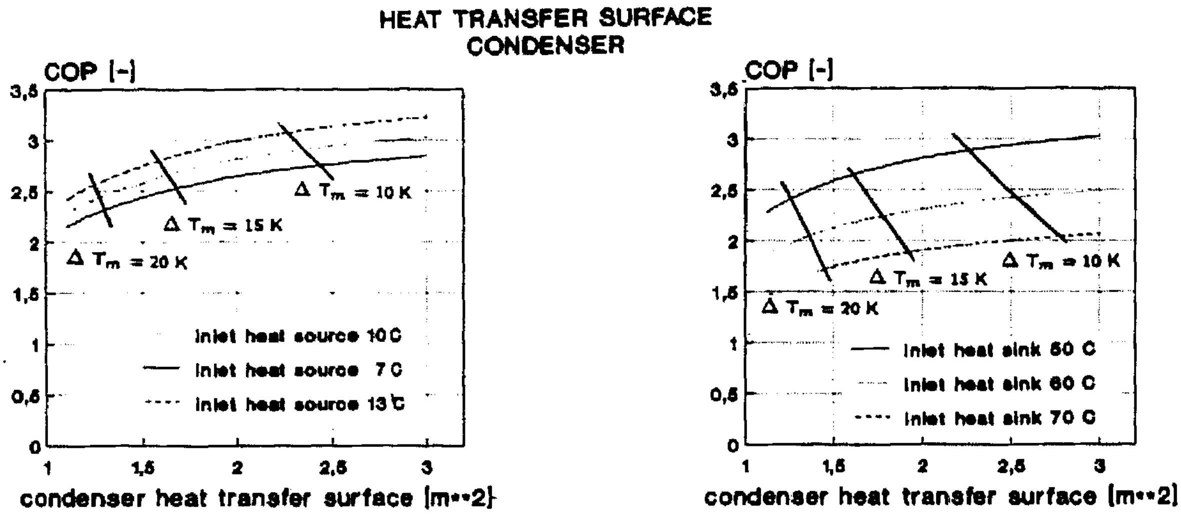
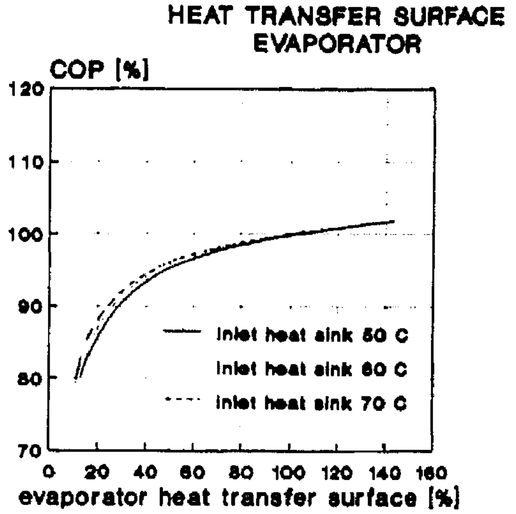


Fig. 7/8. Condensor: Influence of heat transfer surface

Figure 7 shows the influence of the condenser heat transfer area on the COP for different heat source temperatures and in Figure 8 the heat sink temperature is the variable parameter. Again the heat sink temperature has stronger influence on spreading of the curves than a variation of the heat source temperature. Contrary to the situation in the evaporator, the lines for constant mean temperature differences between primary and secondary side now have positive inclination.

A perhaps better impression of the influence of heat exchanger variations mediates a plotting of relative data related to a normal sizing which may be regarded as a 100 % design. As it can be seen in Figure 9, a reduction of the heat transfer area of the evaporator by 60 % impairs the COP only by 5 points. On the other side an oversizing even by 40 % brings almost no benefit.

The much greater sensitivity of the condenser to an improper heat transfer sizing is demonstrated more pronounced in Figure 10. Already a reduction of the heat transfer area by 20 % reduces the COP by 5 points. So much more care has to be taken for the design of the



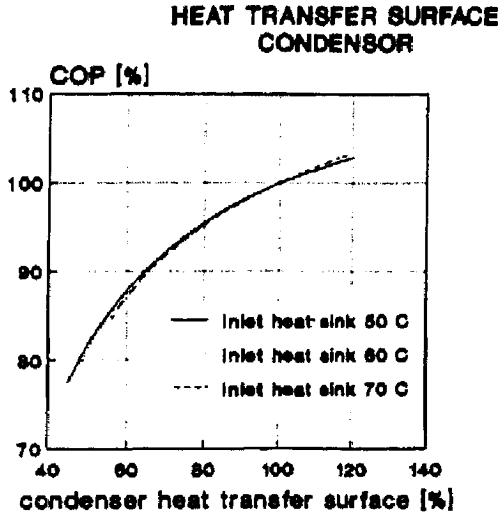


Fig. 9/10. Relative data related to normal sizing which is regarded as 100 % design

condenser than for that of the evaporator, when sizing a heat pump system.

As mentioned above, the heat transfer area of a tubular heat exchanger can be altered by shortening the tubes or by reducing the tube diameter. Smaller tube diameters increase the pressure drop. In two-phase flow there is more uncertainty to predict the pressure drop than in single-phase flow. The difference in these predictions by using Martinelli's or Friedel's equation for the evaporator under concern here are documented in Figure 11.

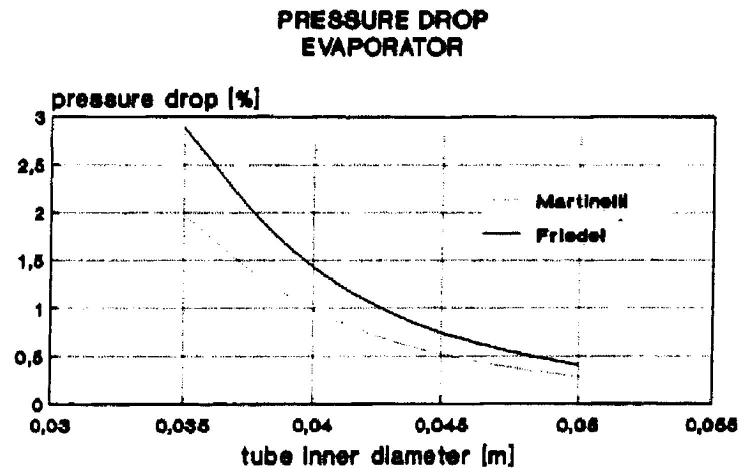
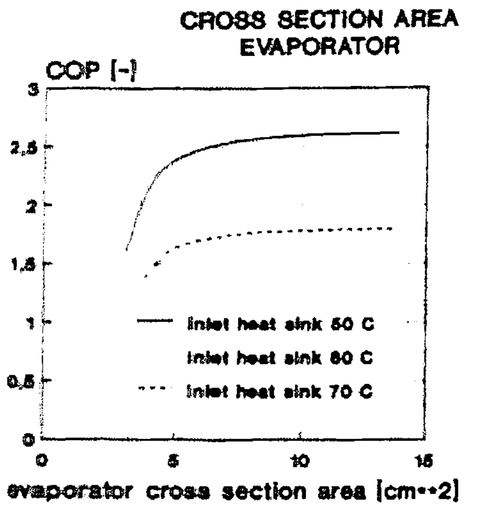


Fig. 11. Pressure losses computed by using the equations of Martinelli and Friedel

The calculation was performed stepwise along the flow path through the evaporator always taking into account the local vapour content which is increased along that tube due to evaporation. From the results of these two equations in Figure 11 one can see that the relative uncertainty of predicting the pressure drop is only in the order of 1 % and also the relative increase of the pressure drop – referred to the 100 % design point with reducing the tube diameter – is moderate. A reduction of the tube diameter of more than 10 % results in a pressure drop increase of approximately 5 %.

From this we can expect that a reduction of the cross section area of the heat exchangers has low influence on the COP. This influence is more pronounced in the evaporator than in the condenser as a comparison of the Figures 12 and 13 shows because the volumetric flowrate of the vapour and by this the two-phase flow multiplier is much higher there. In the condenser low vapour velocities exist and the fluid finally leaves the condenser in pure liquid form, whilst in the evaporator all fluid has to be changed into the vapour phase.



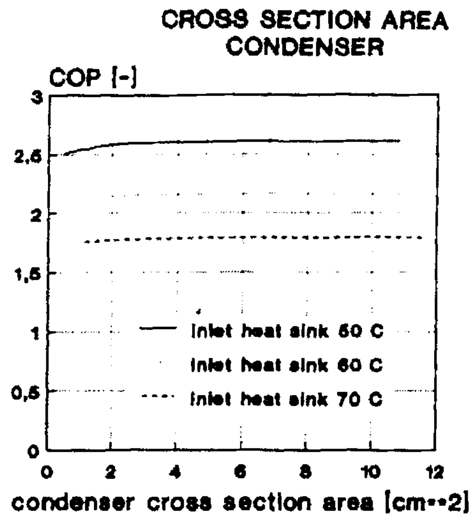


Fig. 12/13. Influence of cross section area

CONCLUSION

The computer programme developed allows to study in detail the influence of heat exchanger parameters – heat transfer area, flow channel cross section, mean temperature difference, heat sink temperature and heat source temperature – on the coefficient of performance. It is helpful for deliberations dealing with an optimum of the balance between investment- and operation-costs. In many designs probably the heat transfer area of the evaporator could be reduced without affecting the COP noticeable, whilst an improper design of the condenser has much stronger effects on the COP.

NOMENCLATURE

a	thermal diffusivity	$m^2 s^{-1}$
A	Friedel parameter	-
c_p	specific heat capacity	$jkg^{-1}k^{-1}$
C_{tt}	Martinelli constant	-
COP	coefficient of performance	-
d	diameter	m
d_{hyd}	hydraulic diameter	m
D_{Bm}	bubble mean diameter	m
Fr	Froude number	-
g	gravity	ms^{-2}
h	specific enthalpy	Jkg^{-1}
k	overall heat transfer coefficient	$Wm^{-2}K^{-1}$
l	length	m
$\dot{m}$	mass flow rate	kgs^{-1}
Nu	Nusselt number	-
p	pressure	Pa
Δp	pressure drop	Pa

P	compressor power	W
Pr	Prandtl number	–
q	surface heat flux	Wm^{-2}
$\dot{Q}$	heating flow rate	W
R	thermal resistance	m^2KW^{-1}
R	Friedel multiplier	–
r	enthalpy of evaporation	Jkg^{-1}
Re	Reynolds number	–
s	specific entropy	$Jkg^{-1}K^{-1}$
T	Temperature	K
ΔT_i	inlet temperature difference	K
ΔT_M	logarithmic mean temperature difference	K
ΔT_o	outlet temperature difference	K
w	velocity	ms^{-1}
We	Weber number	–
$\dot{x}$	quality	–
X_{tt}	Martinelli parameter	–

Greek symbols

α	heat transfer coefficient	$Wm^{-2}K^{-1}$
β	wetting angle	deg
η	dynamic viscosity	Pas
$\eta_{em} = \eta_{el} \times \eta_m$	electrical efficiency × mechanical efficiency	–
η_{is}	isentropic compression efficiency	–
ϑ_S	saturation temperature	$°C$
λ	friction coefficient	–
λ	thermal conductivity	$Wm^{-1}K^{-1}$
ϱ	density	kgm^{-3}
σ	surface tension	Nm^{-2}
ϕ	Martinelli multiplier	–

Subscripts

1,2,3,4,5,6,7	thermodynamic cycle points due to Figure 1
i	inner
L	liquid
V	vapor
W	wall

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