Int. J. Therm. Sci. (1999) 38, 325–330

Determination of the optimum high pressure for transcritical CO_2-refrigeration cycles

Friedrich Kauf*

Daimler-Benz AG, G 254, 70546 Stuttgart, Germany

(Received 8 June 1998, accepted 17 July 1998)

Abstract — Modern refrigeration systems have to fulfil many requirements, which include a rapidly growing number of environmental aspects. For compression systems, this entails demands for low energy for propulsion of the systems and a high coefficient of performance (*COP*) respectively. Compression systems using the ecological refrigerant carbon dioxide (CO_2) are often run at transcritical conditions. This means that the evaporation of the refrigerant takes place at subcritical conditions, whereas the heat rejection into the environment occurs at supercritical pressures and temperatures. For such transcritical CO_2-systems with the main components compressor, gas cooler, expansion unit and evaporator, a maximum *COP* exists, depending on system, environmental conditions and operating conditions. The *COP* is influenced mainly by high pressure. A graphical method from the literature to determine the optimum high pressure is too time-consuming if several operating conditions are investigated. With a simulation model, the optimum high pressure can easily be determined for different operating conditions as a function of the refrigerant temperature at the gas cooler outlet. With these data it is possible to determine a control function. This function is used to adjust the high pressure so that the system can be run with a *COP* that deviates from the maximum values by less than 5.8 %. © Elsevier, Paris.

compression system / air-conditioning / ecological / refrigerant / carbon dioxide / transcritical / overcritical / coefficient of performance / high pressure / control

Résumé — Détermination de la haute pression optimale pour des cycles frigorifiques à CO_2 supercritiques. Les machines frigorifiques modernes doivent répondre à de nombreuses exigences, notamment environnementales. Il est donc nécessaire que ces machines opèrent avec une consommation faible de puissance et un meilleur rendement. Les machines à compression qui utilisent du dioxyde de carbone (CO_2), neutre vis-à-vis de l'environnement, sont souvent utilisées avec des cycles supercritiques. Pour de telles machines, comprenant essentiellement un compresseur, un refroidisseur de gaz, un détendeur et un évaporateur, la puissance de froid maximum, qui dépend de l'installation, du milieu environnant et de l'utilisation qui en est faite, est fortement influencée par la valeur de la haute pression. Il existe, dans la littérature, une méthode graphique pour déterminer la haute pression optimale. Cependant, cette méthode est coûteuse en temps de calcul dans le cas où différentes conditions de fonctionnement sont étudiées. À l'aide d'un modèle de simulation, il est possible d'obtenir facilement et rapidement la valeur optimale de la haute pression en fonction de la température du fluide frigorigène à la sortie du refroidisseur de gaz. Les données fournies conduisent à déterminer une fonction de régulation. Cette fonction permet d'ajuster la haute pression, de telle façon que le système fonctionne avec un *COP* qui ne s'écarte pas de plus de 5,8 % de son maximum. © Elsevier, Paris.

machines frigorifiques à compression / climatisation / environnement / fluide frigorigène / dioxyde de carbone / supercritique / coefficient de performance / haute pression / régulation

Nomenclature

A	state of refrigerant	
B	state of refrigerant	
C	state of refrigerant	
D	state of refrigerant	
E	state of refrigerant	
F	state of refrigerant	
h	specific enthalpy	$kJ \cdot kg^{-1}$
$\dot{m}$	mass flow	$kg \cdot s^{-1}$
n	compressor speed	min^{-1}
P	power	W
p	pressure	bar
Q	state of refrigerant	
$\dot{Q}$	heat flux	W

* kauffl@dbay.stg.daimlerbenz.com

R	state of refrigerant	
S	state of refrigerant	
s	specific entropy	$kJ \cdot kg^{-1} \cdot K^{-1}$
t	temperature	°C
V	volume	cm^3

Greek symbols

ε	exchanger heat transfer effectiveness
η	efficiency

Subscripts

A	air
A	state of refrigerant
amb	ambient
B	state of refrigerant
C	state of refrigerant
D	state of refrigerant
E	evaporator
GC	gas cooler
geo	geometric
H	high
i	inlet
is	isentropic
L	low
max	maximum
mech	mechanical
o	outlet
sh	shaft
t	target
0	refers to the output of the refrigeration system

1. INTRODUCTION

Modern refrigeration systems have to fulfil many requirements such as good performance, reliability, spontaneity, controllability, low size and weight, and they should be non-polluting. In compression systems pollution is mainly caused by a loss of refrigerant out of the cooling cycle and by the production of energy for propulsion of the system. Due to the fact that a loss of refrigerant in mobile systems cannot be completely avoided, especially in the area of automobile air-conditioning, an ecological refrigerant has to be used. These requirements can best be fulfilled by using carbon dioxide as refrigerant and optimized components.

To keep the energy necessary for propulsion of the refrigeration system small, the coefficient of performance (*COP*) of the system has to be high. At transcritical CO_2-refrigeration cycles, the *COP* is mainly influenced by the high pressure of the system. The paper reports first on a graphical method to determine the optimum high pressure. Because this method is too time-consuming and not very accurate, another method is introduced, that determines the optimum high pressure with help of a simulation model. Finally a control function is developed so that the system can be run with a high *COP*.

2. CARBON DIOXIDE AS REFRIGERANT

As early as in the 19th century, CO_2 (R744) was a wide-spread refrigerant. In the 1930s, however, it was displaced by the newly developed synthetical refrigerants (HCFCs) due to lower system pressures and a simpler technique. In 1974 Moulina and Rowland [1] reported for the first time on the damage of the ozone layer caused by chlorine emissions. This lead to the introduction of HFCs like R134a used currently. Although R134a has an ozone depleting potential (ODP) of zero, the global warming potential (GWP) is 1 300 times higher than that of CO_2.

2.1. The transcritical CO_2-refrigeration cycle

Nowadays the high system pressures in CO_2-refrigeration cycles can safely be controlled. Due to the critical point of CO_2 (73,8 bar and 31 °C) the refrigeration cycle is often operated transcritically, for example in automobile air-conditoning systems with high ambient temperatures. The evaporation takes place at subcritical, and the heat rejection at supercritical, pressures and temperatures.

The main components of a CO_2-compression cycle are compressor, gas cooler (instead of a condenser because of the supercritical heat rejection), expansion unit and evaporator (*figure 1*). Ideally the refrigerant is sucked as saturated vapour (A) from the compressor and compressed isentropically to a high pressure p_H (B). At the supercritical pressure the hot CO_2 is cooled in the gas cooler by transferring a heat flux $\dot{Q}_{amb}$ to the ambient air (C). Then the refrigerant is throttled to the low pressure p_L (D). In the evaporator the heat flux $\dot{Q}_0$, often called the refrigerating capacity, is removed by the cold refrigerant from the ambient air.

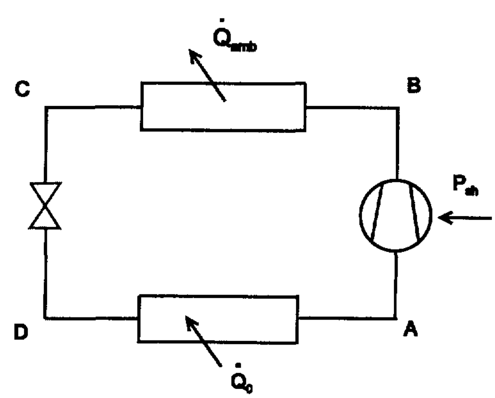

Figure 1. Simplified block diagram of the CO_2-compression cycle.

2.2. Determination of the optimum high pressure

The coefficient of performance is generally defined as:

$$COP = \frac{\text{heat flux } \dot{Q}_0}{\text{shaft power } P_{sh}}$$

In the pressure enthalpy diagram (*figure 2*) the heat flux $\dot{Q}_0$ is represented by the difference of h_A und h_D. The shaft power P_{sh} is represented by h_B and h_A in case of an ideal compression.

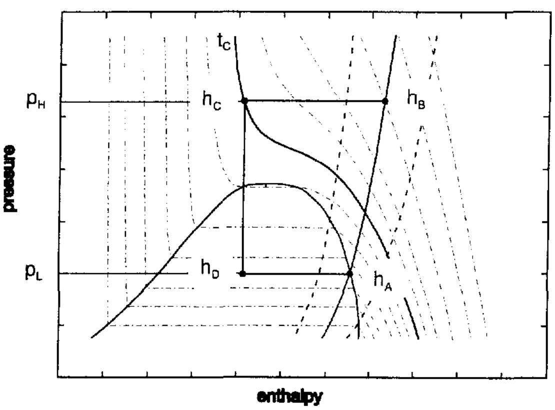

Figure 2. Pressure enthalpy diagram of carbon dioxide.

Assuming a constant low pressure p_L and assuming that the refrigerant leaves the evaporator as saturated vapour, variation of the high pressure p_H with an isentropic compression and a constant refrigerant temperature t_C at the gas cooler outlet, leads to a change in refrigerating capacity $\dot{Q}_0$, shaft power P_{sh} and coefficient of performance COP. The COP defined by:

$$COP = \frac{h_A - h_D}{h_B - h_A} = \frac{h_A - h_C}{h_B - h_A}$$

has a maximum depending on p_H. because of the S-shape of the isotherm t_c and the shape of the compression line. A rise in the high pressure p_H that is only stightly above the critical point leads to a big decrease of h_C and h_D, whereas for higher pressures the influence of p_H on h_C could nearly be neglected. But a raise in the high pressure p_H always has a nearly linear influence on h_B and consequently on the shaft power P_{sh}. To find the maximum, the derivative of the COP as a function of p_H has to be zero:

$$\frac{\partial COP}{\partial p_H} = \frac{-\left(\frac{\partial h_C}{\partial p_H}\right)_{t_C}(h_B - h_A) - \left(\frac{\partial h_B}{\partial p_H}\right)_s (h_A - h_C)}{(h_B - h_A)^2} \stackrel{!}{=} 0.$$

This is fulfilled for

$$-\frac{\left(\frac{\partial h_C}{\partial p_H}\right)_{t_C}}{(h_A - h_C)} = \frac{\left(\frac{\partial h_B}{\partial p_H}\right)_s}{(h_B - h_A)}.$$

2.2.1. Graphical determination of the optimum high pressure

Already in 1928 Inokuty [2] presented a graphical method to find the optimum pressure p_H. With the help of the tangent in points B and C (*figure 3*) he obtained $h_A = \overline{EA}$, $h_B = \overline{FB}$ and $h_C = h_D = \overline{FC} = \overline{ED}$. The COP is consequently:

$$COP = \frac{h_A - h_D}{h_B - h_A} = \frac{\overline{EA} - \overline{ED}}{\overline{FB} - \overline{EA}} = \frac{\overline{DA}}{\overline{AQ}}$$

The gradient of the isentrop in point B is approximately given by:

$$\left(\frac{\partial h_B}{\partial p_H}\right)_s = \frac{\overline{RQ}}{\overline{EF}}$$

and the gradient of the isotherm in point C by:

$$-\left(\frac{\partial h_C}{\partial p_H}\right)_{t_C} = \frac{\overline{DS}}{\overline{EF}}$$

Inserted in equation (1) this yields:

$$\frac{\overline{DS}}{\overline{DA}} = \frac{\overline{RQ}}{\overline{AQ}}$$

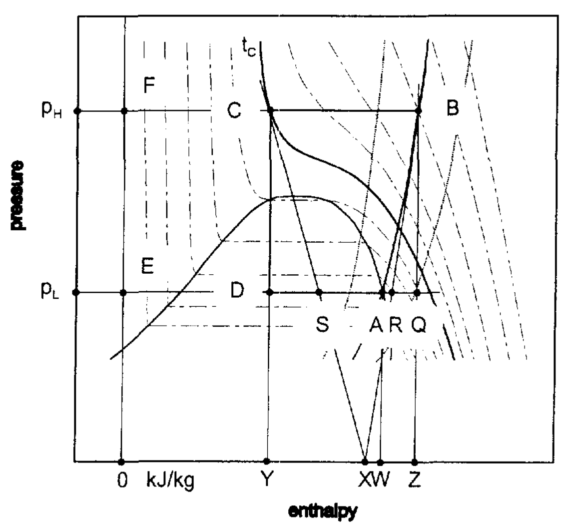

Figure 3. Graphical determination of the optimum high pressure.

The graphical solution of equation (2), that is to say the determination of the optimum high pressure, can be expedited easily by an elongation of the tangent SC and BR until they cross. The optimum is reached when this crossing point X is at the same enthalpy as point A, i.e. X equals W. In *figure 3* an increase of the actual pressure p_H leads to a dislocation of the crossing point X to the left and a decrease to a dislocation to the right. In *figure 3* the high pressure p_H is above the optimum high pressure.

With this procedure and the assumptions of isentropic compression, constant low pressure p_L and constant refrigerant temperature t_C at the gas cooler outlet, the optimum high pressure for one operating condition can be determined easily. The accuracy achievable depends on the read-out of the diagram, mainly on the determination of the gradient of the tangents, and is within 5 bar. The influence of deviations of the optimum high pressure on the cycle efficiency will be discussed at the end of the paper

If the isentropic efficiency η_{is} and the mechanical efficiency η_{mech} [3] of the compressor depending on the operating condition, also has to be taken into consideration, and the determination of the optimum high pressure has to be done for plenty of operating points as occurs in automobile A/C systems [4], then the graphical method becomes too time-consuming. In this case it is meaningful to use a steady-state simulation model for the determination of the optimum high pressure.

2.2.2. Determination of the optimum high pressure by simulation

In 1994 Pettersen and Skaugen [5] presented results of their investigations on the operation of transcritical CO_2-refrigeration cycles. They also made use of a simulation model to determine the optimum high pressure for different conditions.

In the following, not an isentropic but a real compression should be considered. For this the characteristics of a real CO_2-compressor were used in the model. A quantity of different operating conditions was measured on a test unit and a map with these data was generated. Using this the isentropic efficiency η_{is} and the mechanical efficiency η_{mech} of the compressor can be calculated, as functions of compressor speed, pressure ratio and degree of superheat at the compressor inlet. Because for automobile A/C systems a variable displacement compressor is useful, the influence of the geometric compressor displacement V_{geo} was considered too, something not been done in [5]. The influence of the superheat on the isentropic efficiency η_{is} and the mechanical efficiency η_{mech} is negligible. Also real refrigeration systems are operated with as little superheat as possible, so that the vapour may be assumed to be saturated at the compressor inlet.

For the gas cooler, a constant exchanger heat transfer effectiveness is used. In Kays and London [6], the exchanger heat transfer effectiveness of a gas cooler ε_{GC} is defined with the refrigerant temperatures t_B and t_C and the temperature of the air at the gas cooler inlet $t_{A,GC,i}$ as

$$\varepsilon_{GC} = \frac{t_B - t_C}{t_B - t_{A,GC,i}}$$

Measurements of the manufacturer of the gas cooler showed for many different operating conditions a value of 95 % in case of a sufficient air mass flow guaranteed by a blower. For the simulation $\varepsilon = 95$ % and an air mass flow at the gas cooler $\dot{m}_{A,GC} = 0{,}2$ kg·s^{-1} is used. This mass flow is used for most automobile operating conditions. With these assumptions the refrigerant temperature t_C at the gas cooler outlet is approximately 2,9 °C higher than the ambient temperature t_{amb}. The expansion is isenthalp. The exchanger heat transfer effectiveness of the evaporator ε_E according to Kays and London is defined with the temperature of the air at the evaporator inlet $t_{A,E,i}$ and outlet $t_{A,E,o}$ and the refrigerant temperature t_D as

$$\varepsilon_E = \frac{t_{A,E,i} - t_{A,E,o}}{t_{A,E,i} - t_D}.$$

The value for the exchanger heat transfer effectiveness of the evaporator ε_E varies for automobile operating conditions between 80 and 90 % according to Bhatti [7]. The average of 85 % is used. For the cool-down phase in an automobile, the air mass flow at the evaporator is $\dot{m}_{A,E} = 0.125$ kg·s^{-1}. We also use this value for the simulation. Because the influence of the moisture on the heat transfer rate is very complicated to model, it will be disregarded. This means a value of 0 % is used. Because the pressure drop at the refrigerant side has a big influence on the *COP*, it is taken into consideration within the gas cooler and the evaporator. For this, correlations are used that were derived from measurements from the manufacturer of the heat exchangers.

For control of the optimum pressure p_H the refrigerant temperature at the gas cooler outlet or the ambient temperature may be used. We need therefore a function

$$p_H = f(t_C) = f(t_{amb} + 2.9)$$

where the temperatures are in °C and the pressure results in bar.

In *figure 4* and *5* the *COP* is shown as function of pressure p_H for a compressor revolution of 800 and 1 800 min^{-1} respectively. The geometric compressor displacement V_{geo} and the low pressure p_L are constant. As can be seen from *figure 4*, the *COP* is for all ambient temperatures a little higher than in *figure 5* for the higher revolution of 1 800 min^{-1}, because of a slightly better compressor efficiency at a lower revolution speed. Although the values of the *COP* are mainly influenced by the ambient temperature, the maximum *COP* for a certain ambient temperature is reached at the same pressure p_H independent of the compressor revolution.

To determine the influence of the geometric compressor displacement V_{geo}, it is increased to 18.3 cm^3 in

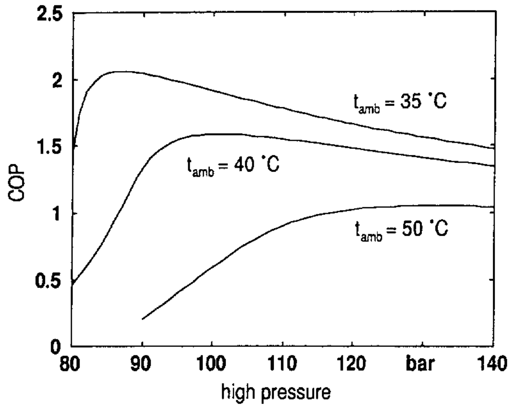

Figure 4. Influence of high pressure on *COP* varying the ambient temperature t_{amb} for $V_{\mathrm{geo}} = 4.6$ cm^3, $n = 800$ min^{-1} and $p_{\mathrm{L}} = 40$ bar.

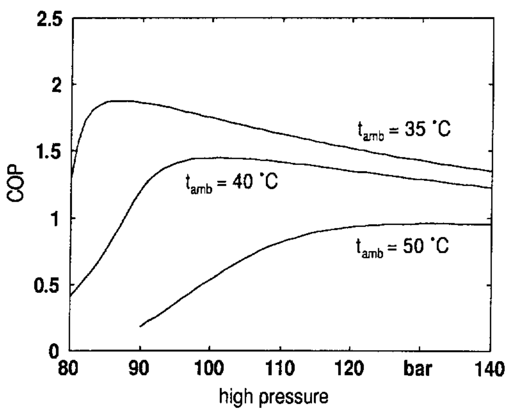

Figure 5. Influence of high pressure on *COP* varying the ambient temperature t_{amb} for $V_{\mathrm{geo}} = 4.6$ cm^3, $n = 1\,800$ min^{-1} and $p_{\mathrm{L}} = 40$ bar.

figure 6 and *figure 7* compared to 4.6 cm^3 in *figure 4* and *figure 5*. The compressor revolutions of 800 and 1 800 min^{-1} respectively shown. Also with the bigger geometric compressor displacement V_{geo}, the position of the maximum *COP* merely depends on the ambient temperature but occurs almost at the same pressure. Apart from the influence of the operating conditions of the compressor on the optimum high pressure, the influence of the low pressure p_{L} on the operating conditions should be considered as well. In *figure 4* to *figure 7* the low pressure was kept constant at 40 bar. In this case a pronounced maximum of the *COP* for low ambient temperatures ($t_{\mathrm{amb}} = 35$ °C) is obtained. *Figure 8* presents results for a constant ambient temperature of 35 °C and different low pressures p_{L}. Although it is noteworthy that the low pressure has a significant influence on the maximum *COP*, the maximum value is nearly at the same pressure p_{H}.

The determination of the optimum high pressure at various operating conditions with the application of a

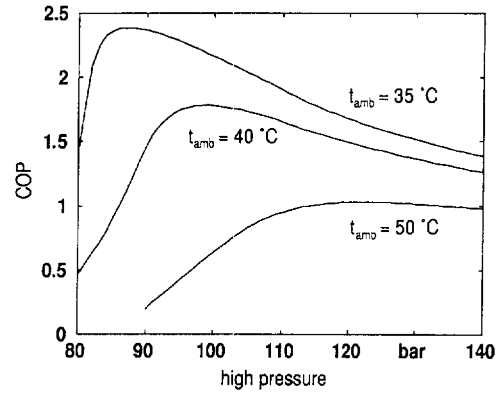

Figure 6. Influence of high pressure on *COP* varying the ambient temperature t_{amb} for $V_{\mathrm{geo}} = 18{,}3$ cm^3, $n = 800$ min^{-1} and $p_{\mathrm{L}} = 40$ bar.

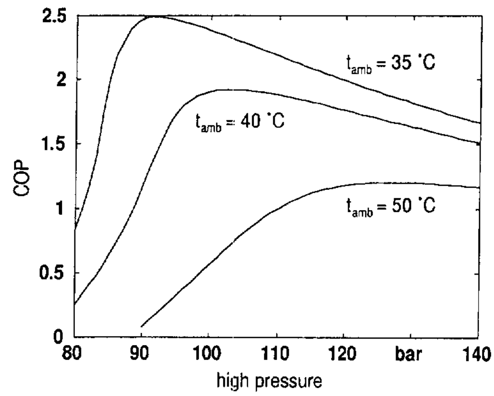

Figure 7. Influence of high pressure on *COP* varying the ambient temperature t_{amb} for $V_{\mathrm{geo}} = 18.3$ cm^3, $n = 1\,800$ min^{-1} and $p_{\mathrm{L}} = 40$ bar.

steady-state simulation model as shown could easily be automat and is therefore superior to the graphical method.

By means of these results it is obvious that the main influence on the optimum high pressure is the ambient temperature t_{amb} and the temperature of the refrigerant t_{C} at the gas cooler outlet respectively. In this may it is possible by measuring t_{amb} or t_{C} and with the help of the control function equation (3) sought by controlling the high pressure p_{H}, to operate the refrigeration cycle at the optimum high pressure.

2.2.3. Control function

To obtain the control function sought in equation (3), the simulated optimum high pressures under the operating conditions investigated and at the different ambient temperatures are presented in *figure 9*. From these results, a relation of the optimum target pressure $p_{\mathrm{t,H}}$ and the ambient temperature or refrigerant temperature at the gas cooler outlet can be established.

Figure 9 shows the minimum and maximum values of the optimum high pressure for different ambient temperatures. Because of the strong influence of the high pressure on the *COP* for low ambient temperatures, particularly at pressures below the optimum, the chosen target pressure for $t_{amb} = 35$ °C is 91 bar. A maximum allowable high pressure $p_{H,max} = 130$ bar is obtained for $t_{amb} = 50$ °C. It is not exceeded even at higher ambient temperatures. The simplest correlation for this relation is a linear function.

$$p_{t,H} = 2.6\ t_{amb} = 2{,}6\ t_C + 7.54$$

with temperatures in °C and $p_{t,H}$ in bar. This equation is valid for an ambient temperature range between 35 and 50 °C.

Although the determination of the optimum high pressure with the simulation model leads to exact values, deviations occur because of the estimated control function. With the target pressure $p_{t,H}$ from equation (4), the biggest deviations from the maximum values of the *COP* result for low ambient temperatures but are below 5.8 %.

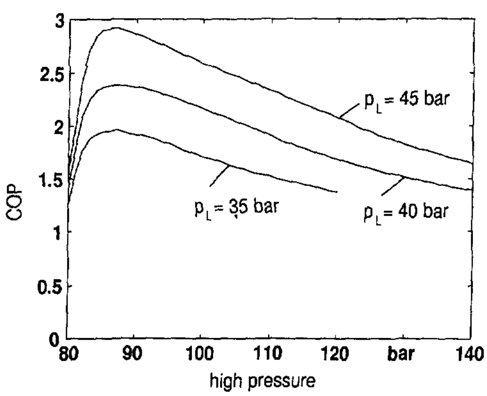

Figure 8. Influence of high pressure on *COP* varying the low pressure p_L for $V_{geo} = 18{,}3$ cm^3, $n = 1\,800$ U·min^{-1} and $t_{amb} = 35$ °C.

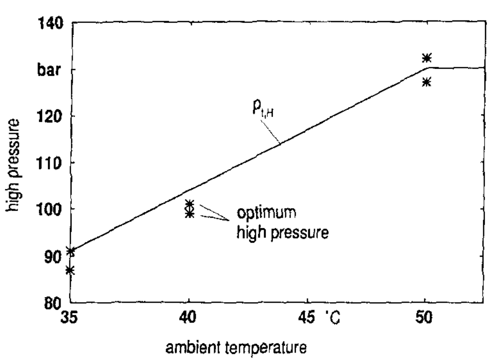

Figure 9. Simulated optimum high pressure (*) and target high pressure $p_{t,H}$ as functions of the ambient temperature.

However the graphical method also leads to inexact values of the optimum high pressure. If the determined optimum high pressure is e.g. about 5 bar below the correct value at low ambient temperatures, then the achieved *COP* is about 20 % below the maximum.

3. CONCLUSION

The determination of the optimum high pressure of a transcritical CO_2-refrigeration cycle can be done with a graphical method. If various operating conditions have to be considered, the use of a control function is helpful. The application of a steady-state simulation model as presented here leads to a simple control equation for the optimum high pressure of a transcritical CO_2-refrigeration cycle, provided that information for heat exchangers and compressors, as given in the example of this paper, is available. Because the optimum high pressure depends not only on the ambient temperature, the use of an approximated control function leads to *COPs* that are slightly below the maximum.

It should also be mentioned that for other heat exchanger rates or another compressor, the coefficients in equation (4) become different. A comparison with the results presented in [5] shows that the optimum high pressures, especially for idling conditions, are much lower than the values presented here. The use of a compressor with higher efficiencies for example raises the optimum high pressure. This shows that if another heat exchanger or compressor is used, the simulation has to be repeated and a new control function has to be determined; however this can easily be automated.

REFERENCES

[1] Moulina M.J., Rowland F.S., Stratospheric sink for chlorofluormethanes: chlorine atom-catalyzed destruction of ozone, Nature 249, 1974.

[2] Inokuty H., Zeichnerisches Verfahren zum Auffinden des günstigsten Kondensatordruckes in Kohlensäurekältemaschinen, Zeitschrift für die gesamte Kälte-Industrie 9 (35) (1928) 180–182.

[3] ISO 1217, Displacement compressors. Acceptance tests, ISO 1217:1996(E).

[4] Wertenbach J., Kauf F., High pressure refrigeration system with CO_2 in automobile air-conditioning, IIR Workshop, Trondheim, Norwegen, 13–14 May 1997.

[5] Pettersen J., Skaugen G., Operation of trans-critical CO_2 vapour compression circuits in vehicle air conditioning, in: New applications of natural working fluids in refrigeration and air conditioning, Hannover, 10–13 May 1994.

[6] Kays W.M., London A.L., Compact heat exchangers, Third edition, McGraw-Hill, 1984 pp. 11–78.

[7] Bhatti M.S., A critical look at R-744 and R-134a mobile air conditioning systems, SAE 970527, 1997.