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# ENERGY EFFICIENCY IMPROVEMENT OF THE REFRIGERATION CYCLE USING AN INTERNAL HEAT EXCHANGER

# POPRAWA EFEKTYWNOŚCI ENERGETYCZNEJ OBIEGU ZIĘBNICZEGO POPRZEZ ZASTOSOWANIE DOZIĘBIACZA

Abstract

A refrigeration cycle modified using an Internal Heat Exchanger is presented. For different refrigerants, the net effect of using an IHX is either positive or negative due to refrigerant properties and working conditions. Wet refrigerant vapor at the inlet of IHX improves the cycle in various aspects. The evaporator performance is much better when not superheating the vapor. A "pure" effect of subcooling the refrigerant liquid at the inlet of the expansion valve results in an increased specific heat of evaporation. For zeotropic mixtures, increased subcooling results in lowering the evaporating temperature without lowering the pressure in the evaporator. The EER value can be improved for some refrigerants and for specific working conditions. Theoretical and experimental evaluations of this concept are presented. Commonly used refrigerants were evaluated theoretically and tested in a 10 kW (cooling capacity) test rig. R22 and R407C were analyzed.

*Keywords: efficiency, refrigeration cycle, internal heat exchanger, refrigerant mixtures*

Streszczenie

W artykule przedstawiono obieg ziębniczy zmodyfikowany przez dodanie doziębiacza. Efekt końcowy zastosowania doziębiacza jest różny w zależności od użytych ziębników oraz warunków pracy. Wprowadzenie pary mokrej do doziębiacza skutkuje wieloma pozytywnymi efektami. Praca parowacza jest efektywniejsza, jeżeli nie służy on do przegrzewania pary ziębnika, doziębienie cieczy przed zaworem rozprężnym skutkuje zwiększeniem właściwego ciepła parowania. Dla mieszanin zeotropowych obserwuje się obniżenie temperatury parowania bez obniżania ciśnienia parowania. Zwiększenie wartości wskaźnika EER może być uzyskane przez odpowiedni dobór ziębników oraz parametrów pracy obiegu. Przedstawiono również wyniki badań eksperymentalnych dla powszechnie używanych ziębników na stanowisku o wydajności ziębniczej 10 kW.

*Słowa kluczowe: twardość wody, renowacja, wykładzina cementowa, przewody wodociągowe*

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## Nomenclature

$c_p$ – specific heat capacity at constant pressure [kJ/kgK]
$c_v$ – specific heat capacity at constant volume [kJ/kgK]
EER – Energy Efficiency of Refrigeration
$h_{rp}$ – specific heat of evaporation [kJ/kg]
$q_o$ – specific enthalpy of evaporation [kJ/kg]
U – heat transfer coefficient [W/m$^2$K]
$l_t$ – specific compressor work [kJ/kg]
$m$ – mass flow rate [kg/s]
$p$ – pressure [MPa]
$P$ – power [kW]
$t$ – temperature [°C]
$T$ – temperature [K]
κ – isentropic coefficient = cp/cv
υ – specific volume
$x$ – vapor quality

Subscripts
$k$ – condenser
$o$ – evaporator
$c$ – liquid
$p$ – vapor
$av$ – average
$co$ – subcooling

## 1. Introduction

In literature, many authors presented the application of Internal Heat Exchanger as a way of improving the efficiency of refrigeration systems. Kruse [1] suggests the use of IHX in domestic refrigerators using flammable refrigerants, and an extensive set of refrigerants was discussed by Domanski et al. [2]. Angelino and Invernizzi [3] make a theoretical assessment of organic refrigerants. They define the "molecular complexity factor" dependent on the vapor specific heat capacity and prove that, for fluids with complex molecular particles, IHX may be necessary to obtain proper energy efficiency of the heat pump. McLinden [4] presents an analysis of refrigerating cycles with IHX using a semi-theoretical model and concludes that fluids with high vapor heat capacity may achieve high efficiency. Domanski et al. [5] presents a simulation of pure refrigerants in cycles with an IHX, and Jung et al. [6] give a theoretical analysis of pure refrigerants with an IHX suggesting that some of the fluids can be used as mixture components. Huelle [7] presents an analysis of the IHX impact on the energy demand for refrigeration machinery, and Vakil [8] presents a general analysis of applying fluid subcooling by vapor superheating for pure and mixed zeotropic refrigerants. Lavrenchenko et al. [9] compare refrigerant cycles with mixtures. Cycles that are modified by subcooling in the condenser, superheating in the evaporator, IHX with wet and superheated vapor and energy efficiencies, are compared. Klein et al. [10] perform a theoretical analysis of IHX for new refrigerants and investigate the pressure drop impact on the exchanger performance.

In every application, a big disadvantage is noted: the excessive superheating of the vapor entering the compressor that leads to extra compressor work and reduces energy efficiency. The reason is that isentropes, which represent the compression process, increase the slope with superheating of the vapor.

## 2. Internal heat exchanger

In a typical refrigerating cycle consisting of four basic elements (evaporator, compressor, condenser and expansion valve – Fig. 1a), one of the disadvantages is the isenthalpic expansion process during which the refrigerant partially evaporates (3–4). Thus, the phase-change process in the evaporator suffers due to increased vapor quality at the evaporator inlet. Increased refrigerant subcooling at the condenser outlet "moves" state (4) of the refrigerant to the left (4′) – Fig. 2, increasing the specific heat of evaporation. Refrigerant subcooling is also very important for the proper operation of some of the refrigerating or air-conditioning machinery. In some installations, the refrigerant flows from the condenser to an expansion valve over a long distance, which can cause partial evaporation due to the pressure drop when the refrigerant leaves the condenser in a saturated state.

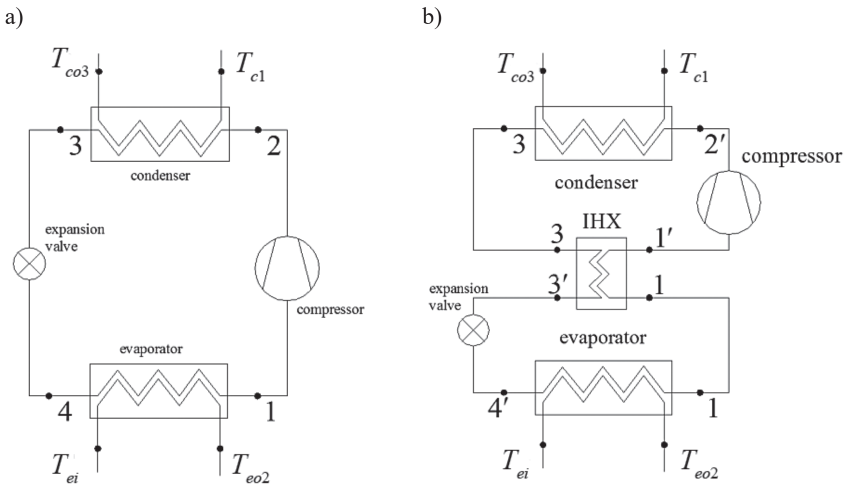


Fig. 1. Refrigerant cycle: a) without an IHX, b) with an IHX

There are several ways to obtain liquid subcooling. Some of them are: increasing the heat exchange area of condenser, increasing condensing pressure, additional external heat exchangers, mechanical subcooling (refrigeration cycle with evaporator-subcooler heat exchanger) etc.

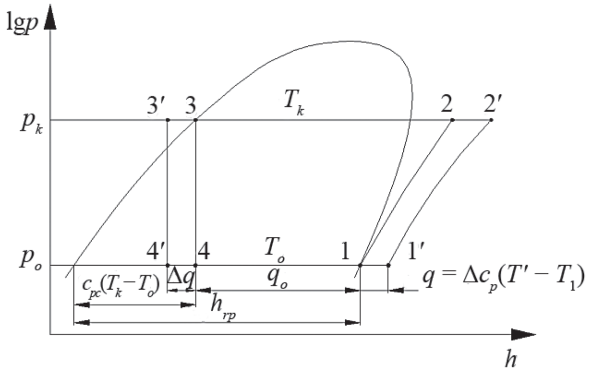


Fig. 2. Refrigerant state points and values used in calculations (Domanski et al. [2])

One of the solutions to the problem is a very the well-known internal heat exchanger (IHX), which subcools liquid refrigerant using cold vapor at the outlet of the evaporator (Fig. 1b). The heat transferred from liquid to the vapor causes superheating of the vapor entering the compressor and prevents the refrigerant liquid droplets from entering the compressor. Extensive vapor superheating is one of the great disadvantages of such a solution. For almost every refrigerant, compressor work increases with increased vapor superheat at the entrance to the compressor. This is caused by the slope of isentropes, which increases with the vapor superheating. Energy efficiency of the cycle, calculated as:

$$\mathrm{EER} = \frac{q_o}{l_t} \tag{1}$$

can increase or decrease, depending on the refrigerant used in the cycle. The heat exchange in the IHX reduces the irreversibility in the expansion process. Energy efficiency will increase only when the proportional increase of specific latent heat in the evaporator is larger than the corresponding specific work increase of the compressor:

$$\frac{\Delta q}{q_o} > \frac{\Delta l_t}{l_t} \tag{2}$$

Domanski et al. [2] compared the performance of cycles with and without the IHX using the same corresponding saturation temperatures in the evaporator and the condenser:

$$\mathrm{EER}' = \frac{q_o + \Delta q}{l_t + \Delta l_t} = \mathrm{EER}\frac{1 + \Delta q / q_o}{1 + \Delta l_t / l_t} \approx \mathrm{EER}\left(1 + \frac{\Delta q}{q_o} - \frac{\Delta l_t}{l_t}\right) \tag{3}$$

The formula in brackets must be larger than 1 if the system is to benefit from using the IHX. For obvious reasons, $\Delta q/q_o$ is always greater than 0 and $\Delta l_t/l_t$ is also positive. After some simplification, the specific enthalpy increase in the evaporator can be described as:

$$q_o = h_{rp} - \overline{c}_{p,c}(t_k - t_o)$$

where:

$$\overline{c}_{p,c} = \frac{1}{t_k - t_o}\int_{t_o}^{t_k} c_{p,c}\,dt$$

Enthalpy change in the evaporator $\Delta q$ is equal to the amount of heat exchanged between high-pressure liquid and low pressure vapor:

$$\Delta q = \overline{c}_{p,p}(t_{1'} - t_1) \tag{4}$$

where:

$$\overline{c}_{p,p} = \frac{1}{t_{1'} - t_1}\int_{t_1}^{t_{1'}} c_{p,p}\,dt \tag{5}$$

Compressor specific work can be described by an equation of isentropic work of the ideal gas at constant heat capacity:

$$l_t = \frac{\kappa}{\kappa - 1} P_1 \upsilon_1 \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{\kappa-1}{\kappa}}\right] \tag{6}$$

and

$$l_t' = \frac{\kappa}{\kappa - 1} P_1 \upsilon_{1'} \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{\kappa-1}{\kappa}}\right] \tag{7}$$

From the above equations, one can derive:

$$\begin{aligned} \frac{\mathrm{EER}'}{\mathrm{EER}} &= \frac{1 + [\overline{c}_{p,p}(t_{1'} - t_1)]/[h_{r,p} - \overline{c}_{p,c}(t_k - t_o)]}{1 + (\upsilon_{1'} - \upsilon_1)/\upsilon_1} = \\ &= \frac{1 + (t_{1'} - t_1)/[h_{r,p}/\overline{c}_{p,p} - (t_k - t_o)\overline{c}_{p,c}/\overline{c}_{p,p}]}{1 + B_p(t_{1'} - t_1)} \end{aligned} \tag{8}$$

The above ratio will be greater than 1 when:

$$\frac{1}{h_{r,p}/\overline{c}_{p,p}-(t_k-t_o)\overline{c}_{p,c}/\overline{c}_{p,p}}>B_p \tag{9}$$

where:

$$B_p=\frac{\upsilon_{1'}-\upsilon_1}{\upsilon_1(t_{1'}-t_1)} \tag{10}$$

is the average coefficient of thermal expansion (Domanski et al. [2]) Equation (9) shows that EER improvement will occur only if

h_(r, p)/c¯_(p, p) and B decrease and (t_k-t_o) c¯_(p, c)/c¯_(p, p) increases

The ratio between vapor and liquid heat capacity plays a significant role when there are large temperature differences between evaporation and condensation. Equations (9) and (10) can be presented in a form of function:

$$F(t_o,t_k,t_{1'}p_o(t_o))=\frac{1}{\dfrac{h_p(p_o)}{\overline{c}_{pp}(t_o,t_{1'})}-(t_k-t_o)\cdot\dfrac{\overline{c}_{pc}(t_k,t_o)}{\overline{c}_{pp}(t_o,t_{1'})}}-\frac{\upsilon_{1'}(t_o,t_{1'})-\upsilon_1(t_o)}{\upsilon_1(t_o)\cdot(t_{1'}-t_o)}>0 \tag{11}$$

The calculations were made using Refprop (Gallager et al. [11])

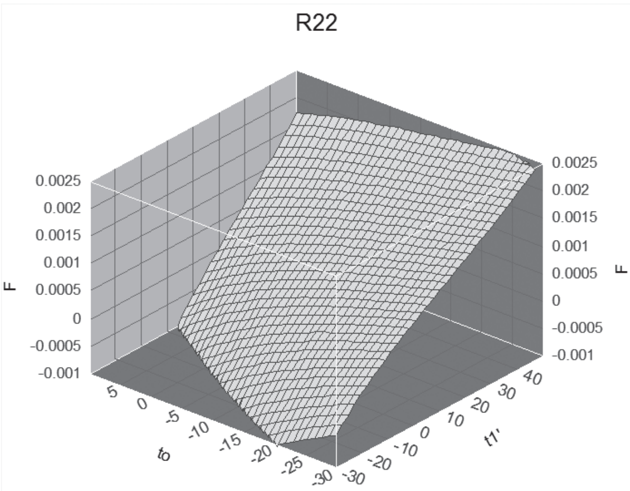


Fig. 3. Function $F(t_o, t_{1'})$ values for R22

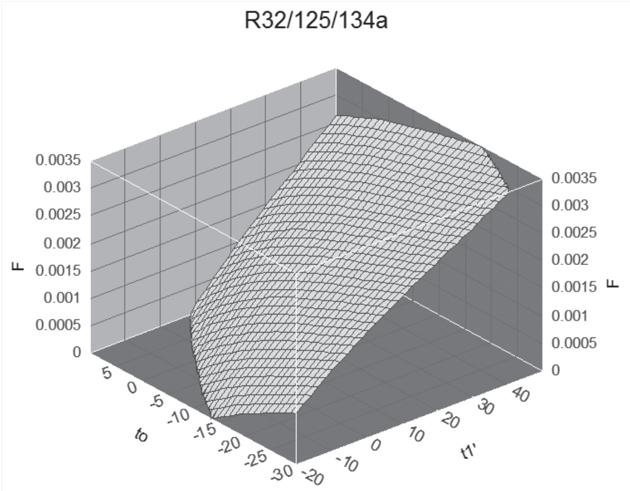


Fig. 4. Function $F(t_o, t_{1'})$ values for R407C

Figures 3 and 4 present the values of function $F(t_o, t_{1'})$ for two refrigerants. The refrigerants selected for tests were evaluated theoretically and experimentally, and presented by Maczek et al. [12, 13].

As one can notice, the value of function $F$ for the mixture is greater than “0”, which means that for such conditions of evaporation temperature to and superheating temperature $t_{1'}$ using the IHX is of interest.

## 3. Wet vapor entering IHX

As mentioned above, the use of the IHX has one disadvantage: excessive superheating of the vapor entering the compressor. If we analyze wet vapor entering the IHX (point 1 in Fig. 2 moved to the left) and saturated vapor entering the compressor (point 1′ on Fig. 2 on the saturated line), we obtain different situation with all advantages of the IHX without superheating of the vapor.

For zeotropic mixtures (Fig. 5), the process of subcooling the liquid using wet vapor in the IHX causes several differences in comparison to the cycle with no IHX and with the IHX using superheated vapor.

The evaporation process is shifted to the left from 4-1 to 4′-1′, where lower temperatures of the evaporator inlet can be noted. Obtaining lower temperatures for pure refrigerants can be achieved only by lowering the evaporating pressure, which is always energy inefficient. Fig. 6 shows the calculation results of the IHX impact on the average evaporation temperature for different evaporation pressures for R407C.

Experimental data show a slight improvement of the energy efficiency of the refrigeration cycle. The change of the cycle was obtained by moving the remote sensor bulb of the thermostatic expansion valve (which is usually set for slight superheating of the vapor) onto the pipe downstream the IHX.

Fig. 7 Shows some of the results of the test for evaporation temperature $p_o$ = 0.58 MPa with wet vapor at the IHX inlet.

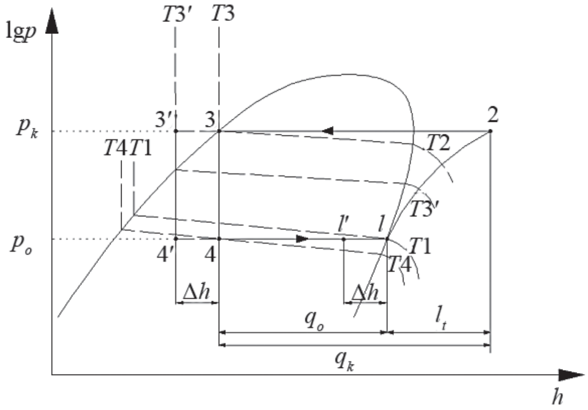


Fig. 5. Refrigerant cycle for zeotropic mixtures with IHX

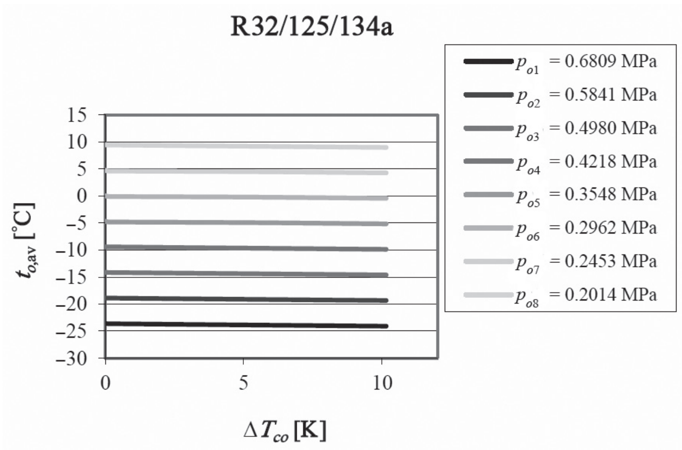


Fig. 6. Average evaporation temperature as a function of refrigerant subcooling at the IHX exit ($p_{o1}$÷$p_{o8}$ – evaporation pressure)

Also, a relative change of EERrel [%] was calculated from the experimental data as:

$$\mathrm{EER}_{\mathrm{rel}} = 100 \cdot (\mathrm{EER}' - \mathrm{EER})/\mathrm{EER}$$

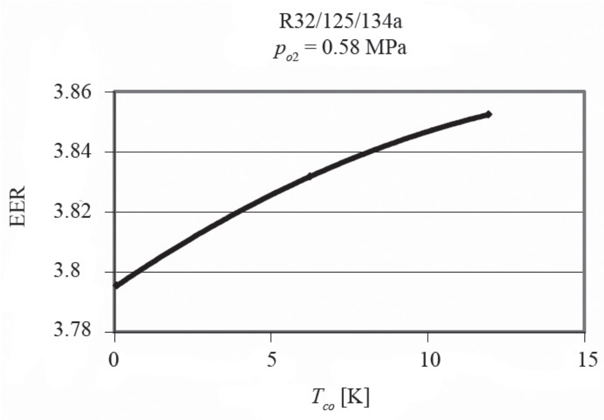


Fig. 7. Energy efficiency as a function of refrigerant subcooling at the IHX exit

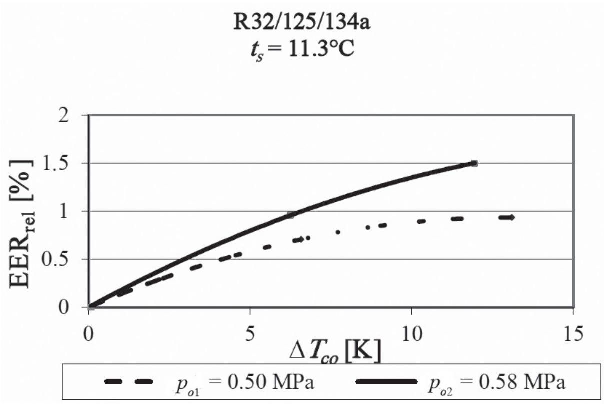


Fig. 8. Relative EER change as a function of refrigerant subcooling at the IHX exit for the same compressor suction temperature $t_s$ = 11.3°C

## 4. Conclusions

The internal heat exchanger can be effectively applied in the refrigerant cycle using zeotropic refrigerant mixtures. The calculations made for R22 and R407C show that IHX is always improving the EER for the cycle with the refrigerant mixture, while there is no improvement in EER in the cycle using R22 with the low evaporation temperature. The experimental tests show a slight EER improvement for R407C with the use of wet vapor in the IHX.

Moreover, the performance of the evaporator is better when there is no superheating of the vapor. The end part of the heat exchange area can be more effectively used. A slightly superheated vapor at the inlet of the compressor is accepted, otherwise a too high discharge temperature is observed, together with an increase of the specific work of the compressor. The positive effect of refrigerant subcooling at the inlet of the expansion valve is the increased specific heat of the evaporation. Subcooling the liquid leaving the condenser results in lowering the average evaporating temperature without lowering the pressure in the evaporator – for zeotropic mixtures with temperature glide during the phase change.

## References


[1] Kruse H., *Energy savings when using hydrocarbon as refrigerant*, International CFC and Halon Alternatives Conference, Washington, DC, USA, 1995.

[2] Domanski P.A. et al., *Evaluation of suction-line/liquid-line heat exchange in the refrigeration cycle*, Int. J. Refrig., Vol. 17, 1994.

[3] Angelino G., Invernizzi C., *General method for the thermodynamic evaluation of heat pump working fluids*, Int. J. Refrig., Vol. 11, 1988.

[4] McLinden M.O., *Optimum refrigerants for non-ideal cycles: an analysis employing corresponding states*, Proc. ASHRAE – Purdue CFC & IIR – Purdue Refrigeration Conferences, W Lafayette, IN, 1990.

[5] Domanski P.A., Didion D.A, Mulroy W.J., Parise J., *A simulation model and study of hydrocarbon refrigerants for residential heat pump systems*, Proceedings Int. Conference New Applications of Natural Working Fluids in Refrigeration and Air Conditioning, IIR, Commission B2, Hannover, Germany 1994.

[6] Jung D., Kim Ch., Hwangbo H. Ji. H., *Effect of suction line heat exchangers on the performance of various HCFC22 alternatives*, International Conference on Ozone Protection Technologies, Washington, DC (USA), 1996.

[7] Huelle Z.R., *Wpływ ukształtowania obiegu chłodniczego na zapotrzebowanie energii urządzeń chłodniczych* (The refrigerant cycle configuration impact on the energy consumption), Konferencja Naukowo-Techniczna: Współczesne Problemy Techniki Chłodniczej, Kraków 1999.

[8] Vakil B.H., *Thermodynamics of heat exchange in refrigeration cycles with non-azeotropic mixtures. Part II, Suctionline heat exchange and evaporative cooling of capillary*, Proceedings XVI Int. Congress of Refrigeration Paris, France, International Institute of Refrigeration, Commission B1, 1983.

[9] Lavrenchenko G., Khmelnuk M., Tikhonova E., *Thermodynamical aspects of using mixtures of substances in refrigerating machines*, CFC's The Day After. Refr. Sci. and Tech. Proceedings Commissions B1, B2, E1, E2, Padova 1994.

[10] Klein S.A., Reindtl D.T., Brownell K., *Refrigeration system performance using liquid-suction heat exchangers*, Int. J. Refrig., 23, 2000.

[11] Gallager J., McLinden M., Morrison G., Huber M., *NIST Thermodynamic Properties of Refrigerants and Refrigerant Mixtures Database. (REFPROP 8.0)*, National Institute of Standards and Technology, Gaithersburg, MD, 2007.

[12] Maczek K., Müller J., Wojtas K., Domański P., *Ternary zeotropic mixture with* $CO_2$ *component for R-22 heat pump application. Clima 2000/Brussels 1997a*, World Congress.

[13] Maczek K., Wojtas K., Müller J., *Ternary mixture as R22 replacement in heat pumps*, IIF-IIR – Commissions E2 with E1 and B2, Linz, Austria 1997b.