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Thermodynamic optimal design of heat exchangers for an irreversible refrigerator

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Abstract — Thermodynamic optimisation of energy systems is essential in reducing the environmental impact of energy utilisation. Yet, the refrigerators commonly used for this purpose have improvable efficiency levels. Their performance, as shown by the literature, is highly influenced by the size of the heat exchangers and by internal irreversibilities. In this paper the maximum coefficient of performance (*COP*) is obtained for an irreversible inverse Rankine cycle refrigerator working with the environmentally friendly fluid R134a. This is a steady-state refrigerator working as an open system which consumes external work, subtracts heat from a cold fluid stream at an inlet fixed temperature and assigns it to a higher fixed inlet temperature stream. Heat transfer irreversibilities in the shell-and-tube heat exchangers and external friction losses in the water streams are considered, ignoring only the internal pressure drop of vapor. A simulation program was developed to search the maximum *COP* at given external fluid temperatures, as a function of mass flows, dimensions and temperature differences in the heat exchangers. Owing to the large number of control variables involved, a numerical optimisation method was used to determine the maximum *COP*. The proposed method is fast, producing the maximum with acceptable approximation. It provides the refrigerating fluid evaporating and condensing pressures, the heat exchanger dimensions, and the water flow rates for a given cooling power with predefined inlet temperatures of cold and hot water streams. The heat exchanger area closely conditions the *COP*, so each maximum represents the optimum thermodynamic working conditions for a given area of the heat exchangers. © 2001 Éditions scientifiques et médicales Elsevier SAS

irreversible cycle / refrigerator / optimisation / heat exchange irreversibilities / heat exchanger design / open system / steady-state

Nomenclature

A	heat exchanger area	m^2
At	total heat exchanger area	m^2
C	capacity rate	$W \cdot K^{-1}$
c_p	specific heat	$J \cdot kg^{-1} \cdot K^{-1}$
COP	coefficient of performance	
D	diameter	m
f	friction factor	
g	gravity factor	$m \cdot s^{-2}$
h	specific enthalpy	$J \cdot kg^{-1}$
	or heat exchanger coefficient	$W \cdot m^{-2} \cdot K^{-1}$
K	heat exchange over temperature difference	$W \cdot K^{-1}$
k	thermal conductivity	$W \cdot m^{-1} \cdot K^{-1}$
L	length	m
$\dot{m}$	mass flow	$kg \cdot s^{-1}$
Nu	Nusselt number	
n	number of tubes	
Pr	Prandtl number	
p	pressure	Pa
Q	thermal power	W
Re	Reynolds number	
r	latent heat	$J \cdot kg^{-1}$
s	specific entropy	$J \cdot K^{-1} \cdot kg^{-1}$
T	temperature	K
U	global heat exchange coefficient	$W \cdot m^{-2} \cdot K^{-1}$
v	specific volume	
W	power	W

Greek symbols

β	$= K_h/K_c$	
ΔT_{ml}	logarithmic mean temperature difference $= [\Delta T_i - \Delta T_o]/\ln(\Delta T_i/\Delta T_o)$	
ε	roughness	m
	or Kays' efficiency	
η	$= COP/COP_C$	
μ	dynamic viscosity	$N \cdot s \cdot m^{-2}$

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ρ	density	$\mathrm{kg{\cdot}m^{-3}}$

Subscripts

C	Carnot
c	relative to lower T
d	relative to condensation
e	boiling
f	refrigerator
h	relative to higher T
i	inlet
j	general
l	liquid
max	related to a maximum
n	net
o	outlet
over	overheating
r	rank
sat	saturation
sub	subcooling
v	vapor
w	wall

1. INTRODUCTION

The literature contains numerous examples of irreversible thermodynamic analyses of refrigerators with endoreversible cycles [1, 2] as well as special methods for examining irreversible cycles [3–5]. The cycle is often examined with the internal transformation included [6–10]. However, the heat exchangers are often considered in a simplified fashion, assuming constant parameters. Chua et al. [11] criticise the models proposed by the so called “finite time thermodynamic” because they hardly correspond to a real solution for commercial machines. Then they propose a new model [12] using Kays’ efficiency for the heat exchangers. Papers and models rarely consider the optimum search, considering the thermal parameters and the heat exchanger friction losses. Among the numerical studies aiming at system optimisation, it is not clearly shown how the components are modelled from the phenomenological standpoint [13, 14].

In this paper, a simulation program was developed to search the maximum *COP* for a reverse cycle with internal and external irreversibilities, with the main goal of studying the influence of heat exchanger design. A simple cycle was considered with an environmentally friendly working fluid, R134a. Shell and tube heat exchangers were chosen in this initial study, to consider constant pressure heat exchange for R134a and a simple correlation was used to simulate the compressor. A numerical method was used to determine the optimum value for 8 different parameters giving the maximum *COP*.

2. STEADY-STATE BALANCES

Let us consider a steady-state refrigerator working between two fluids as an open system. The system, which consumes external power, subtracts heat from a cold fluid stream at inlet temperature T_{ci}, and assigns the heat to a higher temperature stream with inlet temperature T_{hi} (*figure 1*).

We shall ignore the kinetic energy variations and the gravitational potential. When p_{o} is equal to p_{i} the first law states:

$$\sum_j W_j = \dot{m}_{\mathrm{c}} \Delta h_{\mathrm{c}} + \dot{m}_{\mathrm{h}} \Delta h_{\mathrm{h}} \tag{1}$$

This can be modified for flowing fluids, assuming ideal liquids or gas:

$$\sum_j W_j = Q_{\mathrm{c}} - Q_{\mathrm{h}} \tag{2}$$

Considering the subsystem where the refrigerating fluid runs the cycle, we can write:

$$W_{\mathrm{f}} = -Q_{\mathrm{h}} + Q_{\mathrm{c}} \tag{3}$$

where Q_{h} and Q_{c} represent the heat being exchanged. At the same time, power is required in the heat exchangers

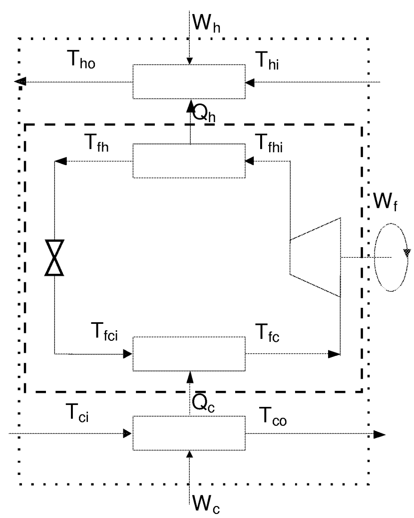

Figure 1. Scheme of the open system considered for equation (1).

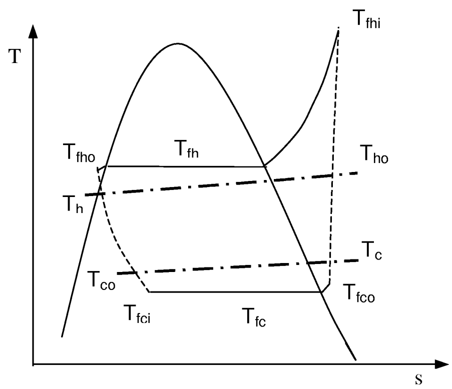

Figure 2. T–s diagram of the irreversible cycle and scheme of the temperature source changes.

to move the fluids against the friction forces. This power can be calculated by

$$\begin{aligned} W_\mathrm{h} &= \dot{m}_\mathrm{h} v_\mathrm{h} \Delta p_\mathrm{h} \\ W_\mathrm{c} &= \dot{m}_\mathrm{c} v_\mathrm{c} \Delta p_\mathrm{c} \end{aligned} \tag{4}$$

where Δp_h and Δp_c represent the pressure losses and always have a negative sign.

The *COP*, defined as the ratio between the net energy taken from the cold fluid and the total work required, gives the effectiveness of the system. Expressing the net energy exchanged by the cooled fluid as

$$Q_\mathrm{cn} = -Q_\mathrm{c} - W_\mathrm{c} = -Q_\mathrm{c} - \dot{m} v_\mathrm{c} \Delta p_\mathrm{c} \tag{5}$$

we obtain

$$\begin{aligned} COP &= \frac{Q_\mathrm{cn}}{\sum_j W_j} \\ &= \frac{-Q_\mathrm{c} - \dot{m}_\mathrm{c} v_\mathrm{c} \Delta p_\mathrm{c}}{Q_\mathrm{c} - Q_\mathrm{h} - \dot{m}_\mathrm{h} v_\mathrm{h} \Delta p_\mathrm{h} - \dot{m}_\mathrm{c} v_\mathrm{c} \Delta p_\mathrm{c}} \end{aligned} \tag{6}$$

Considering our objective as the value of Q_cn, we need only calculate Q_h, W_c, and W_h to evaluate the *COP*.

3. THE PHYSICAL MODEL

We considered the R134a as working fluid in a simple compression cycle (*figure 2*) with output temperatures from condenser and evaporator:

$$\begin{aligned} T_\mathrm{fho} &= T_\mathrm{fh} - \Delta T_\mathrm{sub} \\ T_\mathrm{fco} &= T_\mathrm{fc} + \Delta T_\mathrm{over} \end{aligned} \tag{7}$$

where ΔT_sub and ΔT_over represent subcooling and overheating. Due to the low values assigned to these variables, no particular section of heat exchangers was considered.

From phase change temperatures T_fh and T_fc, the enthalpy difference is evaluated and the mass flow needed to derive heat Q_c assuming isenthalpic throttling:

$$\dot{m}_\mathrm{134a} = \frac{Q_\mathrm{c}}{h_\mathrm{fco} - h_\mathrm{fho}} \tag{8}$$

The refrigerating fluid inlet temperature in the hot heat exchanger T_fhi is evaluated from the evaporator outlet temperature T_fco considering an isentropic compression modified by an isentropic efficiency calculated as a function of the pressure ratio and derived from literature data [15] (see appendix).

Power Q_h given to the high-temperature fluid is

$$Q_\mathrm{h} = \dot{m}_\mathrm{134a}(h_\mathrm{fho} - h_\mathrm{fhi}) \tag{9}$$

The thermodynamic properties of the R134a working fluid were evaluated using REFPROP 5.0 [16].

The thermophysical characteristics of the water were derived from average temperatures with the polynomials obtained from Raznjevic [17]:

$$\begin{aligned} \rho &= 2.6896 - 7.1084{\cdot}10^{-3} T + 7.4569{\cdot}10^{-6} T^2 \\ &\quad - 2.6636{\cdot}10^{-9} T^3 \\ \mu &= \big(3.9416 + 0.0521 T - 1.6449{\cdot}10^{-5} T^2 \\ &\quad + 2.7827{\cdot}10^{-9} T^3\big){\cdot}10^{-6} \\ k &= 1.898{\cdot}10^{-3} + 8.9614{\cdot}10^{-5} T - 3.73188{\cdot}10^{-8} T^2 \\ &\quad + 9.9215{\cdot}10^{-12} T^3 \end{aligned} \tag{10}$$

Starting from defined water inlet temperatures T_ci and T_hi, plus cooling load Q_c, the *COP* is related to the 8 variables:

$$T_\mathrm{fh},\ T_\mathrm{fc},\ \dot{m}_\mathrm{h},\ \dot{m}_\mathrm{c},\ D_\mathrm{h},\ D_\mathrm{c},\ n_\mathrm{h},\ n_\mathrm{c} \tag{11}$$

where D and n are the diameter and the number of water tubes in the heat exchangers. The *COP* can be determined with conditions:

$$\begin{aligned} T_\mathrm{fho} &\geqslant T_\mathrm{hi}; \qquad T_\mathrm{fh} \geqslant T_\mathrm{ho} \\ T_\mathrm{ci} &\geqslant T_\mathrm{fco}; \qquad T_\mathrm{co} \geqslant T_\mathrm{fci} \end{aligned} \tag{12}$$

The second condition is due to the pinch-point in the hot heat exchanger. Friction losses in the working fluid were considered negligible because of the shell and tube type of heat exchangers chosen, with water flowing in horizontal parallel tubes.

4. DEFINING HEAT EXCHANGER DIMENSIONS

Starting from fixed values of variables (11) forced convection heat exchange factor h_l for water side can be calculated by the Petukhov relation [18]:

$$Nu = \frac{(Re - 1\,000)Pr\,z}{1 + 12.7(Pr^{2/3} - 1)z^{1/2}} \tag{13}$$

with $z = 0.5/(1.58 \ln Re - 3.28)^2$. This equation is valid for $2\,300 < Re < 5{\cdot}10^6$. In the laminar field we assumed $Nu = 3.66$, considering fully developed condition.

Since the evaporator was considered to be flowed, the mean boiling heat transfer factor is evaluated by [19]

$$h_\mathrm{e} = 0.62\left[\frac{k_\mathrm{v}^3 g\rho_\mathrm{v}(\rho_\mathrm{l} - \rho_\mathrm{v})r'}{\mu_\mathrm{v}(T_\mathrm{w} - T_\mathrm{sat})D}\right]^{1/4} \tag{14}$$

with

$$r' = r + 0.40c_\mathrm{pv}(T_\mathrm{w} - T_\mathrm{sat})$$

Wall temperature T_w is calculated by an energy balance, with Q_e known.

Neglecting thermal resistance due to thermal conduction in the tube, global heat exchange factor $U = 1/(1/h_\mathrm{e} + 1/h_\mathrm{l})$ can be calculated and, from the classical relation using the logarithmic mean temperature difference

$$Q = UA\Delta T_\mathrm{ml} \tag{15}$$

the heat exchanger area A, with unknown temperatures evaluated by energy balances.

In a similar way we can calculate the condenser area using equation (13) for water side heat exchange and the mean condensing heat exchange factor evaluated from [19]

$$h_\mathrm{d} = \frac{0.728k_\mathrm{l}}{n_\mathrm{r}D}\left[\frac{g(\rho_\mathrm{l} - \rho_\mathrm{v})(n_\mathrm{r}D)^3 r}{k_\mathrm{l}\nu_\mathrm{l}(T_\mathrm{sat} - T_\mathrm{w})}\right]^{1/4} \tag{16}$$

In this case we impose first rank cooling of vapor to the saturation line. By knowledge of inlet and condensing temperatures, it is possible to evaluate the cooling load and using the heat transfer coefficient vapor side in this rank from [20]

$$Nu = 0.33Re^{0.6}Pr^{1/3} \tag{17}$$

the number of tubes and the distance between them can be calculated for the first rank of the condenser.

As a result of the heat exchanger design it is possible to calculate the two lengths L_c and L_h, then the pressure losses using the explicit Moody relation, which Haaland [21] proposed again for the friction factor:

$$f = 0.0055\left[1 + \left(2{\cdot}10^4\frac{\varepsilon}{D} + \frac{10^6}{Re}\right)^{1/3}\right] \tag{18}$$

Then the COP is evaluated using equation (6).

5. APPLICATION OF A DIRECT SEARCH METHOD TO FIND THE MAXIMUM *COP*

Direct search methods must be used when the gradient of the objective function is a complex vector of the design variables, which appreciably complicate the analytical expression. The procedure we used, the well-known complex method proposed by Box et al. [22], begins by randomly and sequentially generating a set of trial points in the space of the independent variables and evaluating the function at each vertex. Each newly generated point is tested for feasibility, and, if found unfeasible, is moved back toward the centroid of the previously generated points until it becomes feasible. The search continues in this way until the pattern of points has shrunk, so that the points are sufficiently close together and/or the difference between the function values at the points becomes small enough.

As the random nature of this method gives rise to the possibility of a premature collapse of the cloud of points, we performed a sampling of trials from different starting conditions, estimating points reliability on their coefficient of variation.

6. RESULTS AND DISCUSSION

Considering a cooling power of 50 kW, there were never less then 7 different trials, and they gave a good agreement in COP maximum values for a given set of ranges of the 8 variables (see (11)). Varying the amplitude of the ranges, different values of COP were obtained with 6 variables always well inside the ranges and the numbers of tubes at the extreme maximum values. When increasing the ranges for the tube numbers, the COPs increase. This means the maxima obtained are strictly conditioned by the heat exchanger area. The results are shown in *figure 3* with COP versus the total

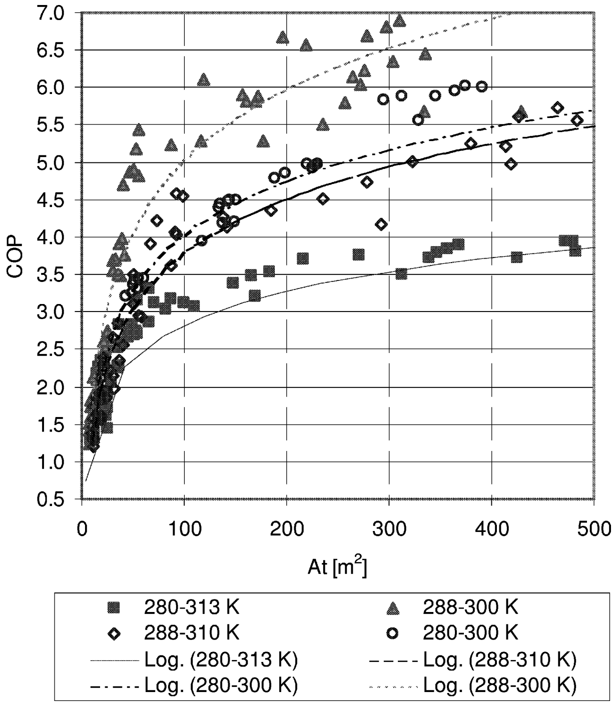

Figure 3. *COP* versus total area of heat exchangers at various inlet temperatures of external fluids $T_{ci} - T_{hi}$. Logarithmic correlations include also data out of the diagram.

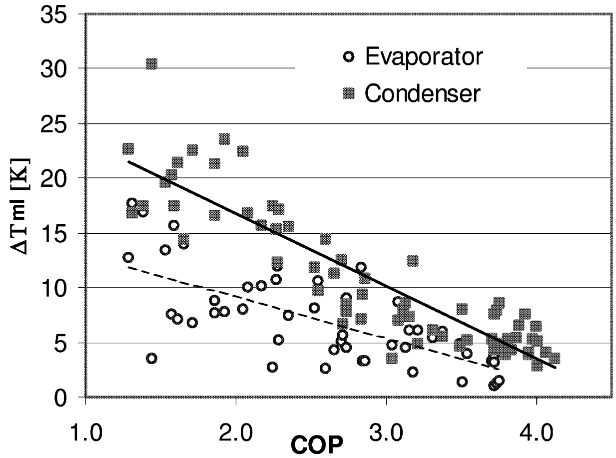

Figure 4. ΔT_{ml} calculated from equation (15) versus *COP* at $T_{ci} = 280$ K and $T_{hi} = 313$ K.

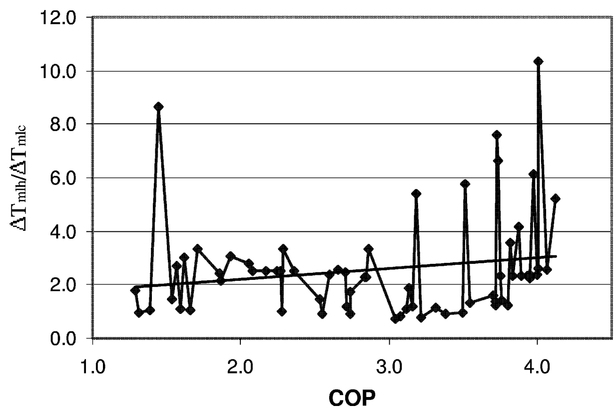

Figure 5. ΔT_{ml} from equation (15), ratio of hot over cold side versus *COP* at $T_{ci} = 280$ K and $T_{hi} = 313$ K.

area of heat exchangers. The water inlet temperatures were also changed, obtaining curves with the same trend. It is easy to see that the increasing values of *COP* have different asymptotic limits when changing the temperature difference between external fluids. The initial growth of *COP* is very high and is a function mainly of the heat exchanger area. Let us also consider the fact that equation (6) includes energy to overcome pressure losses: thermodynamic efficiency η, obtained as the ratio of *COP* over COP_C, ranges from 0.15 to 0.49. The Carnot efficiency COP_C is evaluated using inlet water temperatures and is 8.49, when $T_{ci} = 280.15$ and $T_{hi} = 313.15$.

For a given total surface of heat exchangers, the evaporator and the condenser have different characteristics. *Figures 4* and *5* show the different decreases of logarithmic mean temperature differences, from equation (15), at increasing *COP*.

The different designs required by the evaporator and condenser are also visible from *figures 6* and *7*, where the UA product is plotted versus *COP*. The scattering of points is due to the number of trials. The ratio between the hot and cold values is lower than one against the expectation due to theory [23]. This theory was developed for constant temperatures of heat exchange, when considering a limiting value to the sum of cold and hot UA, because of limited resources.

Considering Kays' efficiency [24]:

$$\varepsilon = \frac{Q}{C_{min}(T_{max} - T_{min})} \tag{19}$$

where C_{min} is the capacity rate of water, because we are still considering heat exchanged to a phase changing fluid, then Q can be expressed as

$$Q_h = K_h(T_{fhi} - T_{hi}) \tag{20}$$

$$Q_c = K_c(T_{ci} - T_{fci}) \tag{21}$$

Figure 7 shows as the ratio:

$$\beta = \frac{K_h}{K_c} \tag{22}$$

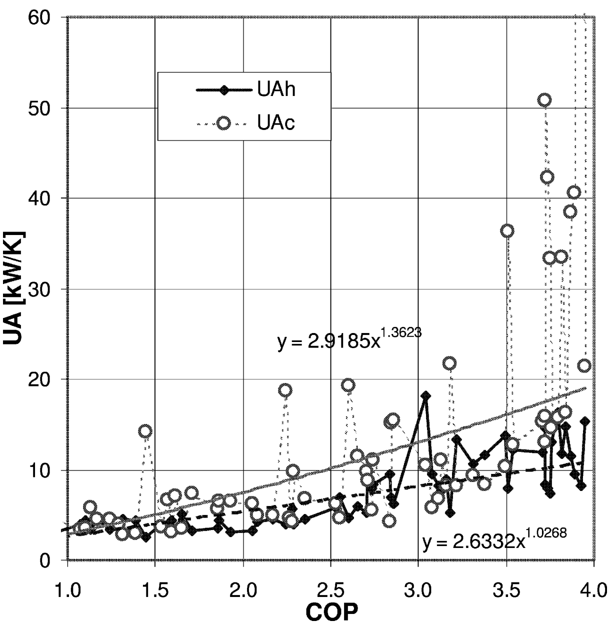

Figure 6. Hot and cold values of UA from equation (15) versus COP; $T_{ci} = 280$ K; $T_{hi} = 313$ K.

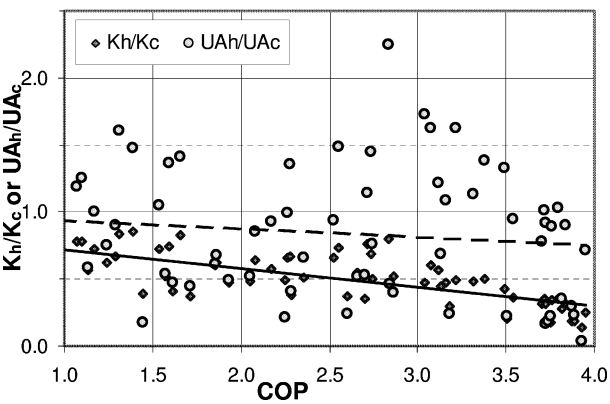

Figure 7. Ratio between hot and cold UA from equation (15) and β from equation (22) versus COP; $T_{ci} = 280$ K; $T_{hi} = 310$ K.

is a decreasing function of COP, consistent with other heat exchange thermal parameters. In other words, a growing COP requires lower and lower thermal resistances, i.e. irreversibilities, particularly at the cold side.

We can say the same from *figure 8*, showing the efficiency decrease versus COP, with higher efficiency at the evaporator. These results disagree with theory [4, 7]. We believe the reason for this difference is the constant temperature considered in theory for sources. Different sizes for heat exchangers are shown in the literature when considering temperatures variations [12, 25]. *Figure 3* shows

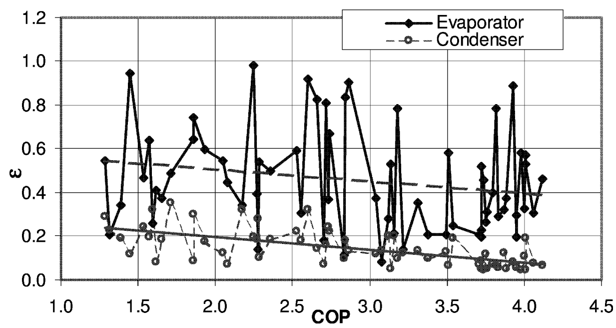

Figure 8. Kays efficiencies, equation (19), as a function of COP; $T_{ci} = 280$ K; $T_{hi} = 310$ K.

there is not a maximum for COP without limits to the heat exchanger area. So we searched for an optimum for COP/At function, the maximum COP with minimum dimensioning. *Table I* shows the results obtained searching those two optima with given input data. The results refer to seven trials. It is easy to see the dispersion of data referring to water mass flow and dimensions, with a quite constant COP.

7. CONCLUSIONS

The software described permits fast thermodynamic optimisation of the proposed simple cycle. Results are in agreement with those presented in the literature and confirm the importance of the heat exchangers when high efficiency is required. The temperature differences between fluids were smaller than those predicted by an earlier isothermal model [5], probably because of the irreversibilities due to friction factors.

The particular kind of heat exchanger chosen because it favours avoidance of internal pressure losses may, as a consequence, augment the surface of heat exchange, due to the fact that the external irreversibilities in water influence the thermodynamic value of the reversed cycle less. Anyway, the cold section is overdimensioned in comparison to the hot one, because the entropy variation due to the thermal irreversibilities is higher when temperature is lower.

It is very important to consider that the maximum COP is obtained with $UA_c > UA_h$. The evaporator requires better efficiency and lower logarithmic mean temperature difference.

Moreover, the greater COP corresponds to lower heat exchanger efficiency. In other words, the best working conditions of the system are different from those corresponding to component optimisation, because phenom-

TABLE I
Example of results showing data dispersion. The maxima refer to each trial.

Input data	T_{ci}	T_{hi}	ΔT_{over}	ΔT_{sub}	Q_c (kW)			
	280.15	313.15	2	3	50			
Variable ranges	T_{fc}	T_{fh}	$\dot{m}_c$	$\dot{m}_h$	D_c	D_h	n_c	n_h
min	250.15	313.15	1	1	0.002	0.002	50	50
max	280.15	363.15	20	25	0.05	0.05	300	300

Max *COP*

	$\dot{m}_c$	$\dot{m}_h$	D_c	D_h	n_c	n_h	L_c	L_h	A_c	A_h	Q_h	*COP*
1	11.53	15.64	0.027	0.040	195	214	24.04	9.51	395.40	252.30	62.28	4.07
2	13.47	18.30	0.027	0.047	146	237	24.40	6.40	298.70	223.80	62.53	3.95
3	8.79	11.12	0.018	0.042	194	198	99.06	7.59	1093.90	199.30	62.13	4.07
4	13.85	12.48	0.028	0.024	221	214	17.10	11.55	329.00	185.40	62.34	4.04
5	11.81	10.24	0.037	0.028	137	196	80.29	5.67	1261.70	96.60	62.49	3.99
6	14.08	14.33	0.043	0.017	145	179	106.21	6.99	2081.60	68.00	62.27	4.03
7	11.20	8.02	0.036	0.037	127	150	91.09	7.18	1303.90	123.80	62.55	3.97

	K_h	K_c	UA_h	UA_c	ΔT_{mlh}	ΔT_{mlc}	$\dot{m}_{134a}$	Δp_h	Δp_c	Re_h	Re_c
1	3.44	17.04	17.46	36.72	3.57	1.36	0.321	0.003	0.010	3709	1917
2	3.37	15.19	12.96	27.91	4.83	1.79	0.323	0.001	0.101	3285	3014
3	3.39	23.72	12.87	150.43	4.83	0.33	0.321	0.001	0.113	2673	2165
4	3.45	15.71	21.21	29.60	2.94	1.69	0.321	0.019	0.007	4902	1973
5	3.23	22.53	9.52	86.16	6.56	0.58	0.324	0.003	0.020	3799	2056
6	3.31	24.09	10.73	120.66	5.80	0.41	0.322	0.121	0.017	9278	1969
7	3.19	22.63	9.24	90.61	6.77	0.55	0.325	0.001	0.026	2952	2138

Max *COP*/*At*

	$\dot{m}_c$	$\dot{m}_h$	D_c	D_h	n_c	n_h	L_c	L_h	A_c	A_h	Q_h	*COP*
1	8.07	1.55	0.004	0.002	296	171	0.88	2.64	2.80	2.90	86.09	1.17
2	1.78	15.09	0.002	0.007	288	189	1.63	2.26	3.20	8.90	75.11	1.72
3	3.92	1.37	0.003	0.003	292	103	1.74	5.13	4.40	4.60	73.23	1.68
4	1.15	2.01	0.002	0.002	191	299	3.15	1.45	3.80	2.80	77.07	1.75
5	1.64	1.69	0.002	0.002	224	210	2.39	2.20	3.70	2.90	75.86	1.67
6	2.01	15.98	0.002	0.004	288	297	1.44	1.07	2.80	4.30	83.34	1.26
7	1.52	12.33	0.002	0.004	263	291	1.69	1.13	2.80	4.00	82.58	1.35

	K_h	K_c	UA_h	UA_c	ΔT_{mlh}	ΔT_{mlc}	$\dot{m}_{134a}$	Δp_h	Δp_c	Re_h	Re_c
1	1.59	1.88	3.73	2.02	23.06	24.81	0.407	2.901	2.028	9865	6805
2	1.99	2.73	3.63	3.61	20.68	13.84	0.384	3.623	0.182	24219	2276
3	2.03	3.29	4.47	3.95	16.38	12.67	0.381	3.705	1.753	10600	4132
4	1.83	2.85	3.53	4.60	21.85	10.86	0.397	0.701	0.473	7070	2268
5	1.89	2.96	3.80	4.13	19.96	12.10	0.392	1.858	1.416	8699	2673
6	1.60	2.54	2.66	3.19	31.33	15.66	0.423	4.706	0.834	25281	2662
7	1.63	2.53	2.74	3.41	30.16	14.66	0.419	3.929	0.233	21846	2278

ena involved in the cycle are not linear. Then we have to model the complete systems to find an optimum.

The optimum itself is variable as a function of external parameters, not only of those of the cycles, and we actually have to search for a compromise.

The reduction of heat exchange irreversibilities is confirmed as a powerful method for increasing the efficiency of the inverse cycles.

REFERENCES

[1] Feidt M., Ramany-Bala P., Benelmir R., Synthesis, unification and generalization of preceding finite time thermodynamic studies devoted to Carnot cycles, in: Carnevale E., Manfrida G., Martelli F. (Eds.), Proc. Florence World Energy Res. Symp., Florence, Italy, SGE, Padua, Italy, 1994, pp. 429–439.

[2] Chen L., Sun F., Chen W., Optimization of the specific rate of refrigeration in combined refrigeration cycles, Energy 20 (1995) 1049–1053.

[3] Alefeld G., Efficiency of compressor heat pumps and refrigerators derived from the second law of thermodynamics, Int. J. Refrig. 10 (1987) 331–341.

[4] Bejan A., Theory of heat transfer irreversible refrigeration plants, Int. J. Heat Mass Tran. 32 (1989) 1631–1638.

[5] Grazzini G., Irreversible refrigerators with isothermal heat exchangers, Int. J. Refrig. 16 (1993) 101–106.

[6] Domansky P.A., McLinden M.O., A simplified cycle simulation model for the performance rating of refrigerants and refrigerant mixtures, Int. J. Refrig. 15 (1992) 81–88.

[7] Klein S.A., Design considerations for refrigeration cycles, Int. J. Refrig. 15 (1992) 181–185.

[8] Jakobsen A., Dynamic modelling and simulation of a domestic refrigerator, in: Proceedings of 19th Intern. Congr. of Refrig., The Hague, The Netherlands, IIR, Vol. IIIa, 1995, pp. 345–352.

[9] Radcenco V., Vargas J.V.C., Bejan A., Lim J.S., Two design aspects of defrosting refrigerators, Int. J. Refrig. 18 (1995) 76–86.

[10] Shen X., Ji S., Wang L., Xi J., Zao G., Experimental verification of a mathematical model for simulation of a compressor refrigeration capacity test system, in: Proceedings of 19th Intern. Congr. of Refrig., The Hague, The Netherlands, IIR, Vol. IIIa, 1995, pp. 530–537.

[11] Chua H.T., Ng K.C., Gordon J.M., Experimental study of the fundamental properties of reciprocating chillers and their relation to thermodynamic modelling and chiller design, Int. J. Heat Mass Tran. 39 (1996) 2195–2204.

[12] Gordon J.M., Ng K.C., Chua H.T., Optimizing chiller operation based on finite-time thermodynamics: universal modelling and experimental confirmation, Int. J. Refrig. 20 (1997) 191–200.

[13] Schiefloe P.A., Application of large-scale optimization in the design of heat pump systems, in: Proceedings of 19th Intern. Congr. of Refrig., The Hague, The Netherlands, IIR, Vol. IVb, 1995, pp. 1235–1243.

[14] Vaisman I.B., Topological approach to thermodynamic and cost analysis of refrigerating systems, in: Proceedings of 19th Intern. Congr. of Refrig., The Hague, The Netherlands, IIR, Vol. IIIa, 1995, pp. 522–529.

[15] Serralach G.F., Cleland A.C., Cleland D.J., Extension of an empirical method for estimating refrigeration system energy usage to new refrigerants, in: Refrig. Climate Control and Energy Conserv., IIR, Melbourne, Australia, February 11–14, 1996, pp. 214–221.

[16] Huber M., Gallagher J., McLinden M., Morrisoon G., Thermodynamic properties of refrigerants and refrigerant mixtures database (REFPROP), NIST, Gaithesburg, MD, USA, 1996.

[17] Raznjevic K., Tabelle Termodinamiche, Del Bianco Editore, Milan, Italy, 1971.

[18] Kays W.M., Perkins H.C., Forced convection internal flow in ducts, in: Handbook of Heat Transfer Fundamentals, Rohsenow W.M., Hartnett J.P., Ganic E.N. (Eds.), McGraw-Hill, 1985.

[19] Carey V.P., Liquid–vapour phase-change phenomena, Hemisphere, Washington, 1992.

[20] ASHRAE Handbook, Fundamentals, ASHRAE, Atlanta, GA, USA, 1997.

[21] Haaland S.E., Simple and explicit formulas for the friction factor in turbulent pipe flow, J. Fluids Engrg. 105 (1983) 89–90.

[22] Box M.J., Davies D., Swann W.H., Non-linear Optimization Techniques, ICI Monograph, Vol. 5, Oliver & Boyd, London, 1969.

[23] Bejan A., Power and refrigeration plants for minimum heat exchanger inventory, J. Energ. Res. Tech. 115 (1993) 148–150.

[24] Kays W., London A.I., Compact Heat Exchangers, McGraw-Hill, New York, 1964.

[25] Khan J.R., Zubair S.M., Design and performance evaluation of reciprocating refrigeration systems, Int. J. Refrig. 22 (1999) 235–243.

APPENDIX

The relation showing compressor isentropic efficiency ie as a function of pressure ratio $pr = p_\mathrm{h}/p_\mathrm{c}$:

$$ie = g_1 + pr\{g_2 + pr[g_3 + pr(g_4 + pr g_5)]\}$$

with

$$g_1 = 0.493812291079678$$
$$g_2 = 9.49097844311382 \cdot 10^{-2}$$
$$g_3 = -1.37579085197785 \cdot 10^{-2}$$
$$g_4 = 6.54918082918954 \cdot 10^{-4}$$
$$g_5 = -1.14009112551078 \cdot 10^{-5}$$

is derived from literature data [15] for reciprocating, rotary vane and twin screw compressors and gives values ranging from 0.30 to 0.65 for a pressure ratio in the range 20 to 2.