

## ON THE EFFICIENCIES OF ABSORPTION HEAT TRANSFORMERS

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**Abstract**—Although commonly used, the coefficient of performance (COP) is not always an adequate measure to describe the effectiveness of a sorption heat pump. Equations for four different efficiencies are derived, discussed and compared for absorption heat transformers. A flow-sheeting computer program, developed for both design and evaluation simulations of arbitrarily complex absorption cycles, is used to exemplify the derived equations. The working pair H<sub>2</sub>O–NaOH is used in two heat transformer systems. The given examples clearly show that the COP can only be used to compare different heat transformers operated at the same circulation ratio. The COP can be considered as an indicator of the effectiveness of heat exchange within and thermal insulation of a heat transformer operated at a fixed circulation ratio. The thermodynamic efficiency,  $E_{th}$ , is shown to be a more logical measure of the heat transformer efficiency, since it takes into account both heat losses, heat exchange and the temperature lift. The exergetic efficiency,  $E_{ex}$ , is an alternative to  $E_{th}$  since both are mathematically compatible. The possibility of taking into consideration the temperature level, at which heat energy may be considered economically worthless, is demonstrated to be a major advantage of the exergetic efficiency. However, its numerical value does not provide a clear interpretation of the importance of exergetic losses in the system. The exergetic index,  $I_{ex}$ , is directly related to the exergetic efficiency but its numerical value is more significant for evaluating the performance of the heat transformer system.

### NOMENCLATURE

COP	coefficient of performance
$E$	efficiency
$\Delta e$	exergy change (kJ/kg)
$F$	flow (kg/s)
$\Delta h$	enthalpy change (kJ/kg)
$I$	index, exergetic
$m$	mass flow rate (kg/s)
$Q$	heat flow (kW)
$\Delta s$	entropy change (kJ/kg K)
$T$	temperature (K)
$X$	concentration (kg/kg)
$\Phi$	exergy losses, irreversibilities (kJ/kg)

#### Subscripts

a	absorber
c	condenser
e	evaporator
ex	exergetic
g	generator
l	liquid of working medium
o	reference state
rev	reversible
s	strong solution
th	thermodynamic
v	vapour of working medium
w	weak solution

### 1. INTRODUCTION

Absorption heat pumps (AHP) are based on the principle of utilization of the enthalpies of evaporation–condensation of solutes in appropriate working fluid mixtures. These systems consume very little electric energy and use mainly a high-temperature (primary heat) and a low-temperature (waste heat) heat source to recover useful heat at an intermediate temperature. Some electrical energy is used only for the purpose of recirculating the working fluid.

A heat transformer is a reversed AHP where heat is supplied at an intermediate level and useful heat is delivered at the highest temperature, provided there is a heat sink with a lower temperature level than that of the waste heat supplied. The heat released in the condenser is normally not used, since it is at a relatively low temperature. The available useful energy output of the heat transformer is about 50% of the waste heat input. However, a heat transformer does not consume high-grade thermal energy and only some electrical energy is used to recirculate the liquid streams.

Among the different systems currently adopted for the recuperation of industrial waste heat, absorption heat transformer systems (AHT) are considered to be the most competitive. This is due primarily to the fact that they are operated using waste heat and require only minimal prime energy for pumps. Although sizable quantities of energy may be extracted from a process waste heat stream, it may not always possess the necessary temperature level for its industrial utilization. Depending on the chemical properties of the working pair applied, the AHT system is highly appropriate for lifting the temperature of thermal energy streams from moderate temperature levels.

Although the principle of operation of AHT was already established at the beginning of this century, its industrial exploitation only began during the last 12–15 years, mostly in Japan and Western Europe [1]. Several recent investigations have clearly demonstrated the technical and economic feasibility of AHT in comparison to other methods of waste heat recovery [2–7]. A comprehensive economic analysis, using both pay-off and levelized annual cost investment methods, has recently been published [8]. The performance of different chemical heat pump systems was compared with both mechanical and thermal vapour recompression systems, MVR and TVR respectively, at different electricity to fuel ( $e/f$ ) price ratios. It was concluded, for the  $e/f$  ratios investigated, that AHT are more economical at higher temperature lifts and higher  $e/f$  ratios. The MVR and TVR systems were found more economical at lower lifts, with MVR being favoured at lower  $e/f$  ratios and TVR at higher  $e/f$  ratios.

Being a thermodynamic function which provides a measure of the energy quality as well as quantity, exergy denotes the maximum work that can be extracted from a given system in a given state in any process which allows the system to come into equilibrium with its surroundings. Whereas the Second Law of Thermodynamics is usually thought of in terms of entropy and the increase in entropy associated with any irreversible process, an alternate viewpoint would consider the exergy decrease associated with all irreversible processes. Exergy losses occurring during a process can be calculated by making exergy balances for each component of the system. In energy balances, the inflow streams are equated to the outflow streams, provided no energy generation or consumption occur. On the other hand, in exergy balances, due to reasons of irreversibilities, the exergy inflow is always greater than the exergy outflow and the difference represents the exergy losses in the system. It is these losses which have to be reduced to improve the thermodynamic efficiency of the system, and hence devise means by which these can be minimized.

Exergy analysis and the concept of the Second Law of Thermodynamics has invoked considerable interest in recent years [9–24], since its application leads to a better understanding of the energy transfer, and helps to identify the thermodynamic losses properly. The importance of Second Law analysis as compared to the conventional energy conservation analysis, is mainly due to the fact that the latter does not distinguish between the quality of energy, and hence assigns high quality forms of energy to low quality purposes. Moreover, it results in assigning efficiency values greater than unity for certain processes like heat pump and refrigeration applications, thus departing from the conventional efficiency concepts. Second Law analysis takes into account both the quantity of energy consumed and the quality of energy conversion. Besides, it allows for the identification of exergy losses which inevitably occur in some components of the system, such as heat transfer, throttling, desuperheating, etc.

## 2. DIFFERENT EFFICIENCIES FOR AHT SYSTEMS

The coefficient of performance (COP) is often used for the evaluation of an absorption heat pump,  $(Q_a + Q_c)/Q_g$ , or a heat transformer,  $Q_a/(Q_g + Q_c)$ . However, this is not always the best criterion to express the effectiveness of sorption heat pump systems, as will be demonstrated later in this paper. Other measures of performance have been mentioned in literature [19–21], including energetic, thermodynamic and exergetic efficiencies.

The main objective of this work is to discuss different measures to express the efficiency of this type of absorption cycles. The derived expressions would provide alternatives to the commonly used COP. For this purpose, the working pair  $H_2O$ - $NaOH$  is used in two AHT systems, simulated for quantitative comparison between the different efficiencies.

### 2.1. Carnot efficiency, $COP_{rev}$

The maximum value of the coefficient of performance, COP, is obtained for an ideal reversible cycle operating at the same specified conditions as the real cycle, and is commonly referred to as the Carnot efficiency,  $COP_{rev}$ , or the Carnot COP. The following entropy and enthalpy balances can be written:

$$\begin{cases} m_a \cdot \Delta s_a + m_c \cdot \Delta s_c = m_g \cdot \Delta s_g + m_e \cdot \Delta s_e & (1) \\ m_a \cdot \Delta h_a + m_c \cdot \Delta h_c = m_g \cdot \Delta h_g + m_e \cdot \Delta h_e & (2) \end{cases}$$

Equation (1) gives

$$m_c = \frac{m_g \cdot \Delta s_g + m_e \cdot \Delta s_e - m_a \cdot \Delta s_a}{\Delta s_c}$$

which together with equation (2) gives

$$m_a \cdot \Delta h_a + m_g \cdot \Delta h_g \cdot \frac{\Delta s_g}{\Delta s_c} + m_e \cdot \Delta h_e \cdot \frac{\Delta s_e}{\Delta s_c} - m_a \cdot \Delta h_c \cdot \frac{\Delta s_a}{\Delta s_c} = m_g \cdot \Delta h_g + m_e \cdot \Delta h_e. \quad (3)$$

Rearranging and solving for  $m_a \cdot \Delta h_a$  we get

$$m_a \cdot \Delta h_a = m_g \cdot \Delta h_g \cdot \frac{1 - \frac{\Delta h_c \cdot \Delta s_g}{\Delta h_g \cdot \Delta s_c}}{1 - \frac{\Delta h_c \cdot \Delta s_a}{\Delta h_a \cdot \Delta s_c}} + m_e \cdot \Delta h_e \cdot \frac{1 - \frac{\Delta h_c \cdot \Delta s_e}{\Delta h_e \cdot \Delta s_c}}{1 - \frac{\Delta h_c \cdot \Delta s_a}{\Delta h_a \cdot \Delta s_c}} \quad (4)$$

where  $m_a \cdot \Delta h_a$  is the heat energy delivered from the absorber,  $(Q_a)_{rev}$ . The following equation can then be written:

$$(Q_a)_{rev} = \frac{Q_g + Q_e - Q_g \cdot \frac{\Delta h_c \cdot \Delta s_g}{\Delta h_g \cdot \Delta s_c} - Q_e \cdot \frac{\Delta h_c \cdot \Delta s_e}{\Delta h_e \cdot \Delta s_c}}{1 - \frac{\Delta h_c \cdot \Delta s_a}{\Delta h_a \cdot \Delta s_c}} \quad (5)$$

Here  $(Q_a)_{rev}$  is the highest possible heat output from the absorber for given real heat inputs in both generator and evaporator, and given temperatures in the absorber, evaporator, generator, and condenser. This gives the following equation:

$$COP_{rev} = \frac{1 - \frac{Q_g}{Q_g + Q_e} \cdot \frac{\Delta h_c \cdot \Delta s_g}{\Delta h_g \cdot \Delta s_c} - \frac{Q_e}{Q_g + Q_e} \cdot \frac{\Delta h_c \cdot \Delta s_e}{\Delta h_e \cdot \Delta s_c}}{1 - \frac{\Delta h_c \cdot \Delta s_a}{\Delta h_a \cdot \Delta s_c}} \quad (6)$$

After some algebraic rearranging, the coefficient of performance for a reversible heat transforming process can be written according to

$$COP_{rev} = \frac{\left(\frac{\Delta s_c}{\Delta h_c} - \frac{\Delta s_g}{\Delta h_g}\right) + \frac{Q_e}{Q_g} \cdot \left(\frac{\Delta s_c}{\Delta h_c} - \frac{\Delta s_e}{\Delta h_e}\right)}{\left(1 + \frac{Q_e}{Q_g}\right) \cdot \left(\frac{\Delta s_c}{\Delta h_c} - \frac{\Delta s_a}{\Delta h_a}\right)} \quad (7)$$

Assuming that heat energy is transferred to and from the heat transformer at constant temperatures  $T_g$ ,  $T_e$ ,  $T_a$  and  $T_c$ , the above equation can be written as:

$$\text{COP}_{\text{rev}} = \frac{\left(\frac{1}{T_c} - \frac{1}{T_g}\right) + \frac{Q_e}{Q_g} \cdot \left(\frac{1}{T_c} - \frac{1}{T_e}\right)}{\left(1 + \frac{Q_e}{Q_g}\right) \cdot \left(\frac{1}{T_c} - \frac{1}{T_a}\right)}. \quad (8)$$

As expressed by equation (8), the maximum efficiency,  $\text{COP}_{\text{rev}}$ , is a simple function of the real energy input to both generator and evaporator as well as the temperature of the four components of the heat transformer system. When heat energy is delivered to the generator and evaporator at the same temperature, equation (8) can be further simplified to the following form:

$$\text{COP}_{\text{rev}} = \frac{T_a}{T_a - T_c} \cdot \frac{T_e - T_c}{T_e}. \quad (9)$$

### 2.2. Thermodynamic efficiency, $E_{\text{th}}$

Consider the thermodynamic efficiency to be defined according to

$$E_{\text{th}} = \frac{\text{COP}}{\text{COP}_{\text{rev}}} \quad (10)$$

that is

$$E_{\text{th}} = \frac{Q_a \cdot \left(\frac{\Delta s_c}{\Delta h_c} - \frac{\Delta s_a}{\Delta h_a}\right)}{Q_g \cdot \left(\frac{\Delta s_c}{\Delta h_c} - \frac{\Delta s_g}{\Delta h_g}\right) + Q_e \cdot \left(\frac{\Delta s_c}{\Delta h_c} - \frac{\Delta s_e}{\Delta h_e}\right)}. \quad (11)$$

Assuming that heat energy is transferred to and from the heat transformer at constant temperatures  $T_g$ ,  $T_e$ ,  $T_a$  and  $T_c$ , the thermodynamic efficiency can then be expressed as:

$$E_{\text{th}} = \frac{Q_a \cdot \left(\frac{1}{T_c} - \frac{1}{T_a}\right)}{Q_g \cdot \left(\frac{1}{T_c} - \frac{1}{T_g}\right) + Q_e \cdot \left(\frac{1}{T_c} - \frac{1}{T_e}\right)}. \quad (12)$$

This equation relates  $E_{\text{th}}$  to the real energy output from the absorber,  $Q_a$ , real energy input to both generator and evaporator,  $Q_g$  and  $Q_e$ , and the four temperatures  $T_g$ ,  $T_e$ ,  $T_a$  and  $T_c$ . When heat energy is delivered to the generator and evaporator at the same temperature, we get

$$E_{\text{th}} = \text{COP} \cdot \frac{T_a - T_c}{T_a} \cdot \frac{T_e}{T_e - T_c} = \frac{\text{COP}}{\text{COP}_{\text{rev}}}. \quad (13)$$

### 2.3. Exergetic efficiency, $E_{\text{ex}}$

The thermodynamic efficiencies discussed above provide expressions for the performance of AHT systems, but do not depict the complete picture. In particular, the quality of the energy output from the absorber and condenser is not specifically addressed by these performance measures.

The exergy change for an external cooling or heating stream can be written as:

$$m \cdot \Delta e = m \cdot \Delta h - m \cdot T_o \cdot \Delta s \quad (14)$$

where  $T_o$  is the reference temperature and may be chosen at a level where heat energy is considered worthless for an actual application. This gives directly

$$m \cdot \Delta e = Q \cdot \left(1 - \frac{T_o \cdot \Delta s}{\Delta h}\right) \quad (15)$$

where  $Q$  is the cooling or heating energy flow. Since the energy output from the condenser cannot be utilized in practice, it will be neglected in this analysis. Consequently, provided that  $T_c \leq T_o$ ,

we may define the exergetic efficiency as the ratio of the exergy gained to the exergy expended by the system, and can thus be written according to

$$E_{\text{ex}} = \frac{Q_a \cdot \left( \frac{1}{T_o} - \frac{\Delta s_a}{\Delta h_a} \right)}{Q_g \cdot \left( \frac{1}{T_o} - \frac{\Delta s_g}{\Delta h_g} \right) + Q_c \cdot \left( \frac{1}{T_o} - \frac{\Delta s_c}{\Delta h_c} \right)}. \quad (16)$$

Assuming that heat energy is transferred to both generator and evaporator, and from the absorber at constant temperatures  $T_g$ ,  $T_e$  and  $T_a$  respectively, the exergetic efficiency can then be expressed as:

$$E_{\text{ex}} = \frac{Q_a \cdot \left( \frac{1}{T_o} - \frac{1}{T_a} \right)}{Q_g \cdot \left( \frac{1}{T_o} - \frac{1}{T_g} \right) + Q_c \cdot \left( \frac{1}{T_o} - \frac{1}{T_e} \right)}. \quad (17)$$

It may be observed that equations (12) and (17) become similar if  $T_o = T_c$ . Consequently, the exergetic efficiency  $E_{\text{ex}}$  is a function of the same process variables as those for the thermodynamic efficiency  $E_{\text{th}}$ . When heat energy is delivered to the generator and evaporator at the same temperature, equation (17) can be simplified to:

$$E_{\text{ex}} = \text{COP} \cdot \frac{T_a - T_o}{T_a} \cdot \frac{T_c}{T_c - T_o}. \quad (18)$$

For the special case when  $T_o = T_c$ , the following simple expression can be obtained:

$$E_{\text{ex}} = \text{COP} \cdot \frac{T_a - T_c}{T_a} \cdot \frac{T_c}{T_c - T_c} = E_{\text{th}} = \frac{\text{COP}}{\text{COP}_{\text{rev}}}. \quad (19)$$

It should be mentioned that the selection of an appropriate reference state is important as it is explicitly involved in calculating exergy. Hence, changing reference state values will lead to different exergy values, but as one is generally interested in calculating the exergy difference and not in the absolute values, this limitation should not be considered too serious.

#### 2.4. Exergy index, $I_{\text{ex}}$

Considered as indicators of the thermodynamic efficiency, the numerical values of both  $E_{\text{th}}$  and  $E_{\text{ex}}$  do not really reflect the importance of exergetic losses in the sorption cycle considered in this work. The exergy index, first proposed by Duarte and Bugarel [22], is a more adequate indicator of efficiency for these systems.

Neglecting the work done by the pumps and the exergy produced in the condenser, the exergy index,  $I_{\text{ex}}$ , is defined as the ratio between the rate of exergy produced in the absorber and the exergy lost or irreversibility,  $\Phi$ , and can thus be written as follows:

$$I_{\text{ex}} = \frac{Q_a}{\Phi} \cdot \left( \frac{1}{T_o} - \frac{\Delta s_a}{\Delta h_a} \right). \quad (20)$$

Using the Second Law of Thermodynamics, the irreversibility  $\Phi$  in any transformation is equal to  $T_o \cdot \Delta s$ , where  $\Delta s$  is the total entropy increase of the system and surroundings. Applying the First and Second Law of Thermodynamics for the case when  $T_c \leq T_o$ ,  $\Phi$  can be expressed as:

$$\Phi = Q_g \cdot \left( \frac{1}{T_o} - \frac{\Delta s_g}{\Delta h_g} \right) + Q_c \cdot \left( \frac{1}{T_o} - \frac{\Delta s_c}{\Delta h_c} \right) - Q_a \cdot \left( \frac{1}{T_o} - \frac{\Delta s_a}{\Delta h_a} \right). \quad (21)$$

Assuming that heat energy is transferred within the heat transformer system at constant temperatures  $T_g$ ,  $T_e$  and  $T_a$ , the exergy index can be written as follows:

$$I_{\text{ex}} = \frac{\left( \frac{1}{T_o} - \frac{1}{T_a} \right)}{\frac{1}{\text{COP}} \cdot \left( \frac{1}{T_o} - \frac{1}{T_e} \right) - \frac{Q_g}{Q_a} \cdot \left( \frac{1}{T_g} - \frac{1}{T_e} \right) - \left( \frac{1}{T_o} - \frac{1}{T_a} \right)}. \quad (22)$$

If the same waste heat stream is used as energy source in both generator and evaporator, i.e.  $T_g = T_c$ , the above equation can then be rearranged to give the relationship between  $I_{ex}$  and COP:

$$I_{ex} = \frac{\left(\frac{1}{T_o} - \frac{1}{T_a}\right)}{\frac{1}{COP} \cdot \left(\frac{1}{T_o} - \frac{1}{T_c}\right) - \left(\frac{1}{T_o} - \frac{1}{T_a}\right)}. \quad (23)$$

From equations (12), (17) and (22), it follows that the exergy index  $I_{ex}$ , thermodynamic  $E_{th}$ , and exergetic efficiency  $E_{ex}$  are interrelated by the following simple expressions:

$$I_{ex} = \frac{E_{ex}}{1 - E_{ex}} \quad (24)$$

and for the special case when  $T_o = T_c$ :

$$I_{ex} = \frac{E_{th}}{1 - E_{th}}. \quad (25)$$

### 3. RESULTS AND DISCUSSIONS

To obtain a numerical appreciation of the four efficiencies outlined above, calculations with different circulation ratios were made for fixed AHT configurations.

The circulation ratio,  $f$ , is defined as:

$$f = \frac{X_v - X_w}{X_s - X_w} = \frac{F_s}{F_v}. \quad (26)$$

Utilizing a flowsheeting program constructed for both design and evaluation calculations of arbitrarily complex absorption cycles [25], it was possible to simulate the different alternatives and plot the results.

The AHT system is assumed to be operated using  $H_2O$ - $NaOH$  as the working pair. Consider the two heat transformers A and B, shown in Fig. 1 and Fig. 2, respectively. They have identical configurations, except that heat transformer A has no solution heat exchanger.

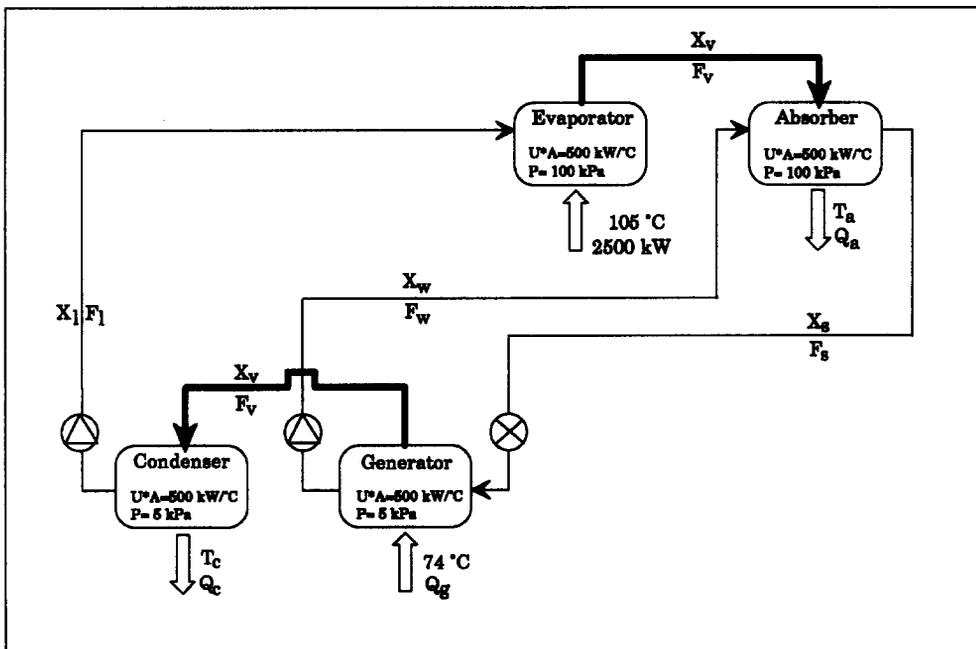


Fig. 1. Heat transformer A, without solution heat exchanger.

Both transformers were assumed to have the following constant data:

- Heat input to evaporator: ..... 2500 kW
- UA-value of evaporator, absorber, generator and condenser: ..... 500 kW/°C
- Temperature of steam to evaporator: ..... 105°C
- Temperature of steam to generator:..... 74°C
- Pressure in generator and condenser:..... 5 kPa

Moreover, the following constant data was assumed for transformer B:

- UA-value of solution heat exchanger: ..... 285 kW/°C.

It may be pointed out that the above information is the only input data required by the program to simulate both transformers. The simulation results are given in Figs 3–13 and Tables 1–4.

The four efficiencies COP,  $COP_{rev}$ ,  $E_{th}$  and  $I_{ex}$  are displayed for both cycles in Figs 3 and 4 as function of the circulation ratio  $f$ . Figure 3 reveals that both heat transformers would yield the same COP value, namely 0.48, in spite of the considerable difference in the circulation ratio of both configurations.

Heat transformer A has a circulation ratio  $f = 2.0$  and the corresponding value for heat transformer B is  $f = 12.5$ . The temperature lift, in terms of absorber temperature, is plotted in Fig. 5 as function of  $f$ . It can be seen that heat transformer A has a much lower temperature lift than heat transformer B. The absorber cooling temperature of heat transformer A is 111°C compared with 134°C for heat transformer B. Obviously, the COP is not sufficient to predict the considerable difference in performance of the two heat transformers.

Figure 3 indicates a rapid decrease in COP as the circulation ratio increases in the absence of a solution heat exchanger. For instance, if transformer A would resume the same circulation ratio as that of transformer B, i.e. 12.5, its COP value would deteriorate, by an order of magnitude, down to 0.046. As can be observed in Fig. 5, the change in circulation ratio gave rise to changes in the temperature lift. Consequently, heat transformer A achieves a higher temperature lift for higher circulation ratios, but its absorber has a very low energy output, only 120 kW, and its generator receives a very low energy input, only 80 kW.

Referring to Fig. 5, compared to heat transformer B, the higher temperature lift achieved by heat transformer A, in the circulation ratio range of  $f = 5$ –12.5, depends on a much lower heat

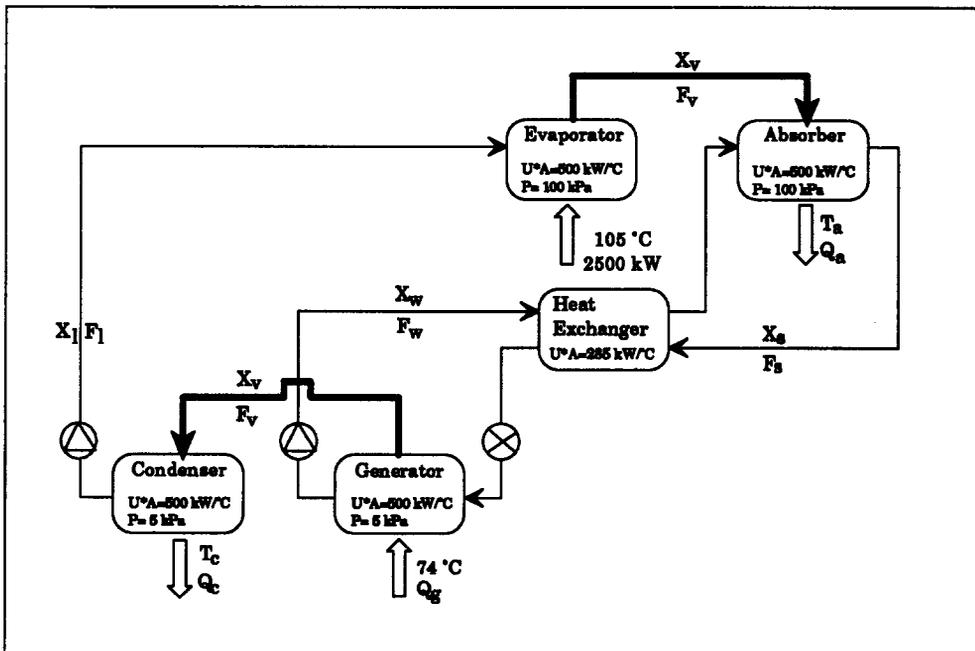


Fig. 2. Heat transformer B, with solution heat exchanger.

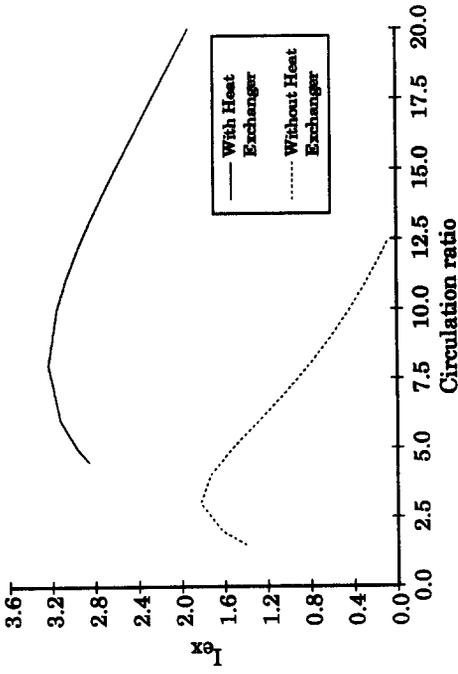


Fig. 4. Exergetic index for configurations A and B.

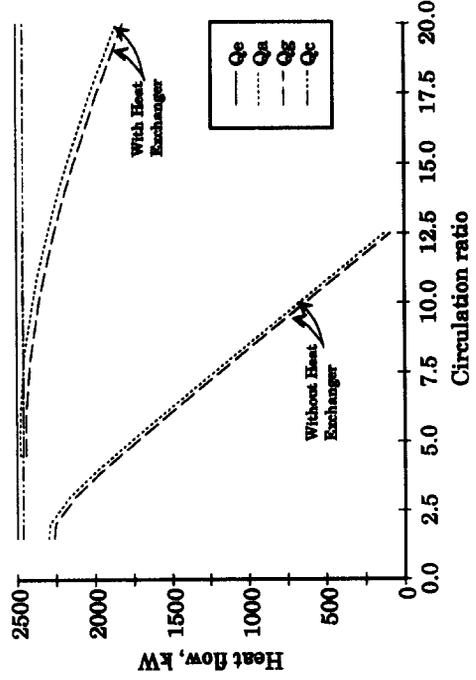


Fig. 6. Heat flows ( $Q$ ) as function of circulation ratio ( $f$ ).

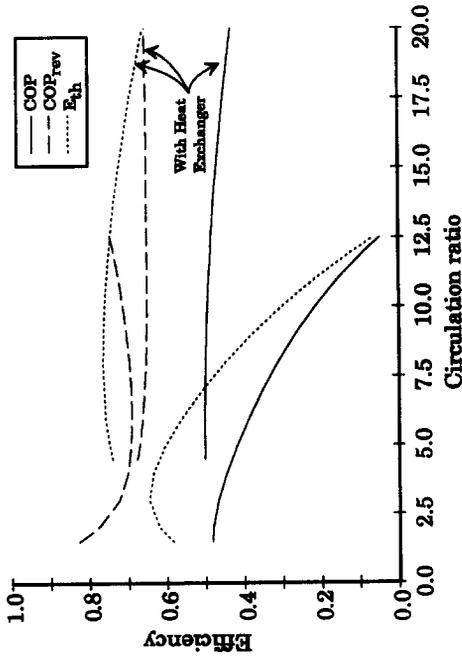


Fig. 3. Heat transformer efficiencies for configurations A and B.

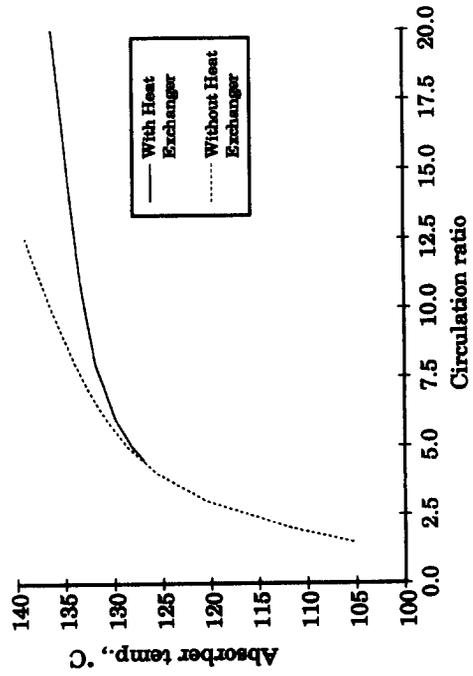


Fig. 5. Absorber temperature ( $T_a$ ) as function of circulation ratio ( $f$ ).

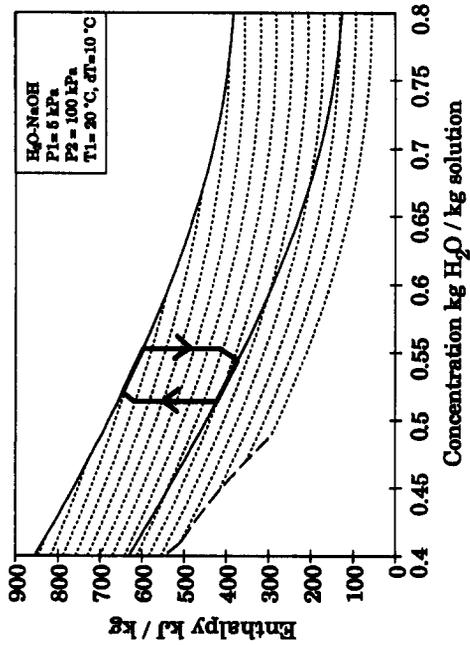


Fig. 8. Operation cycle for heat transformer B ( $f = 12.5$ ) with solution heat exchanger.

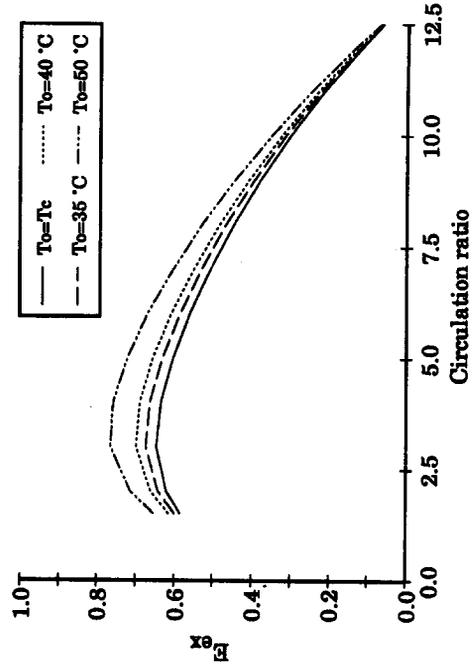


Fig. 10. Influence of the reference temperature ( $T_c$ ) on exergetic efficiency ( $E_a$ ) for heat transformer A.

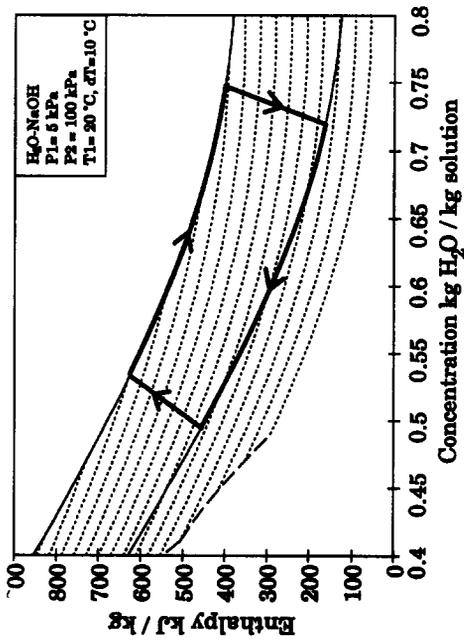


Fig. 7. Operation cycle for heat transformer A ( $f = 2.0$ ) without solution heat exchanger.

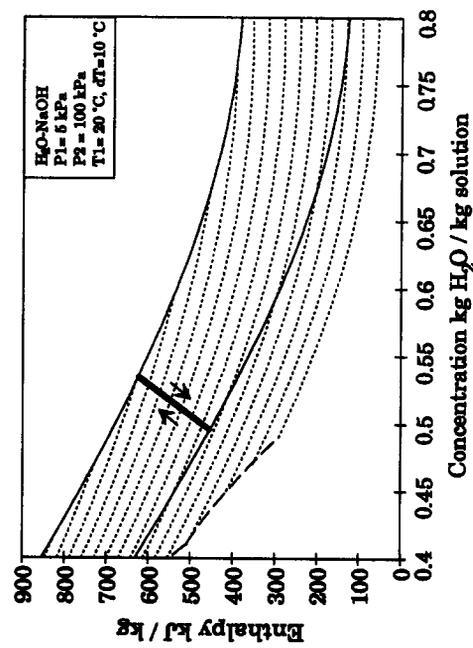


Fig. 9. Operation cycle for heat transformer A ( $f = 12.5$ ) without solution heat exchanger.

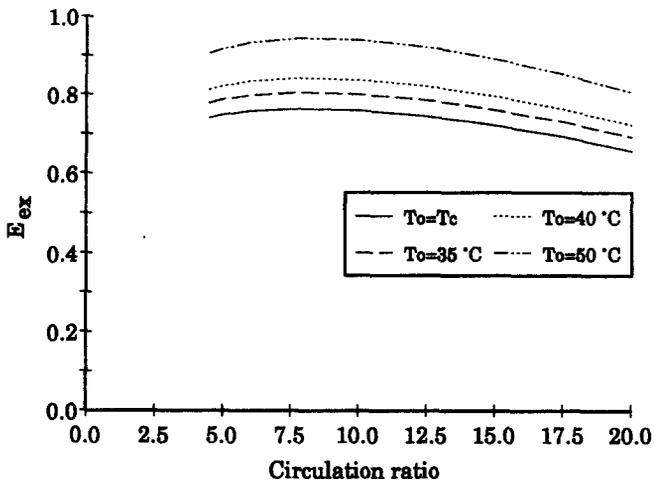


Fig. 11. Influence of the reference temperature ( $T_o$ ) on exergetic efficiency ( $E_{ex}$ ) for heat transformer B.

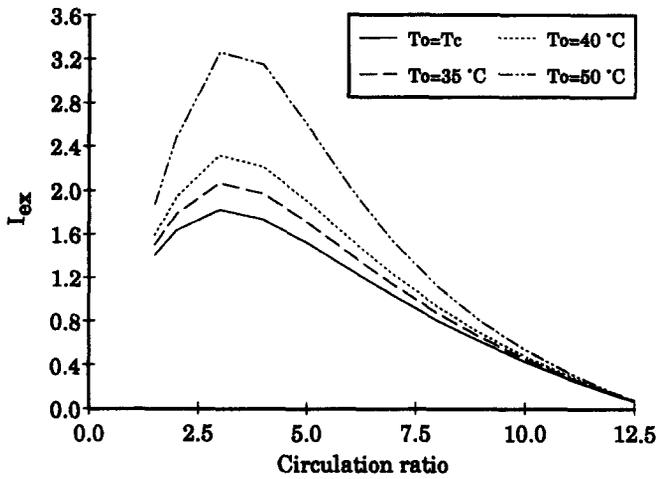


Fig. 12. Influence of the reference temperature ( $T_o$ ) on exergetic index ( $I_{ex}$ ) for heat transformer A.

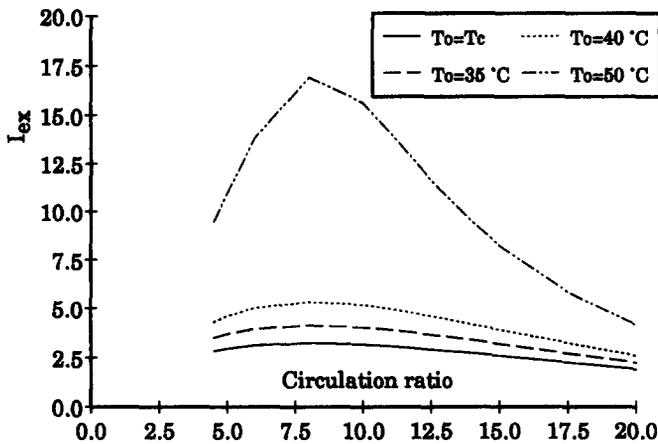


Fig. 13. Influence of the reference temperature ( $T_o$ ) on exergetic index ( $I_{ex}$ ) for heat transformer B.

Table 1. Comparison between different efficiencies for both heat transformers

Efficiency	$T_o$ (°C)	Heat transformer		
		A ( $f = 2.0$ )	B ( $f = 12.5$ )	A ( $f = 12.5$ )
COP		0.48	0.48	0.046
COP <sub>rev</sub>		0.78	0.65	0.74
$E_{th}$		0.62	0.74	0.062
$E_{ex}$	28	0.62	0.74	0.062
	50	0.71	0.92	0.071
$I_{ex}$	28	1.63	2.91	0.066
	50	2.48	11.6	0.076

Table 2. Simulation results for heat transformer A ( $f = 2.0$ ) without solution heat exchanger

Component	$Q$ (kW)	$T$ (°C)	Stream	$F$ (kg/s)	$X$ (kg/kg)
Evaporator	2500	105	Strong	1.97	0.747
Absorber	2300	111	Weak	0.98	0.495
Generator	2260	74	Vapour	0.98	1.0
Condenser	2460	28	Liquid	0.98	1.0

input to the generator of heat transformer A, as can be observed in Fig. 6. This results in a smaller logarithmic mean temperature difference and lower concentration in the generator and consequently a higher temperature lift in the absorber.

Figures 7 and 8 depict the operation cycle of heat transformers A with  $f = 2.0$  and B with  $f = 12.5$ , respectively. These cycles are simulated and plotted in enthalpy–concentration diagrams for  $H_2O$ – $NaOH$  as the working pair. As can be seen, the proportion between heat liberated, when vapour is condensed in the absorber to heat the solution to its boiling point, and heat transferred from the absorber is approximately the same in both cases. This gives the same COP value for both heat transformers, irrespective of the considerable difference in the temperature lift. The operation cycle of heat transformer A with  $f = 12.5$  is plotted in Fig. 9 on the same enthalpy–concentration diagram. As mentioned above, the main objective of this alternative is to increase the temperature lift by changing the circulation ratio from 2.0 to 12.5, without using a solution heat exchanger. It is obvious from Fig. 9 that practically all heat energy liberated from vapour condensation in the absorber is used to heat the solution to its boiling point, resulting in the very low COP value.

This indicates that the COP can be used to compare different heat transformers only when they are operated at the same circulation ratio. It obviously substantiates the effectiveness of heat exchange in and thermal insulation of a heat transformer for a fixed circulation ratio. However, the COP is unlikely to be an important criterion of performance, since the energy source is usually cheap or free in the form of process waste heat streams or in some cases solar or geothermal energy.

The Carnot efficiency, COP<sub>rev</sub>, takes into consideration the real heat inputs in the generator and evaporator and the temperature lift. For the three discussed operation cases, where heat transformer A has  $f = 2.0$  and  $f = 12.5$  and where heat transformer B has  $f = 12.5$ , the COP<sub>rev</sub> values 0.78, 0.74 and 0.65, respectively, were obtained. Like COP, the Carnot efficiency COP<sub>rev</sub> is not the correct way to describe the best cycle operation. A better criterion should take into consideration both the highest temperature attainable in the absorber, i.e. the temperature lift which can be achieved in the AHT, and the real output from the absorber.

The thermodynamic efficiency,  $E_{th}$ , is also shown in Fig. 3 for different circulation ratios. For the three discussed operation cases, the  $E_{th}$  values 0.62, 0.062 and 0.74, respectively, were obtained. The type of efficiency seems to be more logical, since it takes into consideration the temperature lift and the heat output from the absorber.

The exergetic efficiency,  $E_{ex}$ , offers a possibility to choose the reference temperature,  $T_o$ , at a level where heat energy is more or less worthless for the actual application. Choosing  $T_o$  to be equal to the temperature of the medium on the cooling side of the condenser, viz. 28°C, the exergetic efficiencies of the two heat transformers will be the same as the thermodynamic efficiency,  $E_{th}$ , as indicated by equation (9). The influence of the reference temperature on the exergetic efficiency is shown in Figs 10 and 11 where  $E_{ex}$  is plotted, for both heat transformers, as function of circulation ratio for different values of temperature  $T_o$ . If for some reason heat at a temperature of 50°C is considered economically worthless, and the reference temperature  $T_o$  retains this value, the

Table 3. Simulation results for heat transformer B ( $f = 12.5$ ) with solution heat exchanger

Component	$Q$ (kW)	$T$ (°C)	Stream	$F$ (kg/s)	$X$ (kg/kg)
Evaporator	2500	105	Strong	12.3	0.553
Absorber	2300	134	Weak	11.3	0.514
Generator	2260	74	Vapour	0.98	1.0
Condenser	2460	28	Liquid	0.98	1.0

Table 4. Simulation results for heat transformer A ( $f = 12.5$ ) without solution heat exchanger

Component	$Q$ (kW)	$T$ (°C)	Stream	$F$ (kg/s)	$X$ (kg/kg)
Evaporator	2500	105	Strong	12.3	0.535
Absorber	120	139	Weak	11.3	0.495
Generator	80	74	Vapour	0.98	1.0
Condenser	2460	28	Liquid	0.98	1.0

exergetic efficiencies would obtain the values 0.71, 0.071 and 0.92, respectively. As can be observed, the relative difference in these efficiency values became somewhat amplified. It can therefore be concluded that the exergetic efficiency,  $E_{ex}$ , can be considered as an alternative to the thermodynamic efficiency,  $E_{th}$ , but it offers a possibility to take into account any temperature level where heat energy may be considered worthless.

From Fig. 4, it can be seen that the relative difference between these two transformers becomes much more amplified when the exergetic index,  $I_{ex}$ , is used. For the three operation cases discussed above, the exergetic index values 1.63, 0.066 and 2.91, respectively, were obtained for  $T_o = T_c = 28^\circ\text{C}$ . The superiority of heat transformer B is very clearly indicated when  $I_{ex}$  is used. This is also illustrated in Figs 12 and 13 where the influence of the reference temperature  $T_o$  on the exergetic index is shown for both heat transformers. For instance, if reference temperature  $T_o = 50^\circ\text{C}$  is chosen, the corresponding values of  $I_{ex}$  for the three operation cases would increase to 2.48, 0.076 and 11.6, respectively. Again, the exergetic index  $I_{ex}$  reflects very clearly the considerable difference in the performance of both heat transformers.

In conclusion, the above discussion clearly demonstrated that the exergetic index should be considered as a more significant measure for evaluating the performance of the AHT systems, since it properly takes into account the exergy losses which inevitably occur in the system. It may however be stressed that exergy analysis should be used as a compliment to the First Law analysis.

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## REFERENCES

1. C. Trepp, History and prospects of heat transformation, *Int. J. Refrig.* **6**, 309–318 (1983).
2. E. C. Clark, Chemical heat pumps drive to upgrade waste heat, *Chem. Engng* **91**(4), 50–51 (1984).
3. F. Moser and H. Schnitzer, *Heat Pumps in Industry*, Elsevier, Amsterdam (1985).
4. H. Bokelmann and F. Steimle, Development of advanced heat transformers utilizing new working fluids, *Rev. Int. Froid* **9**, 51–59 (1986).
5. K. Eriksson and Å. Jernqvist, Heat transformers with self-circulation: Design and preliminary operational data, *Int. J. Refrig.* **12**, 15–20 (1989).
6. K. Abrahamsson, G. Aly and Å. Jernqvist, Heat transformer systems for evaporation applications in the pulp and paper industry, *Nordic Pulp Paper Res.* **6**, 9–16 (1992).
7. G. Aly, K. Abrahamsson and Å. Jernqvist, Application of absorption heat transformers for energy conservation in the oleochemical industry, Submitted for publication in *J. Am. Oil Chem. Soc.* (1992).
8. P. E. Scheihing and L. A. Cuervo, Market opportunities of industrial heat pumps in the United States, *IEA Heat Pump Centre Newsletter* **8**(4), 16–19 (1990).
9. T. A. Brzustowski and P. J. Golem, Second Law analysis of energy processes, Part I: Exergy—an introduction, *Trans. Can. Soc. Mech. Engrs* **4**, 209–218 (1976–1977).
10. P. J. Golem and T. A. Brzustowski, Second Law analysis of energy processes, Part II: The performance of simple heat exchangers, *Trans. Can. Soc. Mech. Engrs* **4**, 219–226 (1976–1977).
11. T. J. Moran, *Availability Analysis: A Guide to Efficient Energy Use*, Prentice Hall (1982).
12. A. L. London, Economics and the Second Law: an engineering view and methodology, *Int. J. Heat Mass Transfer* **25**, 743–751 (1982).
13. T. J. Kotas, *The Exergy Method of Thermal Plant Analysis*, Butterworths (1985).
14. K. P. Tyagi, Second Law analysis of  $\text{NH}_3$ – $\text{NaSCN}$  absorption refrigeration cycle, *Heat Recovery Systems & CHP* **6**, 73–82 (1986).
15. G. Alefeld, Efficiency of compressor heat pumps and refrigeration derived from the Second Law, *Int. J. Refrig.* **10**, 331–341 (1987).
16. G. Alefeld, Probleme mit der exergie, *BWK* **40**, 72–80 (1988).
17. G. Alefeld, Exergy and the Second Law of Thermodynamics, *BWK* **40**, 458–464 (1988).
18. G. Alefeld, Problems with the exergy concept (or the missing Second Law), *IEA Heat Pump Centre Newsletter* **6**, 19–23 (1988).
19. W. van Gool and A. J. Hoogendoorn, Heat pumps and exergy analysis, *Proceedings 3rd International Energy Agency Heat Pump Conference*, Tokyo, pp. 639–646 (1990).
20. C. B. Colenbrander, J. F. van der Horst and A. J. Meijnen, Thermoeconomic analysis of heat transformers, *Koeltechniek* **9**, 16–20 (1988).
21. S. Kumar, M. Prevost and R. Bugarel, Exergy analysis of a compression refrigeration system, *Heat Recovery Systems & CHP* **9**, 151–157 (1989).
22. S. I. Pereira Duarte and R. Bugarel, Optimal working conditions for an absorption heat transformer—analysis of the  $\text{H}_2\text{O}$ – $\text{LiBr}$  theoretical cycle, *Heat Recovery Systems & CHP* **9**, 521–532 (1989).
23. K. Stephan, Absorption heat transformer cycles, *Heat Pump Fundamentals*, Ed. Berghmans, J., *NATO ASI Series, Series E: Applied Sciences*, No. 53, 352–373 (1983).
24. J. W. J. Bouma, Experience with a heat transformer in the chemical industry, *IEA Heat Pump Centre Newsletter* **8**(4), 12–15 (1990).
25. K. Abrahamsson and Å. Jernqvist, Modeling and simulation of absorption heat pump cycles, Submitted for publication in *Comput. Chem. Engng* (1992).