

# PERFORMANCE THERMODYNAMIC CRITERIA OF THE ABSORPTION HEAT TRANSFORMERS

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**Rezumat.** Coeficientul de performanță COP este adesea folosit pentru a evalua performanțele sistemelor cu absorbție. Acest parametru este definit pe baza primului principiu al termodinamicii și nu ține seama de calitatea energiei consumate și produse. În scopul de a lua în considerare calitatea energiei, a fost utilizat principiul al II-lea al termodinamicii, ceea ce a permis stabilirea unor criterii de performanță incluzând eficiența energetică, termodinamică și exergetică. Se arată că eficiența termodinamică  $E_{th}$  reprezintă o măsură mai logică a eficienței sistemelor cu absorbție deoarece ia în calcul căldura reală introdusă la generator. Eficiența exergetică  $\eta_{ex}$  poate fi considerată drept o alternativă la eficiența energetică dar oferă posibilitatea de a lua în considerare orice nivel de temperatură la care la care căldura ar putea fi considerată nefolosibilă. Pe de altă parte, indicele exergetic  $I_{ex}$  este în directă legătură cu  $\eta_{ex}$ . Dar valoarea lui numerică ar trebui considerată ca o măsură mai semnificativă în evaluarea performanțelor sistemelor cu absorbție deoarece ia în calcul în mod corect pierderile exergetice care intervin în mod inevitabil în sistem. Potențialul de îmbunătățire este o măsură a cât de mult și cât de ușor poate fi îmbunătățit sistemul în scopul optimizării. De asemenea, capacitatea de pompare a căldurii primite de la surse secundare de căldură reprezintă un factor foarte important în aprecierea transformatoarelor de căldură cu absorbție. În această lucrare sunt prezentate relații matematice de definire a acestor criterii de performanță precum și relații de legătură între ele.

## 1. INTRODUCTION

The aim of this work is to investigate and suggest improvements to the methods presently available for the recuperation of industrial waste heat. Many industrial installations such as agro-feed, paper mills, dairies, petroleum refineries, textile mills, power generating stations and processing industries reject heat to the environment at a pretty temperature and, consequently, local thermal pollution can occur if the wasted heat is not recovered immediately. Recovered waste heat can directly reduce the energy cost per unit of industrial product and the size requirement and energy conversion devices. Thermal pollution of the atmosphere can also be reduced as a by-product of waste heat recovery.

Among the different systems currently adopted for the recuperation of industrial wasted heat, absorption plants are considered to be the most competitive. There are three types of absorption plants: refrigerating plant, heat pump and heat transformer [1,2,3]. Refrigerating plants and heat pumps permit considerable savings in primary energy by utilizing low-quality heat sources to release cooling and heating loads, respectively. Heat transformers ensure the recovering energy available from waste heat sources to provide temperatures above these from the heat sources itself. Although the principle of operation of the AHT was already established by Altenkirch [4] in 1913, this idea was developed later, by Nesselmann [5] who, in 1933 published a detailed paper on its different applications. In 1982 Wilkinson [6] proposed different types of single and multi-stage absorption heat transformers. A comparative study of different working fluid combinations with R22 as refrigerant and six absorbents in a single AHT was performed by Fatouh and Srinivasain 1992, [7]. A new

type of AHT operating with reverse rectification with water-glycol and ammonia-water mixtures was presented by Le Goff in 1992 [8, 9]. In the same year, Rivero and Le Goff [10] described and compared the different performance criteria available for analyzing heat pump and heat transformers

Experimental work using an ammonia-water heat transformer has been reported by Mostofzadeh [11]. Stephan and Seher [12, 13] have discussed the heat transformer cycles for single and double-stage processes. Kripalani *et al.* [6,7] have studied the performance analysis of a vapour absorption heat transformer with different working fluid combinations. Further it was reported by Kripalani *et al.* [14], and by Porneala in 1989, [15], that the transformation possibilities with a single stage process with aqua-ammonia are quite limited and probably uneconomical. The present state of new technology for heat transformers has been discussed by Rojey *et al.* [16]. Up to now only limited information is available. Porneala studied the heating load optimization of the absorption heat transformer as an endoreversible system [17]

## 2. DESCRIPTION OF THE SINGLE ABSORPTION HEAT TRANSFORMER

The absorption heat transformer is shown schematically in Fig.1a, wherein the thermodynamic state points of working fluid are numbered. The generator G and the evaporator E are supplied with waste heat at the same medium temperature. The absorber Ab delivers heat at a higher delivery temperature, whereas part of heat flowing into the process is rejected at ambient temperature from the condenser K. The pressure at the generator corresponds to the saturation pressure of the

pure refrigerant at the condensing temperature ( $p_G = p_K$ ), while the pressure at the absorber corresponds to the saturation pressure at the evaporating temperature ( $p_{Ab} = p_E$ ). Thus, the absorber and evaporator operate at higher pressure than generator and condenser. Absorption of refrigerant vapour at high pressure releases heat at higher temperature at the absorber. Condenser K, pump P2 and evaporator E behave as an *thermo-compressor* unit (TC) that ensures the increase of vapour pressure from  $p_K$  to  $p_E$ . Generator G, absorber Ab,

pump P1, expansion valve EV and regenerative heat exchanger RHE form a *thermodetentor* unit (TD). The thermocompressor Tc produces the exergy flux  $\dot{E}x_{TC}$ , that, together with the exergy flux  $\dot{E}x_{TC}$  ensures the pumping of the heat flux  $\Phi_{Ab}$  from the temperature  $t_{wh}$  to the higher temperature  $t_u$ . Figures 1band 1c show the diagrams of the energetical and the exergetical fluxes. The thermodetentor TD releases the temperature lift  $\Delta t_p$  (pumping the heat flux  $\Phi_{Ab}$ )

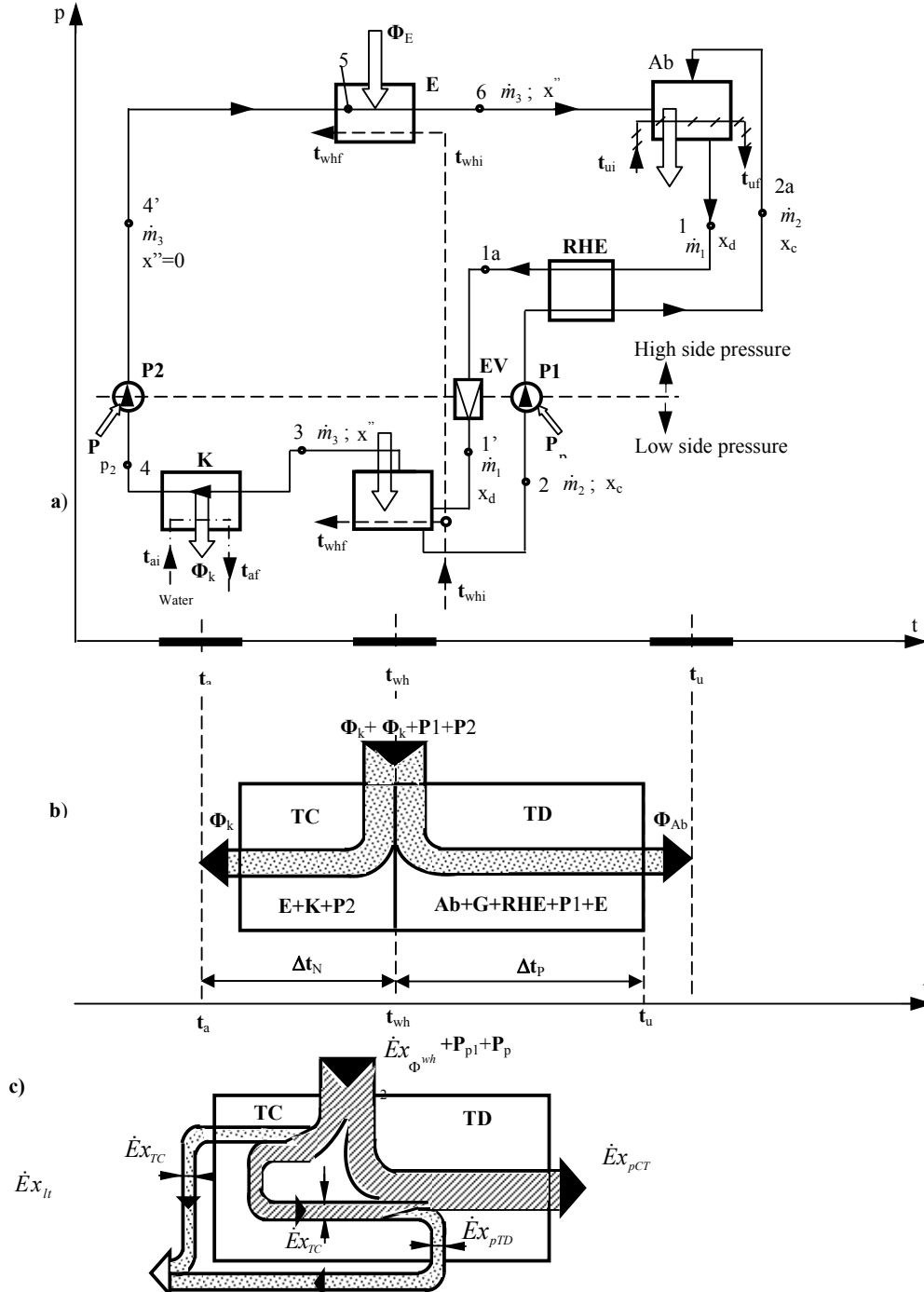


Fig. 1. Single absorption heat transformer.

### 3. DIFFERENT EFFICIENCIES FOR ABSORPTION HEAT TRANSFORMERS

#### 3.1. Carnot efficiency, COPREV

The maximum value of the coefficient of performance, COP, is obtained for ideal reversible cycle operating at the same specified conditions as real cycle, and is commonly referred to as the Carnot efficiency, COP rev, or the Carnot COP. Using the entropy and the enthalpy balances, Jerqvist [6,11] was established the expression for the COPrev in the case of the absorption heat pump. In this paper was established COP rev for absorption heat transformer, by using two different methods:

**I. Exergy balances.** The coefficient of performance for a reversible process, assuming  $P_{P1} = P_{P2} = 0$ :

$$COP_{rev} = \frac{\Phi_{Ab}}{\Phi_E + \Phi_G} \quad (1)$$

The exergy introduced in evaporator and generator is equal with the exergy delivered by the absorber:

$$\dot{E}x_{\Phi_E} + \dot{E}x_{\Phi_G} = \dot{E}x_{\Phi_{Ab}} \quad (2)$$

$$\Phi_E \cdot \left(1 - \frac{T_K}{T_E}\right) + \Phi_G \cdot \left(1 - \frac{T_K}{T_G}\right) = \Phi_{Ab} \cdot \left(1 - \frac{T_K}{T_{Ab}}\right) \quad (3)$$

$$\Phi_{Ab} + \Phi_K = \Phi_E + \Phi_G \Rightarrow \Phi_E = \Phi_{Ab} + \Phi_K - \Phi_G \quad (4)$$

From (3) and (4) results:

$$\begin{aligned} & (\Phi_{Ab} + \Phi_K - \Phi_G) \cdot \left(1 - \frac{T_K}{T_E}\right) + \\ & + \Phi_G \cdot \left(1 - \frac{T_K}{T_G}\right) = \Phi_{Ab} \cdot \left(1 - \frac{T_K}{T_{Ab}}\right) \end{aligned} \quad (5)$$

$$\Phi_K = \frac{\Phi_{Ab} \cdot \left(\frac{T_K}{T_E} - \frac{T_K}{T_{Ab}}\right) - \Phi_G \cdot \left(\frac{T_K}{T_E} - \frac{T_K}{T_G}\right)}{1 - \frac{T_K}{T_E}} \quad (6)$$

$$\Phi_K + \Phi_{Ab} = \frac{\Phi_{Ab} \left(1 - \frac{T_K}{T_{Ab}}\right) - \Phi_G \left(\frac{T_K}{T_E} - \frac{T_K}{T_G}\right)}{1 - \frac{T_K}{T_E}} \quad (7)$$

$$\begin{aligned} COP_{rev} &= \frac{\Phi_{Ab} \left(1 - \frac{T_K}{T_E}\right)}{\Phi_{Ab} \left(1 - \frac{T_K}{T_{Ab}}\right) - \Phi_G \left(\frac{T_K}{T_E} - \frac{T_K}{T_G}\right)} = \\ &= \frac{1 - \frac{T_K}{T_E}}{1 - \frac{T_K}{T_{Ab}} - \frac{\Phi_G}{\Phi_{Ab}} \cdot \left(\frac{T_K}{T_E} - \frac{T_K}{T_G}\right)} \end{aligned} \quad (8)$$

If  $T_G = T_E$  results:

$$COP_{rev} = \frac{T_E - T_K}{T_E} \cdot \frac{T_{Ab}}{T_{Ab} - T_K} \quad (9)$$

**II. Using the powers.** The thermocompressor TC (Fig.1a) produces the power  $P_D$ . To release the cooling

cycle from the thermodetentor TD, the power  $P_F$  must be used. There are the following expressions;

$$P_D = P_F \quad (10)$$

$$P_D = \Phi_E - \Phi_K \quad (11)$$

$$P_F = \Phi_{Ab} - \Phi_G \quad (12)$$

$$P_D = \dot{E}x_{\Phi_E} - \dot{E}x_{\Phi_K} = \Phi_E \cdot \left(1 - \frac{T_K}{T_E}\right) - \Phi_K \cdot \left(1 - \frac{T_K}{T_K}\right) \quad (13)$$

$$\begin{aligned} P_F &= \dot{E}x_{\Phi_{Ab}} - \dot{E}x_{\Phi_G} = \\ &= \Phi_{Ab} \cdot \left(1 - \frac{T_K}{T_{Ab}}\right) - \Phi_G \cdot \left(1 - \frac{T_K}{T_G}\right) \end{aligned} \quad (14)$$

$$\begin{aligned} \Phi_E - \Phi_K - \Phi_E \cdot \frac{T_K}{T_E} + \Phi_K \cdot \frac{T_K}{T_K} &= \\ = \Phi_{Ab} - \Phi_G - \Phi_{Ab} \cdot \frac{T_K}{T_{Ab}} + \Phi_G \cdot \frac{T_K}{T_G} \end{aligned} \quad (15)$$

$$-\Phi_E \cdot \frac{T_K}{T_E} + \Phi_K \cdot \frac{T_K}{T_K} = -\Phi_{Ab} \cdot \frac{T_K}{T_{Ab}} + \Phi_G \cdot \frac{T_K}{T_G} \quad (16)$$

By substitution:

$$\Phi_E = \Phi_{Ab} + \Phi_K - \Phi_G \quad (17)$$

results:

$$\begin{aligned} -(\Phi_{Ab} + \Phi_K - \Phi_G) \cdot \frac{T_K}{T_E} + \Phi_K \cdot \frac{T_K}{T_K} &= \\ = -\Phi_{Ab} \cdot \frac{T_K}{T_{Ab}} + \Phi_G \cdot \frac{T_K}{T_G} \end{aligned} \quad (18)$$

$$\Phi_G \left(\frac{T_K}{T_E} - \frac{T_K}{T_G}\right) + \Phi_K \left(\frac{T_K}{T_K} - \frac{T_K}{T_E}\right) = \quad (19)$$

$$= \Phi_{Ab} \left(\frac{T_K}{T_E} - \frac{T_K}{T_{Ab}}\right)$$

$$\Phi_K = \frac{\Phi_{Ab} \left(\frac{1}{T_E} - \frac{1}{T_{Ab}}\right) - \Phi_G \left(\frac{1}{T_E} - \frac{1}{T_G}\right)}{\frac{1}{T_K} - \frac{1}{T_E}} \quad (20)$$

By substitution  $\Phi_K$  from equation (20), results the same expression for the COPrev:

$$\begin{aligned} COP_{rev} &= \frac{\Phi_{Ab}}{\Phi_{Ab} + \Phi_K} = \\ &= \frac{\Phi_{Ab} \left(\frac{1}{T_K} - \frac{1}{T_E}\right)}{\Phi_{Ab} \left(\frac{1}{T_K} - \frac{1}{T_{Ab}}\right) - \Phi_G \left(\frac{1}{T_E} - \frac{1}{T_G}\right)} \end{aligned} \quad (21)$$

#### 3.2. Thermodynamic efficiency, $E_{TH}$

Consider the thermodynamic efficiency to be defined according to [18]:

$$E_{th} = \frac{COP}{COP_{rev}} \quad (22)$$

Assuming that heat energy is transferred to and from the absorption heat transformer at constant temperatures  $T_E$ ,  $T_G$ ,  $T_K$  and  $T_{Ab}$ , the thermodynamic efficiency can then be expressed as:

$$E_{th} = \frac{\Phi_{Ab} \cdot \left( \frac{1}{T_K} - \frac{1}{T_{Ab}} \right)}{\Phi_G \cdot \left( \frac{1}{T_K} - \frac{1}{T_G} \right) + \Phi_E \cdot \left( \frac{1}{T_K} - \frac{1}{T_E} \right)} \quad (23)$$

This equation relates  $E_{th}$  to real energy input to the generator and evaporator, real energy delivered from the condenser and absorber, and the four temperatures  $T_G$ ,  $T_E$ ,  $T_K$  and  $T_{Ab}$ . When the energy is introduced in generator and evaporator at the same temperature,  $T_E = T_G$ , the expression for the thermodynamic efficiency can be further simplified to:

$$\begin{aligned} E_{th} &= \frac{\Phi_{Ab} \cdot \left( \frac{1}{T_K} - \frac{1}{T_{Ab}} \right)}{(\Phi_G + \Phi_E) \cdot \left( \frac{1}{T_K} - \frac{1}{T_E} \right)} = \\ &= COP \cdot \frac{T_{Ab} - T_K}{T_{Ab}} \cdot \frac{T_E}{T_E - T_K} = \frac{COP}{COP_{rev}} \end{aligned} \quad (24)$$

### 3.3. Exergetic efficiency, $\eta_{ex}$

The exergetic efficiency,  $\eta_{ex}$ , is defined as [19]:

$$\begin{aligned} \eta_{ex} &= \frac{\text{revalorised exergy}}{\text{consumed exergy}} = \frac{\dot{E}x_{Ab}}{\dot{E}x_E + \dot{E}x_G + P_{P1} + P_{P2}} \quad (25) \\ \eta_{ex} &= \frac{\Phi_{Ab} \cdot \left( 1 - \frac{T_o}{T_{Ab}} \right)}{\Phi_E \cdot \left( 1 - \frac{T_o}{T_E} \right) + \Phi_G \cdot \left( 1 - \frac{T_o}{T_G} \right) + P_{P1} + P_{P2}} \end{aligned} \quad (26)$$

where  $T_o$  is the environmental temperature

Assuming that heat energy is transferred to and from the absorption heat transformer at constant temperatures  $T_E$ ,  $T_G$ ,  $T_K$  and  $T_{Ab}$ , neglecting  $P_{P1}$  and  $P_{P2}$ , the thermodynamic efficiency can then be expressed as:

$$\eta_{ex} = \frac{\Phi_{Ab} \cdot \left( \frac{1}{T_o} - \frac{1}{T_{Ab}} \right)}{\Phi_G \cdot \left( \frac{1}{T_o} - \frac{1}{T_G} \right) + \Phi_E \cdot \left( \frac{1}{T_o} - \frac{1}{T_E} \right)} \quad (27)$$

It may be observed that equations (23) and (27) become similar if  $T_K = T_o$ . Consequently, the exergetic efficiency  $\eta_{ex}$  is a function of the same process variables as those for the thermodynamic efficiency  $E_{th}$ . When the energy is introduced in generator and evaporator at the same temperature,  $T_E = T_G$ , the above equation can be simplified to:

$$\eta_{ex} = COP \cdot \frac{T_{Ab} - T_K}{T_{Ab}} \cdot \frac{T_E}{T_E - T_K} = E_{th} = \frac{COP}{COP_{rev}} \quad (28)$$

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 differences and not in the absolute values, this limitation should not be considered too serious

### 3.4. Exergy index, $I_{ex}$

Considered as indicator of thermodynamic efficiency, the numerical value of both  $E_{th}$  and  $\eta_{ex}$  do not really reflect the importance of exergetic losses in the absorption cycle. The exergetic index, first proposed by Pereira Duarte and Bugarel [19], is more adequate indicator of efficiency for these systems. Neglecting the work done by the pumps, the exergy index,  $I_{ex}$ , is defined as ratio between the rate of exergy produced in the absorber and the exergy lost or irreversibility, IR, and can thus be written as follows [18]:

$$I_{ex} = \frac{\Phi_{Ab} \cdot \left( \frac{1}{T_o} - \frac{\Delta s_{Ab}}{\Delta i_{Ab}} \right)}{IR} \quad (29)$$

Using the Second Law of thermodynamics, the irreversibility IR 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 absorption heat transformer in the case when  $T_K = T_o$ , IR can be expressed as:

$$\begin{aligned} IR &= \Phi_G \cdot \left( \frac{1}{T_o} - \frac{\Delta s_G}{\Delta i_G} \right) + \Phi_E \cdot \left( \frac{1}{T_o} - \frac{\Delta s_E}{\Delta i_E} \right) - \\ &- \Phi_{Ab} \cdot \left( \frac{1}{T_o} - \frac{\Delta s_{Ab}}{\Delta i_{Ab}} \right) \end{aligned} \quad (30)$$

Assuming that energy is transferred within the heat transformer system at constant temperatures  $T_{Ab}$ ,  $T_G$  and  $T_E$ , the exergy index can be written as follows:

$$I_{ex} = \frac{\Phi_{Ab} \cdot \left( \frac{1}{T_o} - \frac{1}{T_{Ab}} \right)}{\Phi_G \cdot \left( \frac{1}{T_o} - \frac{1}{T_G} \right) + \Phi_E \cdot \left( \frac{1}{T_o} - \frac{1}{T_E} \right) - \Phi_{Ab} \cdot \left( \frac{1}{T_o} - \frac{1}{T_{Ab}} \right)} \quad (31)$$

$$I_{ex} = \frac{\eta_{ex}}{1 - \eta_{ex}} \quad (32)$$

If heat energy is introduced in the generator and evaporator at the same temperature,  $T_G = T_E$ , the above equation can be rearranged to give the relationship between  $I_{ex}$  and COP:

$$I_{ex} = \frac{1}{COP \cdot \frac{T_E - T_o}{T_E} \cdot \frac{T_{Ab}}{T_{Ab} - T} - 1} \quad (33)$$

For the special case  $T_o = T_K$

$$I_{ex} = \frac{E_{th}}{1 - E_{th}} \quad (34)$$

The importance of introducing  $I_{ex}$  will become clear from the following example. If the  $\eta_{ex}$  of a given system is 0,9 at a particular time and is later reduced to 0,75, the exergetic loss shown by 0,15 units does not appear very serious. However, if the analysis is done using the exergetic index for which the respective values are 9 and 3, it clearly indicates that there has been a threefold exergy destruction. However, it should be kept in mind that this is particularly true for high exergetic efficiencies

### 3.5. Effluent exergy losses

The Effluent Exergy Losses are simply the sum of all exergy streams rejected to the environment. For the three types of the absorption systems there are:

$$Efl_{AHP} = 0; Efl_{ARP} = \dot{E}x_K + \dot{E}x_{Ab}; Efl_{AHT} = \dot{E}x_K \quad (35)$$

Anytime it is necessary to evaluate the impact on the environment of the rejection of exergy streams, the Effluent Exergy Losses should be evaluated separately. This partition of exergy losses is very important in system's optimization and reflects the ecologic implications of the exergy concept.

### 3.6. Improvement potential

The Improvement Potential (or thermodynamic imperfection) of a system is a measure of how much and how easily the system could be improved for optimization purpose and it is obtained from the Exergy Losses and the Effectiveness of the system. This potential is made up of three contributions: the relative potential, the absolute potential and the environmental potential. The relative potential is a measure of how easily a system could be improved; if the effectiveness is very low, the relative potential approaches its maximum value so, in principle, the system should be able to be improved easily. The absolute potential (IR) is a measure of how much a system can improve inside itself. The environmental potential (Efl) is a measure of how much a system can be improved by using the effluent streams instead of rejecting their exergy to the surrounding medium.

## 4. THERMAL CALCULUS OF THE ABSORPTION HEAT TRANSFORMER

By applying an original calculation method, the maximum number of distinct equations of energetic and material balances can be determined by formula [2]:

$$n_t = 2 \cdot n_2 + n_e + s \quad (36)$$

where:  $n_2$  is the number of distinct nodes of order 2,  $n_e$  is the number of energetic nodes and  $s$  is the number of components with energetic exchange through the surface. In this case (Fig. 1):  $n_2 = 1$  (Ab or G),  $n_e = 2$  (Ab

and G),  $s = 3$  (E, K and RHE). We consider:  $P_{p1} = P_{p2} = 0$ ,  $x'' = 0$ ;  $i_{1a} = i_1'$ ,  $i_4' = i_4$ ,  $i_2' = i_2$  and  $\Phi_{Ab}$  is a known value. The total number of distinct equations is:

$$n_t = 2 + 2 + 3 = 7 \text{ eqs.}$$

The following calculus chart can be used:

$$\left. \begin{aligned} \dot{m}_1 &= \dot{m}_2 + \dot{m}_3 && m.b.eq.Ab \\ \dot{m}_1 \cdot x_d &= \dot{m}_2 \cdot x_c && m.b.LiBr.eq.Ab \\ \Phi_{Ab} &= \dot{m}_3 \cdot i_6 + \dot{m}_2 \cdot i_{2a} - \dot{m}_1 \cdot i_1 && en.b.Ab \\ \dot{m}_1 \cdot (i_1 - i_{1a}) &= \dot{m}_2 \cdot (i_{2a} - i_2) && en.b.RHE \end{aligned} \right\} \Rightarrow \dot{m}_1, \dot{m}_2, \dot{m}_3, i_{2a} \quad (37)$$

$$\Phi_G = \dot{m}_3 \cdot i_3 + \dot{m}_2 \cdot i_2 - \dot{m}_1 \cdot i_1, \text{ en. b. G} \Rightarrow \Phi_G \quad (38)$$

$$\Phi_E = \dot{m}_3 (i_6 - i_4), \text{ en. b. E} \Rightarrow \Phi_E \quad (39)$$

$$\Phi_K = \dot{m}_3 (i_3 - i_4), \text{ en. b. K} \Rightarrow \Phi_K \quad (40)$$

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## ASOCIAȚIA GENERALĂ A INGINERILOR DIN ROMÂNIA

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