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IRREVERSIBILITIES IN TWO CONFIGURATIONS OF THE DOUBLE GENERATOR ABSORPTION CHILLER
Comparison of performance

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This paper deals with the modeling, thermodynamic analysis and comparison of irreversibility in two configurations of the double generator absorption chiller. First a computer simulation model is developed for each configuration on the basis of mass and energy balances. Simulation results were then used to analyse the entropy generation and irreversibility (or exergy destruction) of each component. It is found that the parallel flow configuration is more powerful than the serie flow configuration. Exergy losses in the parallel flow configuration are lower than that of the serie flow. The results indicate that the absorber and the solution heat exchangers have the most potential to improve the chiller energy efficiency. Also they indicate that focusing on irreversibility is more direct way of analysing the potential for improving the efficiency of ammonia-water absorption chiller.

Keywords: *ammonia-water, double effect absorption chiller, exergy, irreversibility, modeling, second law, simulation, thermodynamic analysis*

Introduction

Absorption refrigerators and heat pumps have been for a long time limited to very specific and marginal uses [1] because of their low *COP* compared to that of vapor compression machines. Over the last few decades however, the field has experienced a resurgence of interest. Unfortunately the *COP*s of simple effect cycles are low in comparison with those of vapor compression cycles even when the differential cost for heat and electric power is considered [2]. This means that the energetic advantage of this kind of chillers remains insignificant unless enhanced structures are developed. This is why the research interest shifted to multiple effect structures designed by combining single stage components and cycles [2, 3].

Usually, to specify the thermal efficiency of absorption chiller is to provide the coefficient of performance *COP*, based on the first law of thermodynamics. This parameter, however, makes no reference to the best possible performance and gives no information regarding where the irreversibilities in the chiller occur. Also, it cannot be used to determine the contribution of each component of the chiller to the overall efficiency. The second law analysis addresses the energy and the entropy balances for the system. It can be used to determine the entropy generation and the irreversibility (or exergy destruction) in each component. Since the COP reflects the amount of destruction of the available energy in the system, a second law analysis gives information on the potential of improvement in each component of the machine [4–6].

The present work is a part of the investigations of our research unit on absorption chilling and solar refrigeration [7–9]. It deals with a particular structure: the double effect double generator chiller operating with ammonia/water mixture.

Its purpose is the comparison of performance and irreversibilities in two configurations of an ammonia–water double generator absorption chiller.

Description of the studied configurations

Figure 1 shows schematically a parallel flow double generator configuration. Such a machine is composed essentially of two condensers, an evaporator, two vapor generators each one provided with a boiler and distillation column, an absorber, four expansion valves, two pumps and an adjustable three-way valve. The vapor (3) flowing from the evaporator to the absorber, is absorbed by the weak solution (9). The absorption heat is rejected towards the environment (cooling water or air) which also receives the energy released by the first condenser. The strong solution (4) is distributed, by an adjustable three-way valve, between the two stages of the machine comprising each a generator, a condenser and a solution heat exchanger. A fraction α of the strong solution (4)

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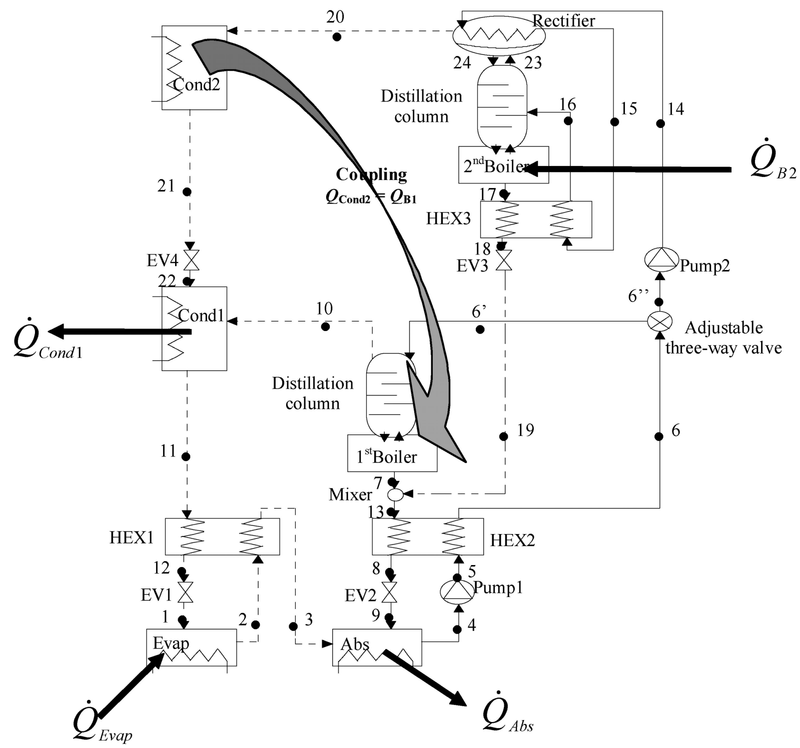

Fig. 1 Parallel flow double generator absorption chiller

is sent to the first generator. The separation of the refrigerant is performed at two temperature levels: the energy needed in the first boiler is supplied by the second condenser (Cond2). Large temperature differences between heat exchanging streams are so avoided. The first condenser (Cond1) receives the throtted condensate from the second condenser and the refrigerant vapor from the distillation column over the first boiler. The strong solution (14) supplying the second generator is a cold source for the distillation column over the 2nd boiler and determines thus the purity limit of refrigerant (23) leaving the last stage of this column. For a rational use of the energy supplied, three counter-current heat exchangers (HEX1, HEX2, HEX3) are used.

For the parallel flow double generator absorption chiller (Fig. 1), the strong solution (6) is distributed between the two generators. When the totality of this solution feeds the first generator and the weak solution of the first stage (7) feeds the second generator, a new configuration of the double generator absorption chiller is obtained, where the two generators are assembled in series as represented in Fig. 2.

Mathematical modeling

The formulation of the simulation model for the various studied configurations chiller proceeds by the following steps:

- determination of the chiller variance (number degrees of freedom)
- specification of the fundamental operating conditions: the cooling capacity, the driving heat source temperature, the useful cold temperature and that of the environment
- formulation of the mass and energy balances governing the various chiller components
- characterisation of the heat transfer in the various heat exchangers (pinch method)

To solve the large set of nonlinear equations of the simulation model the program CONLES, available as a FORTRAN code [10], is used. The fluid thermodynamic properties are calculated in a subroutine incorporated in the program.

Thermodynamic analysis

Components analysis

Neglecting the power input to the 2 pumps the governing equations used to evaluate the irreversibility in each component are:

Overall mass balance:

$$\sum_{\mathrm{e}} \dot{m}_{\mathrm{e}} = \sum_{\mathrm{i}} \dot{m}_{\mathrm{i}} \tag{1}$$

Energy balance:

$$\dot{Q}_{\mathrm{K}} = \sum_{\mathrm{e}} \dot{m}_{\mathrm{e}} h_{e} - \sum_{\mathrm{i}} \dot{m}_{\mathrm{i}} h_{\mathrm{i}} \tag{2}$$

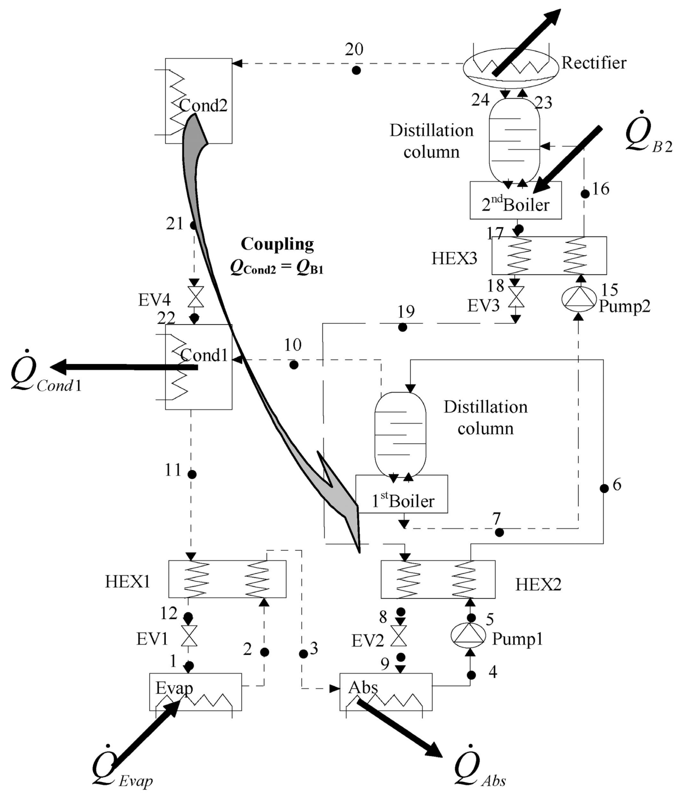

Fig. 2 Serie flow double generator absorption chiller

Entropy balance:

$$\dot{S}_{\mathrm{gen,K}} = \sum_{\mathrm{e}} \dot{m}_{\mathrm{e}} s_{\mathrm{e}} - \sum_{\mathrm{i}} \dot{m}_{\mathrm{i}} s_{\mathrm{i}} - \frac{\dot{Q}_{\mathrm{K}}}{T_{\mathrm{K}}} \tag{3}$$

Irreversibility:

$$\dot{I} = T_0 \dot{S}_{\mathrm{gen,K}} \tag{4}$$

where: $\dot{Q}_{\mathrm{K}}$ – heat added or removed from component, K, at temperature T_{K}, T_{K} – the entropic average temperature at which $\dot{Q}_{\mathrm{K}}$ heat is exchanged [4, 11], $\dot{S}_{\mathrm{gen,K}}$ the entropy generation in each component, K, of the chiller.

Cycle analysis

The coefficient of performance of the chiller is defined as

$$COP = \frac{Q_{\mathrm{Evap}}}{Q_{\mathrm{B2}}} \tag{5}$$

Second law analysis begins by applying the first and second law to the entire system, with only heat crossing the chiller boundaries, as shown in Figs 1, 2.

$$\dot{Q}_{\mathrm{Evap}} + \dot{Q}_{\mathrm{B2}} + \dot{Q}_{\mathrm{Cond1}} + \dot{Q}_{\mathrm{Abs}} = 0 \tag{6}$$

$$\frac{\dot{Q}_{\mathrm{Evap}}}{T_{\mathrm{Evap}}} + \frac{\dot{Q}_{\mathrm{B2}}}{T_{\mathrm{B2}}} + \frac{\dot{Q}_{\mathrm{Cond1}}}{T_{\mathrm{Cond\,1}}} + \frac{\dot{Q}_{\mathrm{Abs}}}{T_{\mathrm{Abs}}} = -\sum_{\mathrm{K}} \dot{S}_{\mathrm{gen,K}} \tag{7}$$

T_{Evap}, T_{B2}, T_{Cond1} and T_{Abs} the corresponding entropic average temperatures. The first condenser and the absorber exchange heat with the same heat sink (air). Their entropic average temperatures are set equal to the environment temperature at (T_{Cond}=T_{Abs}=T_0). The second boiler exchanges heat at varying temperature. Calculation of the exact entropic average temperature for this desorbing process is complex, and was therefore set to the highest temperature of the weak solution. T_{Evap} was estimated as the numerical average of the entering and leaving temperatures. The error introduced by these approximations is minimal as shown in [4, 11].

When Eq. (7) is multiplied by T_0 and subtracted from Eq. (6), the following equation is obtained:

$$\frac{\dot{Q}_{\mathrm{Evap}}}{\dot{Q}_{\mathrm{B2}}} = \left(\frac{T_{\mathrm{Evap}}}{T_0 - T_{\mathrm{Evap}}}\right)\left(\frac{T_{\mathrm{B2}} - T_0}{T_{\mathrm{B2}}}\right) - \left(\frac{T_{\mathrm{Evap}} T_0}{T_0 - T_{\mathrm{Evap}}}\right)\left(\frac{\sum_{\mathrm{K}} \dot{S}_{\mathrm{gen,K}}}{\dot{Q}_{\mathrm{B2}}}\right) \tag{8}$$

Where $T_{\mathrm{Evap}}/(T_0 - T_{\mathrm{Evap}})\cdot[(T_{\mathrm{B2}} - T_0)/T_{\mathrm{B2}}]$ is the reversible COP_{rev} of a thermally-driven refrigerator operating between the three temperature reservoirs at T_{Evap}, T_0 and T_{B2}. Equation (8) may thus be rewritten as:

$$COP = COP_{\mathrm{rev}} - \sum_{\mathrm{K}} COP_{\mathrm{lost,K}} \tag{9}$$

where

$$COP_{\text{lost,K}} = \left(\frac{T_{\text{Evap}}T_0}{T_0 - T_{\text{Evap}}}\right)\left(\frac{\dot{S}_{\text{gen,K}}}{\dot{Q}_{\text{B2}}}\right) \tag{10}$$

Equation (10) shows how much the entropy generation of each component, *K*, of the chiller degrades the reversible COP_{rev} to the actual *COP*.

Second law efficiency, η, is defined as the ratio of the actual *COP* to the maximum (reversible) *COP* under the same operating conditions:

$$\eta = \frac{COP}{COP_{\text{rev}}} \tag{11}$$

Results

The operating conditions for the cycles are:

- evaporator pressure, maximizing the *COP*, P_E=3.5 bar,
- first condenser and absorber temperature: 312 K,
- evaporator exit temperature: 275 K,
- maximum second generator temperature, T_{17}=443 K,
- cooling capacity 17.5 kW at 300–285 K,
- pinchs in heat exchangers: 5 K,
- sub-cooling of absorber and condenser: 3 K.

Figure 3 shows the *COP* evolution of the two configurations *vs.* at the same operating conditions listed above. The parallel flow configuration is more performant. This difference may be explained by the fact that when the two generators are assembled in series the second generator is supplied by the weak solution of the first stage (7), whereas in the parallel configuration the two generators are supplied by the strong solution. In addition, in the first configuration (Fig. 1), an internal rectifier, where the energy of cooling is recovered by the strong solution, is used whereas in the second configuration the weak solution (7) can not be used to cool the rectifier. Computer simulation results are used in the thermodynamic analysis.

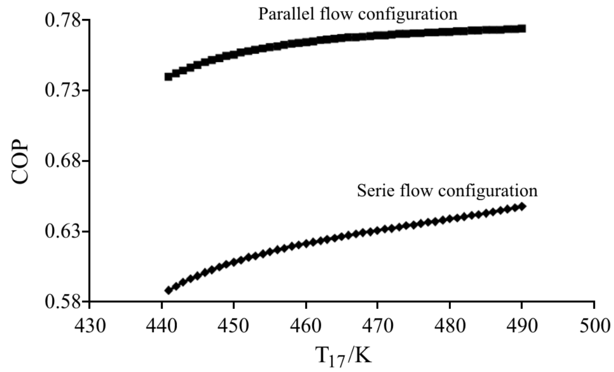

Fig. 3 *COP* of the two configurations *vs.* chiller driving temperature T_{17}

Table 1 provides the analysis results of the parallel flow configuration. Further examining of the irreversibility in each component reveals that the absorber, the solution heat exchangers and the first condenser contribute most to the *COP* decrease (70%).

For the ideal process, the reversible *COP* is 2.292. Degradations in all the components results in an actual *COP* of 0.761. The absorber contribution to the degradation in the machine is the largest, 0.337.

It is shown that the parallel flow configuration is more performant that the serie flow configuration. The performance difference can be explained by the fact that first configuration utilize the given driving heat source better than the second configuration. Figure 4 presents a comparison of irreversibility in various components of the two configurations. It is clear

Table 1 Simulation and thermodynamic analysis results for the parallel flow configuration

Component	Simulation results			Thermodynamic analysis results				
	Q/kW	COP_{rev}	COP	$\dot{S}_{\text{gen,K}}$/W K^{-1}	$\dot{I}$/W	$COP_{\text{degradation,K}}$	$\dot{I}/\dot{I}_{\text{total}}$/%	η
Evaporator	17.5	2.292	0.761	0.461	141.982	0.040	3.076	0.332
HEX1	2.544	2.292	0.761	0.436	134.257	0.038	2.909	0.332
EV1	0.00	2.292	0.761	0.181	55.707	0.016	1.207	0.332
1st Condenser (Cond1)	–18.31	2.292	0.761	1.915	589.823	0.167	12.78	0.332
Absorber	–25.925	2.292	0.761	3.854	1186.95	0.337	25.72	0.332
EV2	0.00	2.292	0.761	0.165	50.971	0.014	1.104	0.332
HEX2	31.311	2.292	0.761	2.591	797.98	0.226	17.288	0.332
Mixer	0.00	2.292	0.761	0.168	51.863	0.015	1.124	0.332
HEX3	30.374	2.292	0.761	2.216	682.587	0.194	14.79	0.332
EV3	0.00	2.292	0.761	0.454	139.929	0.039	3.031	0.332
2nd Boiler (B2)	22.9965	2.292	0.761	1.361	419.27	0.119	9.084	0.332
Rectifier	3.535	2.292	0.761	0.975	300.183	0.085	6.504	0.332
Coupling (Cond2_1st Boiler)	3.514	2.292	0.761	0.208	64.21	0.018	1.391	0.332

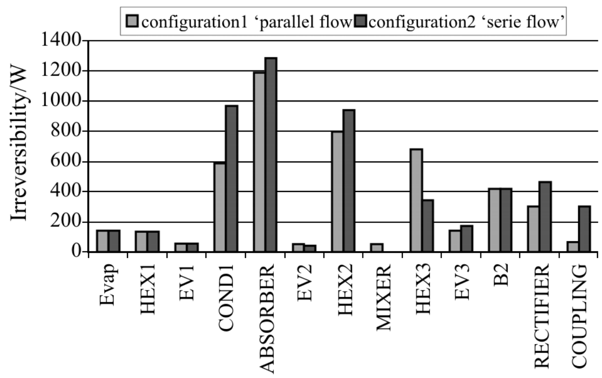

Fig. 4 Comparison of irreversibility in various components of the two studied configurations

that irreversibility (or exergy destruction) in serie flow configuration is more important than that of parallel flow. Expect HEX3 irreversibility in all the other components of the serie flow configuration is significantly important.

Conclusions

In this study, the parallel and serie flow configurations of ammonia–water double generator absorption chiller are simulated and compared. The exergy analysis is performed to quantify the irreversibility in each component of studied configurations. The comparison shows that the parallel flow configuration utilize the given heat source better than the serie flow. At this end the first configuration is more performant and present less irreversibility than the second. Thermodynamic analysis results indicate that in the absorber, the solution heat exchangers, and the condenser may lay the greatest potential for chiller efficiency improvement. Also they indicate that focusing on irreversibility analysis is a more direct way of analysing the potential for improving the efficiency of ammonia–water absorption chiller.

Nomenclature

COP	coefficient of performance	(–)
h	enthalpy	(J mol^{-1})
$\dot{I}$	irreversibility	(kW)
$\dot{m}$	molar flow rate	(mol s^{-1})
P	pressure	(bar)
$\dot{Q}$	heat transfer	(kW)
$\dot{S}_{gen}$	entropy generation	(kW K^{-1})
T	Temperature	(°C, K)

Greek

η	second law efficiently	(–)
α	distribution coefficient	(°C, K)

Subscripts

Abs	absorber
B	boiler
Cond	condenser
Evap	evaporator
EV	Expansion valve
Rect	rectifier
HEX	heat exchanger
f	refrigerant
p	weak solution, poor
r	rich, strong solution

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