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SECOND LAW STUDY OF AMMONIA-WATER DOUBLE EFFECT ABSORPTION CHILLER

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ABSTRACT

A second law analysis of an ammonia-water double-effect double-generator absorption chiller is performed. The irreversibility in each component in the chiller is quantified and the contribution of each component to the overall system efficiency determined. Computer simulation was carried out in order to determine the stream properties and heat and work exchanged with the surroundings [1]. Simulation results where then used to analyse the entropy generation and irreversibility (or exergy destruction) of each component. The results indicate that in the absorber, the solution heat exchangers, and the condenser the greatest irreversibilities occur. In these equipements may lay the most improvement potential of the cycle efficiency.

1. INTRODUCTION

The common way to specify the energy efficiency of an 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 [2, 4, 5].

2. CYCLE DESCRIPTION

Figure 1 shows schematically a double effect-double generator configuration. Such a machine is composed essentially of two condensers, an evaporator, two steam generators each one provided with a boiler and distillation column, an absorber, four expansion valves, two pumps and an adjustable three-way valve. The steam (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 **a** of the strong solution (4) 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 deparation 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.
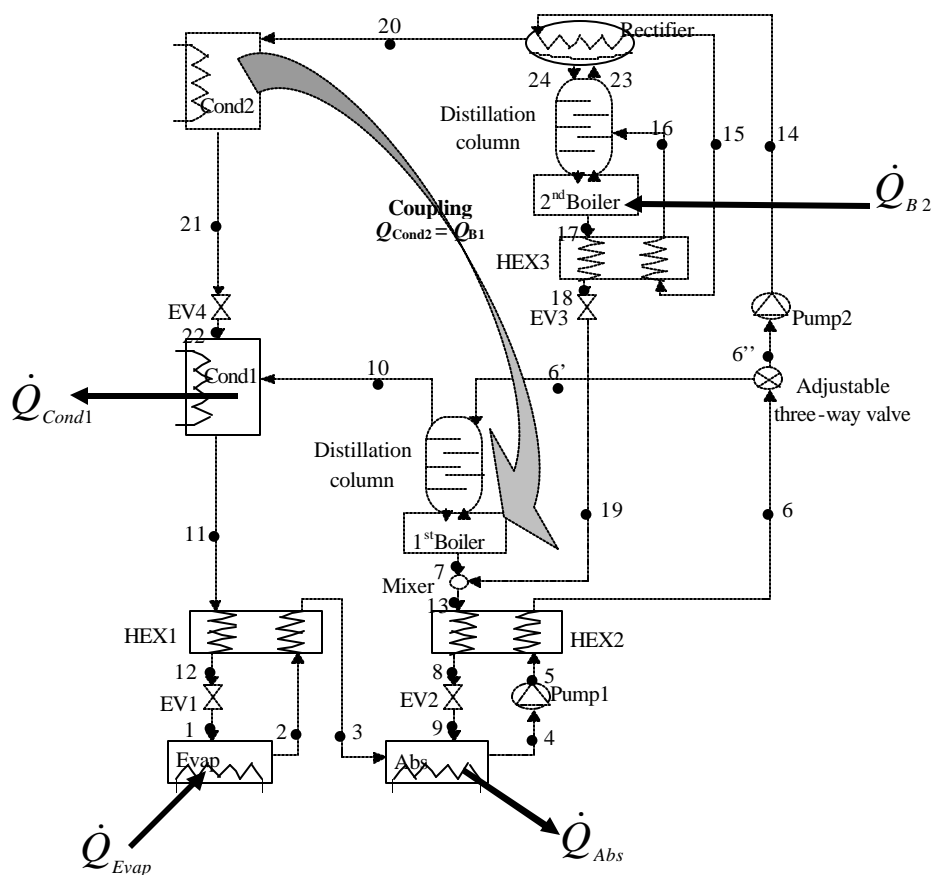

Figure 1 : double effect double generator absorption chiller

3. METHODOLOGY OF SIMULATION

The formulation of the simulation model for the 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 77 code [11], is used. The fluid thermodynamic properties are calculated in a subroutine incorporated in the program.

4. THERMODYNAMIC ANALYSIS

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

Overall mass balance :
$$\sum_e \dot{m}_e = \sum_i \dot{m}_i \qquad (1)$$

Energy balance :
$$\dot{Q}_K = \sum_e \dot{m}_e h_e - \sum_i \dot{m}_i h_i \qquad (2)$$

Entropy balance :
$$\dot{S}_{gen,K} = \sum_e \dot{m}_e s_e - \sum_i \dot{m}_i s_i - \frac{\dot{Q}_K}{T_K} \qquad (3)$$

Irreversibility :
$$\dot{I} = T_0 \dot{S}_{gen,K} \qquad (4)$$

Where:

$\dot{Q}_K$: Heat added or removed from component, K, at temperature T_K ,

T_K : The entropic average temperature at which heat $\dot{Q}_K$ is exchanged [4, 6],

$\dot{S}_{gen,K}$: The entropy generation in each component, K, of the chiller,

4.2. Cycle Analysis
The coefficient of performance of the chiller is defined as

$$COP = \frac{Q_{Evap}}{Q_{B2}} \qquad (5)$$

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 Figure1.

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

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

$T_{Evap}, T_{B2}, T_{Cond1}$, and T_{Abs} the corresponding the entropic average temperatures. The first condenser and the absorber exchange heat with the same heat sink (air). Their entropic average temperature are set equal to the environment temperature at ($T_{Cond1} = T_{Abs} = T_0$). 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 by Alefeld [4, 6].

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

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

$\left(\frac{T_{Evap}}{T_0 - T_{Evap}} \right)\left(\frac{T_{B2} - T_0}{T_{B2}} \right)$: 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 :

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

where

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

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, h, is defined as the ratio of the actual COP to the maximum (reversible) COP under the same operating conditions :

$$h = \frac{COP}{COP_{rev}} \tag{11}$$

5. RESULTS

The operating conditions for the cycle are:
- Evaporator pressure, maximizing the COP [1], $P_E = 3.5$ 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.

Computer simulation results presented in [1] are used in this thermodynamic analysis. Table 1 provides the analysis results.

Table 1: Results of the first and the second law analysis for the chiller

Component	First law analysis			Second law analysis				
	Q (kW)	COP_{rev}	COP	$\dot{S}_{gen,K}$ (W/K)	$\dot{i}$ (W)	$COP_{deg\,radation,K}$	$\dot{i}/\dot{i}_{total}$ %	h
Evaporator	17.5			0.461	141.982	0.040	3,076	
HEX1	2.544			0.436	134.257	0.038	2,909	
EV1	0.00			0.181	55.707	0.016	1,207	
1st Condenser(Cond1)	-18.31			1.915	589.823	0.167	12,78	
Absorber	-25.925			3.854	1186.95	0.337	25,72	
EV2	0.00			0.165	50.971	0.014	1,104	
HEX2	30.311	2.292	0.761	2.591	797.98	0.226	17,288	0.332
Mixer	0.00			0.168	51.863	0.015	1,124	
HEX3	30.374			2.216	682.587	0.194	14,79	
EV3	0.00			0.454	139.929	0.039	3,031	
2nd Boiler (B2)	22.9965			1.361	419.27	0.119	9,084	
Rectifier	3.535			0.975	300.183	0.085	6,504	
Coupling (Cond2_1st Boiler)	3.514			0.208	64.21	0.018	1,391	

Further examining of the irreversibility in each component (figure 2) 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.

6. CONCLUSION

This study applies the second law analysis to quantify the irreversibility of each component in ammonia-water double effect double generator absorption chiller. Simulation results indicate that in the absorber, the solution heat exchangers, and the condenser may lay the greatest potential for chiller efficiency improvement.
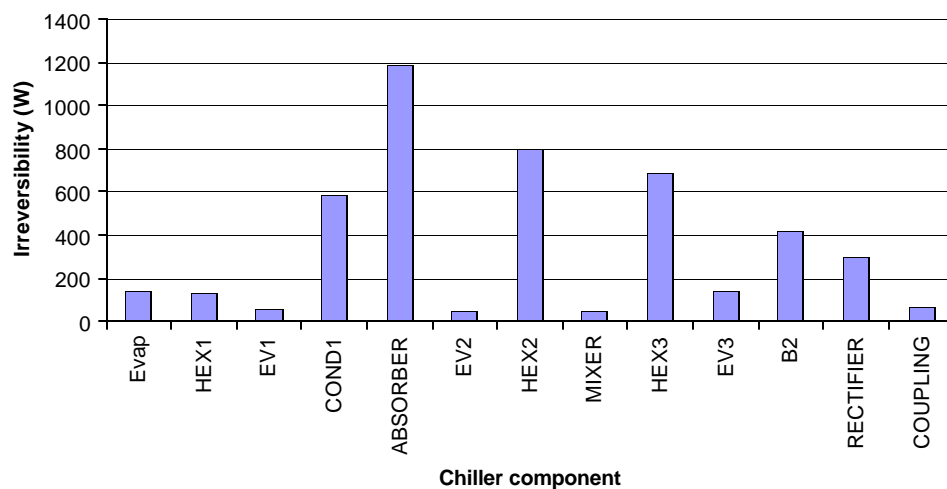
Figure 2 : Irreversibility in chiller components.

NOMENCLATURE

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

Greek

h	second law efficiency	
	distribution coefficient	(–)
a		(°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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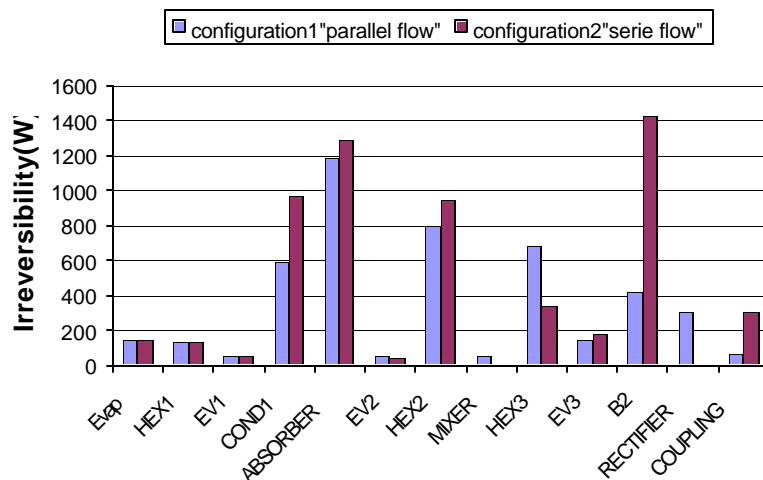
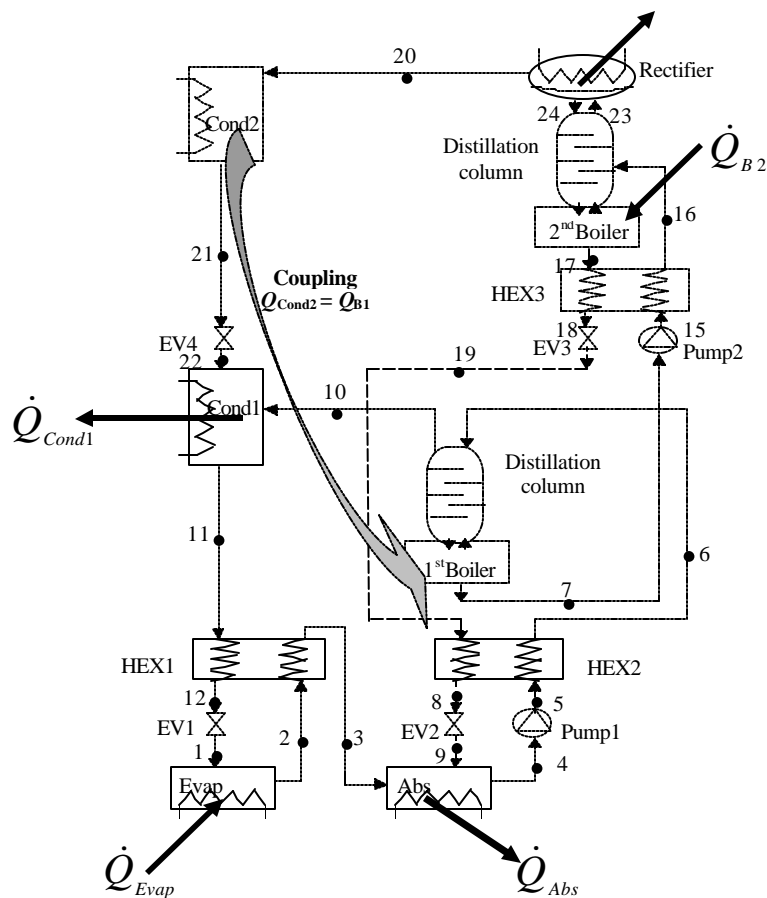Figure 2 : Comparison of irreversibility of the chiller components of two configurations.

Figure 7 : Serie flow double effect double generator absorption chiller