CHAPTER VII

Recommendations for System Improvement

According to the experimental results described in chapter 6, performance of DAR was relatively poor compared with other refrigeration systems. In the past, many researchers tried to improve its performance as discussed in chapter 2. Heat recovery within the system was an option proposed in the literature. In the DAR, heat input is supplied only at the generator while the other components reject or lose heat to the surroundings.

To improve the system performance, heat transferred in all components should be calculated so that the analysis could be more precise. Any modification carried out to improve the system performance can be done pertinently. Based on the mathematical model proposed in chapter 5, some calculation samples are presented in this chapter. The calculation model can be used as an effective tool for determination of the system heat balance. An experimental result was used as a calculation example. The operating conditions are tabulated in Table 7.1.

Table 7.1 An operating condition used for heat balance calculation

operating pressure	13.3bar
cooling water temperature	30-32°C.
rectification temperature (averaged)	75-80°C
auxiliary gas charge (initially)	6.1bar
solution concentration	0.23
mass of chilled water	30kg.
chill water temperature	30 down to 0°C.
heat input at the generator	1.3kW.

7.1 Increase evaporator and absorber mass transfer performance

According to the discussion provided in chapter 6, it was found that after the system was operated continuously, cooling capacity of the system was limited. Increment of system heat input did not raise much of the cooling effect when the system limit was reached. It was proposed that this limitation occurred as a result of limitation in the absorption and evaporation capabilities. It could be occurred by too little absorption or evaporation from wetted areas. Or it could occur from too short a period due to fast flow of the fluid. To improve the system performance, these two factors must be considered simultaneously.

Table 7.2 Comparison of modified systems

	energy transfer (W)					energy transfer (%)				
	A	B	C	D	E	A	B	C	D	E
generator heat input	1322*	1322	2992	962	966	100.0*	100.0	100.0	100.0	100.0
evaporator heat input	135*	174	174	174	174	10.2*	13.2	5.8	18.0	18.0
total energy input	1457*	1496	3166	1136	1140	110.2*	113.2	105.8	118.0	118.0
absorber heat reject	899	936	2606	576	936	68.0	70.8	87.1	59.9	96.9
condenser heat reject	202	202	202	202	202	15.3	15.3	6.7	21.0	20.9
rectifier heat reject	356	356	356	356	0	26.9	26.9	11.9	37.0	0
total heat reject	1457	1494	3164	1134	1138	110.2	113.0	105.7	117.9	117.8
heat transfer at solution heat exchanger	1505	1670	0	2030	1670	113.8	126.3	0	211.0	172.9

* obtained from the experiment
A: based on the experiment (estimated evap.-abs. $\varepsilon = 0.8$, and estimated SHX $\varepsilon = 0.7$)
B: with evap.-abs. $\varepsilon = 1.0$
C: without the SHX (evap.-abs. $\varepsilon = 1.0$, and SHX $\varepsilon = 0$)
D: increased SHX effectiveness (evap.-abs. $\varepsilon = 1.0$, SHX $\varepsilon = 0.9$)
E: heat recovery from the rectifier (evap.-abs. $\varepsilon = 1.0$, and SHX $\varepsilon = 0.7$)
The evap.-abs. ε is defined by equation 5-18.
The SHX ε is defined as $\dfrac{t_9 - t_{10}}{t_9 - t_{13}}$.

Table 7.2 shows calculated results based on experimental conditions listed in table 7.1. The sum of heat input at the generator and at the evaporator is equal to the sum of heat rejected at the absorber, the condenser, and the rectifier. Percentage value of the evaporator input, which is a ratio of the heat input at the evaporator to that at the generator, is recognized as the COP of the system. Based on the experimental refrigerator, column A, the heat input to the evaporator or cooling capacity is 135W or 10.2% of the generator input. When compared with the calculated value, the absorber-evaporator effectiveness is estimated to be around 0.8.

The low effectiveness value results from imperfect evaporation or absorption processes. If a better evaporator and absorber are used, a higher cooling capacity will be achieved. If the ammonia is completely evaporated and absorbed, the evaporator and absorber effectiveness is unity as listed in column B. It should be noted that the COP and cooling capacity can be increased by up to 29% by improving the evaporator and absorber performances as shown in column B.

7.2 Increase the solution heat exchanger effectiveness

According to the performance of the experimental refrigerator, column A, it can be noted that recovered heat at the solution heat exchanger is surprisingly high. The heat exchanger recovered 1505 W of heat, which is 1.14 times of the generator input. This high heat recovery value can be confirmed by the time required for the start up period. After transferring heat to the generator, it took almost an hour before the cooling effect could be produced. The heat input to the generator was accumulated in the working fluid and recovered internally at the solution heat exchanger.

If the solution heat exchanger is excluded from the experimental refrigerator, the system performance will drop drastically. Heat input must be increased to compensate the disappeared amount of heat from the solution heat exchanger. Heat balance of the system excluding the heat exchanger is shown in column C. It is found that heat input increased up to 3kW and the COP dropped drastically to 0.06. At the same time, heat rejected at the absorber is increased. The strong solution flowing from the separator is hot. Its temperature is a little bit lower than temperature of the pumped liquid. Therefore, the absorption capability might drop due to this high temperature of the strong solution entering the absorber. Therefore, the obtained cooling capacity might be lower than the proposed result, depending on the absorption capability of the strong solution.

Based on the experimental results, the solution heat exchanger effectiveness is estimated to be around 0.7. If a larger heat exchanger with effectiveness of 0.9 is used, the heat recovered will be increased to 2030W. This results in a higher solution temperature at the generator and a cooler solution temperature at the absorber inlet. The heat input at the generator and heat rejected at the absorber will reduce to 962W and 576W respectively as shown in column D. It results in the COP increasing up to 0.18, which is 76% improvement over the experimental refrigerator or 210% improvement over the system without a solution heat exchanger.

7.3 Recover heat from the rectifier

According to table 7.3, at the rectifier (location 5, 6, and 7), the operating temperature is reduced as a result of rectification process. It is reduced from the generator temperature, 141.3°C, down to the rectification temperature, 73.2°C. Heat is rejected from the rectified refrigerant vapor so as to partially condense the solution. From the calculated result of heat balance shown in table 7.2, the rejected heat from the rectifier is 356W. This amount of heat is released to the cooling water supply at the rectifier. Since the temperature of this amount of heat is high, the waste heat can be recovered for transferring to solution as input heat.

It was found that this amount of rejected heat could be used to improve the system performance. It can be transferred to the generator as its temperature is high enough. If all the rectification heat is transferred to the generator rather than reject to the ambient, the generator input will reduce by the same amount as the heat rejected at the rectifier. The calculated results, column E, show that percentage of heat saving at the generator is 27% of input heat of case B. This results in an improvement of system COP up to 38%.

Table 7.3 Properties of working solution in column B

$\varepsilon = 1.0$

location	T (°C)	X	h (kJ/kg)	m (g/sec)	phase
1	112.0	0.225	328.7	5.538	liquid
2	141.3	0.225	567.4	5.538	mixture
3	141.3	0.192	476.3	5.216	sat liquid
4	141.3	0.750	1851.7	0.322	sat vapor
5	73.2	0.517	84.3	0.162	sat liquid
6	73.2	0.750	717.6	0.322	mixture
7	73.2	0.985	1418.1	0.160	sat vapor
8	36.0	0.985	156.0	0.160	liquid
9	139.6	0.202	462.1	5.377	liquid
10	70.8	0.202	151.5	5.377	liquid
11	45.8	0.225	27.1	5.535	liquid
12 liq	0.0	0.000	-0.04	0.002	liquid
12 vap	0.0	1.000	1264.9	0.158	vapor
13	45.8	0.225	27.1	5.538	liquid

7.4 Redesign of the bubble pump

According to the discussion in chapter 6, the DAR operation was not consistent. It had an optimum operating range, which differed from one operating condition to another. The bubble pump operation dominated the operating condition. The DAR cannot be operated unless the bubble pump is started. After the system is started, the bubble pump still dominates the system operating characteristics. The bubble pump operation is dependent upon some parameters such as head ratio, pump tube size, and length, etc.

To improve the system performance, the bubble pump should be designed in a manner so that the vapor generated corresponds to the pumped solution. This should be realized at the beginning of the design stage. The absorption capability is limited by flowing characteristics of the strong solution. Therefore, the bubble pump should be redesigned in a manner that the generated vapor would correspond to the amount of the pumped liquid solution.

Cooling capacity is not only dependent on the amount of refrigerant flow but it also relies on the liquid solution circulation rate. The system will provide maximum COP when the refrigerant is completely evaporated in the evaporator. This can be achieved only when there is enough solution circulating in the absorber. If there is too much refrigerant for the solution to absorb, the unevaporated refrigerant will return to the absorber in liquid phase without producing any refrigeration effect. This can be considered as waste, since heat is always required to produce the refrigerant vapor. If there is too much solution circulating in the absorber, the refrigerant will be completely evaporated in the evaporator. However, this can be considered as waste, since the cold solution from the absorber is heated at the pumping boiler. Then it is returned to and cooled down in the absorber without absorbing the refrigerant vapor. As the amount of refrigerant and solution flow rate is depended on the bubble pump design, it can be implied that the bubble pump is a critical component in a design of DAR.

7.5 Proper charge of the auxiliary gas

It was shown in chapter 6 that the auxiliary gas charge pressure caused different characteristics in the DAR operation. Too low or too high auxiliary gas charge pressure caused the DAR operating performance to become lower. Too low auxiliary gas charge increased partial pressure of ammonia in the evaporator, resulting in a higher temperature of evaporation. Then, temperature difference of the evaporation and chilled water temperature was reduced. Resulting in lower cooling capacity, the system performance was degraded. However, too high charge pressure of the auxiliary gas, overcharged, also degraded the operating performance of the system. As the system is more pressurized, specific volume of the vaporized solution based on the same mass basis is decreased. The minimum heat input required to start the system is increased due to increased heat input required to start the bubble pump.

The operating temperature at the generator is elevated with increased operating pressure, which resulted from the auxiliary gas charge pressure. With overcharged auxiliary gas, operating temperature of the systems is raised. Higher operating temperature causes greater heat losses from the system due to greater temperature difference between the system and the ambient, which degraded the system performance.

To optimize the system performance, the auxiliary gas must be properly charged otherwise, the system will be operated inefficiently. However, the auxiliary gas charge pressure cannot be specified exactly. It is dependent upon the system application and surrounding conditions.

7.6 Conclusion

The DAR was analyzed theoretically based on the experimental results. Four alternatives are proposed for implementation with the system to enhance the performance. Heat recovery from the rectifier is shown to reduce heat input at the generator so that the performance can be increased. To implement with the system, the generator, the bubble pump, the separator, and the rectifier must be rearranged so that the rectification heat could be transferred to the generator. Redesign of the bubble pump and the absorber is another option that could be useful. The new-design bubble pump ought to be operated with the proper ratio of pumped liquid solution and vaporized solution. Then, all of the evaporated solution can be absorbed back to the strong solution in the absorber resulting in the better performance. The third choice is to increase the heat exchanger effectiveness of the SHX. It can be easily done by enlarge the SHX. Then, the absorption capability of the strong solution at the absorber inlet can be increased too. At the same time, the generator heat input could be reduced. Finally, it is recommended that the auxiliary gas charge pressure should be properly considered. As the system operation is directly affected by this parameter. Proper gas charge could help the system to reduce heat loss due to temperature difference of the system and surroundings. It also increased heat transfer capability in the evaporator by maintaining the optimum partial pressure of ammonia in both evaporator and condenser. However, the proper amount of auxiliary gas charge pressure cannot be exactly specified. It is dependent upon the operating conditions and the system applications.