EXPERIMENTAL INVESTIGATIONS AND EXERGY ANALYSIS OF A MODIFIED HYBRID HEAT PUMP SYSTEM USING NH_3-H_2O AS WORKING FLUID

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Abstract: The energy consumption for space air conditioning bears huge share of total energy consumption around the world. Due to the increasing pressures of energy demand, the degradation of environment, global warming and depletion of ozone layer etc., there is need of efficient energy utilization and waste heat recovery. The researchers are concentrating on the alternate and environment friendly refrigerants, especially after the Kyoto and the Montreal Protocols. In this paper, the investigations on a modified hybrid vapour compression/absorption heat pump system using ammonia-water as the working fluid have been presented. This system can also utilize the non-conventional energy sources such as solar energy and bio-gas in addition to the waste heat from different industries. The theoretical as well as the experimental performance based on the energy and exergy analysis have been investigated.

Key words: ***hybrid heat pump system, ammonia-water mixture, exergy analysis***

1 INTRODUCTION

Thermal comfort plays a very important role on the health, working efficiency and feeling of human beings. Human beings can work efficiently in the comfortable weather/climate especially temperature and humidity. In the excessively hot climates it is necessary to reduce the temperature and humidity whereas in the cold climate there is a need to increase the temperature. When the temperature drops below thermal comfort level, especially in the winter season, the heat pump systems are employed to produce heating. In some countries, where the atmospheric temperature is very low, natural heating like solar energy is not sufficient. In such a case refrigeration and fuel fired systems are proven to be suitable heating devices. Continuous efforts have been made by numerous researchers on different type of heat pumps in order to improve their performance and to make them cost effective. Some of the heat pumps developed so far, still have not gained much importance. This may be due to various factors, such as low COP, high investment and operational costs and/or their limited heat producing capacity. Among the different types of heat pumps, the vapour compression machines have been most popular all over the world. This is due to their better performance and low costs while occupying small space. Improvements in their performance have been done by providing intermediate cooling, sub cooling and super heating in both single and multi stage vapour compression systems.

Due to limited resources of energy and increasing demand, there is a concern in the scientific community to rethink and to develop the energy efficient system which is not only economical but also environment friendly. The energy consumption in buildings, commercial installations and space air conditioning constitutes a huge share of total energy consumption not only in the developed world but also in the developing countries. Facing ever increasing pressure of energy demand, environmental degradation, global warming and depletion of ozone layer due to industrialization, the efficient use of energy is a hot topic of research in the scientific community especially after the Kyoto and the Montreal Protocols. The industrial waste heat, the depletion of ozone layer, the release of green gases etc. is creating big problem in the present environmental degradation and global warming. Some governments have decided to restrict by means of penalty on the industries for the release of high temperature waste water and residual waste heat in the environment. In Korea, big penalty is imposed for the release of waste water its temperature is more than 40°C. The heat pump system can extract heat from waste water and utilize it for other industrial applications such as in dye industry, heating and cleaning of buildings. This not only utilizes the waste heat for higher temperature application but also saves the environment degradation.

The advantage of large temperature difference between the solution mixture and the pure saturated refrigerant which is till now being used generally in absorption cycles (Baik et al. 2004; Shah and Sekulic 2003; Zhou and Radermacher 1997; Groll and Radermacher 1994; Ahlby et al. 1991; Arora 1990; Malewski 1988; Gupta and Prasad 1983), can also be realised in the compression cycle, the evaporator and condenser are replaced by a desorber and an absorber. Separation of refrigeration from the solution mixture takes place in the desorber, while mixing of them takes place in the absorber. The desorber operates at low pressure and takes heat from the ambient, while in the present investigations; the heat is absorbed from the industrial waste hot water. The absorber releases the heat of absorption to be used for space heating and/or for hot water applications for different purposes. The compression and expansion devices being retained and a solution pump has been employed as an extra component which pumps the remaining refrigerant-absorbent mixture from desorber to absorber, while the pure refrigerant (NH_3) from the generator is being used after compression (in two stages). In this way, high temperature can be achieved in the absorber during absorption process, while the saturation pressure and temperature of the refrigerant leaving the compressor and entering the absorber remains as that of the condenser of a vapour compression cycle.

In order to study the importance, utility and performance of the vapour compression/ absorption hybrid heat pump system, the exergy analysis has been made, as reported by earlier workers (Xu et al. 2007; Li et al. 2007; Dikici and Akbulut 2007; Morosuk and Tsatsaronis 2007; Sözen and Yücesu 2007; Tarique and Siddiqui 1999; Siddiqui 1997; Kaygusuz and Ayhan 1993, Tyagi et al. 2007; Kotas, 1985; Tyagi et al. 2004).

2 SYSTEM DESCRIPTION

The hybrid vapour compression/absorption heat pump cycle combines two well known heat pump concepts, the compression heat pump and the absorption heat pump as shown in the line diagram of Figure 1. It uses a mixture of refrigerants as the working fluid, one as the absorbent and the other as the desorbent. A key advantage of the hybrid heat pump is the extended range of temperature available for a mixture compared to pure refrigerants. This is the effect of the reduced vapour pressure obtained for a refrigerant in a mixture with less volatile component. Another advantage is the gliding temperature obtained in the absorber and desorber. Gliding temperature here refers to the temperature change obtained as the volatile component is absorbed into and evaporated out of the solution. It reduces irreversibility during heat exchange process between working fluids and results in improved system performance.

In a hybrid cycle, the condenser and evaporator in a conventional heat pump are replaced by an absorber and a desorber. In the desorber the more volatile component of the mixture evaporates over a gliding temperature. Depending on the volatility of the heavy component of the mixture, water, some of this component evaporates as well. As shown in Figure 1, the vapour is compressed (7 to 8) while the remaining solution is pumped to the high pressure level (1-2) where absorption takes place (3 to 4). During the absorption process, the heat is rejected to the heat sink which, in turn, is taken away by the hot water being used for useful applications. The hybrid heat pump cycle and its components were designed by simulation method in which each component was modelled thermodynamically. The absorber and desorber were by given UA values and a mass, energy and specific balance equations. The absorber and desorber outlets were assumed to be saturated states. Adiabatic absorption was assumed in the first part of the absorber i.e. until equilibrium is reached.

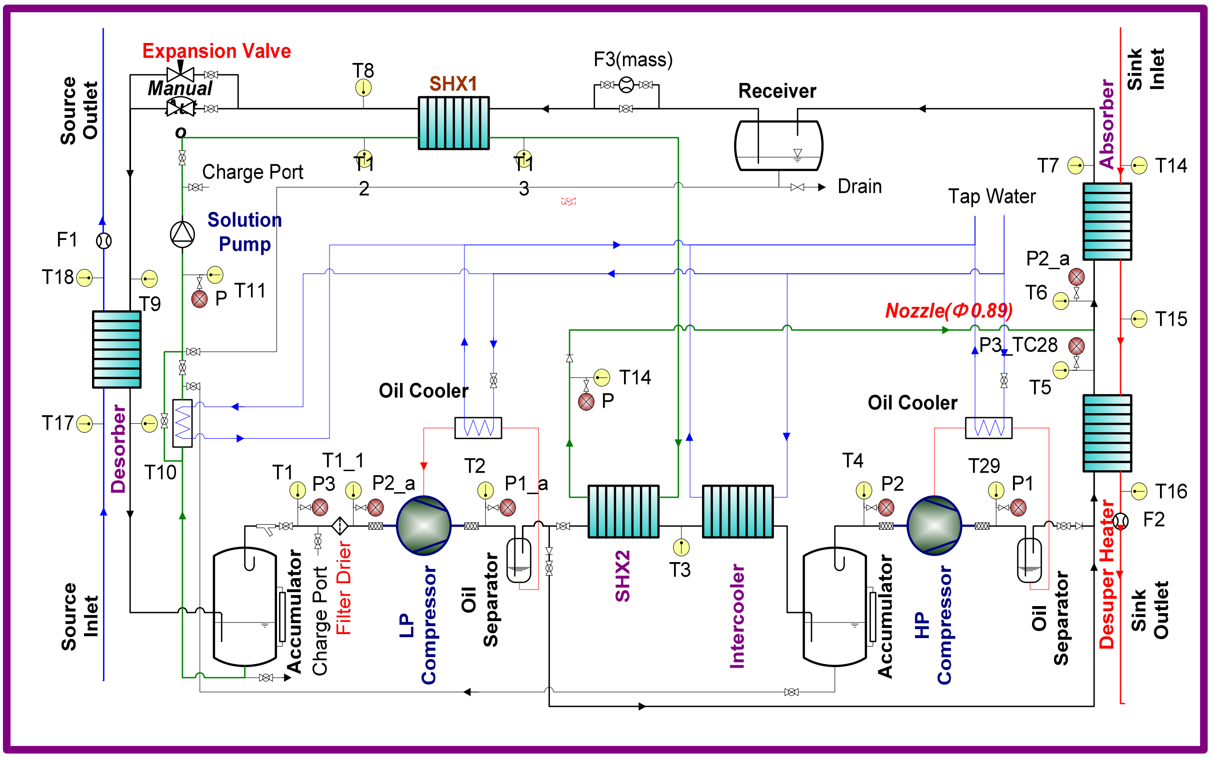

Figure 1: Experimental setup of a compression/absorption hybrid heat pump system

3 EXERGY ANALYSIS

In order to investigate the hybrid heat pump cycle performance, the detail analysis has been carried out by means of exergy analysis. For steady state flow the exergy balance for a thermal system is given as below:

$$Ex_W = \sum_{j=1}^{n}(1-T_a/T_j)Q_j + \sum_{k=1}^{r}\left[(mex)_i-(mex)_o\right]_k + m\frac{C^2}{2} + mgZ_0 - T_a\Delta S_{gen} \qquad (1)$$

where Ex_W represents the useful work done on/by system, the first term on the right hand side represents the exergy summation supplied through heat transfer, while changes in the exergy summation of the working fluid is represented by the second term where i and o refers the inlet and outlet states, the kinetic and potential exergy are given by the term $mC^2/2$ and mgZ_0, respectively, while the exergy loss or the irreversibility in the system is given by the last term on the right hand side, $T_a\Delta S_{gen}$. The other notations namely, Q is the heat transfer rate, m is the mass flow rate of the working fluid, ex is the exergy flow rate per unit mass, C is the bulk velocity of the working fluid, Z_0, is the altitude of the stream above the sea level, g is the specific gravitational force, S_{gen} is the entropy generation rate, T_a is the ambient temperature, T_i is the temperature of the heat source/sink at which the heat is transferred.

The component wise exergy balance of the absorption-compression hybrid system is given as below:

3.1 Low Pressure Compressor (LP)

The exergy balance for the low pressure compressor is give by:

$$-W_{LP} = m_c(ex_1 - ex_2) - T_a \Delta S_{gen,LP} \tag{2}$$

which gives,

$$T_a \Delta S_{gen,LP} = m_c(ex_1 - ex_2) + W_{LP} = m_c T_a (s_2 - s_1) \tag{3}$$

and the entropy generation rate is:

$$\Delta S_{gen,LP} = m_c(s_2 - s_1) \tag{4}$$

The irreversibility = exergy loss is:

$$I_{LP} = T_a \Delta S_{gen,LP} = m_c T_a (s_2 - s_1) \tag{5}$$

The second law efficiency is:

$$\eta_{II,LP} = 1 - \frac{I_{LP}}{W_{LP}} = \frac{m_c(ex_2 - ex_1)}{W_{LP}} \tag{6}$$

3.2 Solution-Heat-Exchanger (SHX-2)

The exergy flow equation for the second solution heat exchanger becomes:

$$0 = m_c(ex_2 - ex_3) - m_p(ex_{28} - ex_{13}) - T_a \Delta S_{gen,SHX-2} \tag{7}$$

which gives:

$$T_a \Delta S_{gen,SHX-2} = \left[\left\{m_c(h_2 - h_3) - m_p(h_{28} - h_{13})\right\} - T_a\left\{m_c(s_2 - s_3) - m_p(s_{28} - s_{13})\right\}\right] \tag{8}$$

The entropy generation rate is:

$$\Delta S_{gen,SHX-2} = \left\{m_c(h_2 - h_3) - m_p(h_{28} - h_{13})\right\}/T_a - \left\{m_c(s_2 - s_3) - m_p(s_{28} - s_{13})\right\} \tag{9}$$

The irreversibility = exergy loss is:

$$I_{SHX-2} = T_a \Delta S_{gen,SHX-2} \tag{10}$$

The second law efficiency is:

$$\eta_{II,SHX-2} = 1 - \frac{I_{SHX-2}}{m_c(ex_2 - ex_3)} = \frac{m_p(ex_{28} - ex_{13})}{m_c(ex_2 - ex_3)} \tag{11}$$

3.3 Intercooler

The exergy flow equation for intercooler becomes:

$$0 = m_c(ex_3 - ex_4) - Ex_{Q_{\mathrm{int}}}(1 - T_a/T_4) - T_a \Delta S_{gen,Int} \tag{12}$$

which gives:

$$T_a \Delta S_{gen,Int} = m_c T_a \left\{(h_3 - h_4)/T_4 - (s_3 - s_4)\right\} \tag{13}$$

The entropy generation rate is:

$$\Delta S_{gen,Int} = m_c \{(h_3 - h_4)/T_4 - (s_3 - s_4)\} \tag{14}$$

The irreversibility = exergy loss is:

$$I_{Int} = T_a \Delta S_{gen,Int} \tag{15}$$

The second law efficiency is:

$$\eta_{II,Int} = 1 - \frac{I_{Int}}{m_c(ex_3 - ex_4)} = \frac{Ex_{Q_{Int}}}{m_c(ex_3 - ex_4)} \tag{16}$$

3.4 High Pressure Compressor (HP)

The exergy balance for the high pressure compressor is give by:

$$-W_{HP} = m_c(ex_4 - ex_{29}) - T_a \Delta S_{gen,HP} \tag{17}$$

which gives,

$$T_a \Delta S_{gen,HP} = m_c(ex_4 - ex_{29}) + W_{HP} = m_c T_a (s_{29} - s_4) \tag{18}$$

and the entropy generation rate is:

$$\Delta S_{gen,HP} = m_c(s_{29} - s_4) \tag{19}$$

The irreversibility = exergy loss is:

$$I_{HP} = T_a \Delta S_{gen,HP} = m_c T_a (s_{29} - s_4) \tag{20}$$

The second law efficiency is:

$$\eta_{II,HP} = 1 - \frac{I_{HP}}{W_{HP}} = \frac{m_c(ex_{29} - ex_4)}{W_{HP}} \tag{21}$$

3.5 Desuperheater (DS)

The exergy flow equation for the desuperheater becomes:

$$0 = m_c(ex_{29} - ex_5) - m_{W1}(ex_{15} - ex_{16}) - T_a \Delta S_{gen,DS} \tag{22}$$

which gives:

$$T_a \Delta S_{gen,DS} = \left[\{m_c(h_{29} - h_5) - m_{W1}(h_{16} - h_{15})\} - T_a \{m_c(s_{29} - s_5) - m_{W1}(s_{16} - s_{15})\}\right] \tag{23}$$

The entropy generation rate is:

$$\Delta S_{gen,DS} = \{m_c(h_{29} - h_5) - m_{W1}(h_{16} - h_{15})\}/T_a - \{m_c(s_{29} - s_5) - m_{W1}(s_{16} - s_{15})\} \tag{24}$$

The irreversibility = exergy loss is:

$$I_{DS} = T_a \Delta S_{gen,DS} \tag{25}$$

The second law efficiency is:

$$\eta_{II,DS} = 1 - \frac{I_{DS}}{m_c(ex_{29} - ex_5)} = \frac{m_{W1}(ex_{16} - ex_{15})}{m_c(ex_{29} - ex_5)} \tag{26}$$

3.6 Absorber (Abs)

The exergy flow equation for the absorber becomes:

$$0 = m_0(ex_6 - ex_7) - m_{W1}(ex_{14} - ex_{15}) - T_a \Delta S_{gen,Abs} \tag{27}$$

which gives:

$$T_a \Delta S_{gen,Abs} = \left[\{m_0(h_6 - h_7) - m_{W1}(h_{15} - h_{14})\} - T_a \{m_0(s_6 - s_7) - m_{W1}(s_{15} - s_{14})\}\right] \tag{28}$$

The entropy generation rate is:

$$\Delta S_{gen,Abs} = \{m_0(h_6 - h_7) - m_{W1}(h_{15} - h_{14})\}/T_a - \{m_0(s_6 - s_7) - m_{W1}(s_{15} - s_{14})\} \tag{29}$$

The irreversibility = exergy loss is:

$$I_{Abs} = T_a \Delta S_{gen,Abs} \tag{30}$$

The second law efficiency is:

$$\eta_{II,\,Abs} = 1 - \frac{I_{Abs}}{m_c(ex_6 - ex_7)} = \frac{m_{W1}(ex_{15} - ex_{14})}{m_0(ex_6 - ex_7)} \tag{31}$$

3.7 Solution-Heat-Exchanger (SHX-1)

The exergy flow equation for the first solution heat exchanger becomes:

$$0 = m_0(ex_7 - ex_8) - m_p(ex_{12} - ex_{13}) - T_a \Delta S_{gen,SHX-1} \tag{32}$$

which gives:

$$T_a \Delta S_{gen,SHX-1} = \left[\{m_0(h_7 - h_8) - m_p(h_{13} - h_{12})\} - T_a \{m_0(s_7 - s_8) - m_p(s_{13} - s_{12})\}\right] \tag{33}$$

The entropy generation rate is:

$$\Delta S_{gen,SHX-1} = \left[\{m_0(h_7 - h_8) - m_p(h_{13} - h_{12})\}/T_a - \{m_0(s_7 - s_8) - m_p(s_{13} - s_{12})\}\right] \tag{34}$$

The irreversibility = exergy loss is:

$$I_{SHX-1} = T_a \Delta S_{gen,SHX-1} \tag{35}$$

The second law efficiency is:

$$\eta_{II,\,SHX-1} = 1 - \frac{I_{SHX-1}}{m_0(ex_7 - ex_8)} = \frac{m_p(ex_{13} - ex_{12})}{m_0(ex_7 - ex_8)} \tag{36}$$

3.8 Expansion Valve (Exp)

The exergy flow equation for the expansion valve becomes:

$$0 = m_0(ex_8 - ex_9) - T_a \Delta S_{gen,Exp} \tag{37}$$

which gives:

$$T_a \Delta S_{gen,Exp} = m_0\left[(h_8 - h_9) - T_a(s_8 - s_9)\right] \tag{38}$$

The entropy generation rate is:

$$\Delta S_{gen,Exp} = m_0\left[(h_8 - h_9)/T_a - (s_8 - s_9)\right] \tag{39}$$

The irreversibility = exergy loss:

$$I_{Exp} = T_a \Delta S_{gen,Exp} \tag{40}$$

The second law efficiency is:

$$\eta_{II,Exp} = 1 - \frac{I_{Exp}}{m_0(ex_8 - ex_9)} \tag{41}$$

3.9 Desorber (Des)

The exergy flow equation for the desorber becomes:

$$0 = m_0(ex_9 - ex_{10}) - m_{W2}(ex_{117} - ex_{18}) - T_a \Delta S_{gen,Des} \tag{42}$$

which gives:

$$T_a \Delta S_{gen,Des} = \left[\{m_0(h_{10} - h_9) - m_{W2}(h_{17} - h_{18})\} - T_a\{m_0(s_{10} - s_9) - m_{W2}(s_{17} - s_{18})\}\right] \tag{43}$$

The entropy generation rate is:

$$\Delta S_{gen,Des} = \{m_0(h_{10} - h_9) - m_{W2}(h_{17} - h_{18})\}/T_a - \{m_0(s_{10} - s_9) - m_{W2}(s_{17} - s_{18})\} \tag{44}$$

The irreversibility = exergy loss is:

$$I_{Des} = T_a \Delta S_{gen,Des} \tag{45}$$

The second law efficiency is:

$$\eta_{II,Des} = 1 - \frac{I_{Des}}{m_{W2}(ex_{17} - ex_{18})} = \frac{m_0(ex_{10} - ex_9)}{m_{W2}(ex_{17} - ex_{18})} \tag{46}$$

3.10 Solution Pump (SP)

The exergy balance for the pump is given by:

$$-W_P = m_p(ex_{11} - ex_{12}) - T_a \Delta S_{gen,P} \tag{47}$$

which gives,

$$T_a \Delta S_{gen,P} = m_p(ex_{12} - ex_{11}) + W_P = m_p T_a(s_{12} - s_{11}) \tag{48}$$

and the entropy generation rate is:

$$\Delta S_{gen,P} = m_p(s_{12} - s_{11}) \tag{49}$$

The irreversibility = exergy loss is:

$$I_P = T_a \Delta S_{gen,P} = m_p T_a(s_{12} - s_{11}) \tag{50}$$

The second law efficiency is:

$$\eta_{II,HP} = 1 - \frac{I_P}{W_P} = \frac{m_p(ex_{12} - ex_{11})}{W_P} \tag{51}$$

4 RESULTS AND DISCUSSION

In order to have a comparative study of the hybrid absorption/compression system, the real time data was measured and the calculations made using EES (Engineering Equation Solver) software. The temperature, pressure and mass flow rate were measured using sensor based thermocouple, pressure meter and flow meter, respectively, at different state points. The basic properties such as entropy, enthalpy, concentration and quality of the refrigerant (NH_3-H_2O mixture) at different state points were calculated using EES software assuming the steady state operation and are shown in Table 1. Similar properties of the cooling/heating fluids were also calculated using REFPROP database software and the entropy generation rates, the exergy loss (irreversibility) and other performance parameters were calculated using a simple Excel sheet and the results are shown in Table 2.

Table 1: Measured parameters and calculated properties at different state points.

State Points	Temper-ature (K)	Pressure (bar)	Enthalpy (kJ/kg)	Entropy (kJ/kg-K)	Concent-ration X	Quality
0	299.15	2.65	-122.42	0.2384	0.4659	0
1	302.18	2.65	1356.02	4.9051	0.9947	1
2	349.19	7.97	1440.04	5.1673	0.9961	1.001
3	308.22	7.97	1321.10	4.5434	0.9961	0.9916
4	303.34	7.97	1303.12	4.4832	0.9961	0.9883
5	316.26	11.06	1323.06	4.4054	0.9961	0.9908
6	346.37	11.04	193.11	1.0051	0.1712	-0.001
7	326.20	11.04	108.72	0.7541	0.1712	-0.001
8	307.21	11.04	28.76	0.5016	0.1712	-0.001
9	291.23	2.65	-39.65	0.2759	0.1712	-0.001
10	317.31	2.65	70.13	0.6374	0.1712	-0.001
11	291.20	2.65	-158.40	0.1168	0.4685	-0.001
12	293.43	11.67	-148.62	0.1466	0.4685	-0.001
13	310.32	11.67	-72.82	0.3978	0.4685	-0.001
14	313.78	1.010	170.25	0.5808	--	0
15	313.82	1.010	170.42	0.5813	--	0
16	315.72	1.01	178.36	0.6065	--	0
17	316.62	1.01	182.12	0.6184	--	0
18	313.99	1.01	171.13	0.5836	--	0
28	321.18	11.67	-23.69	0.5534	0.4685	-0.001
29	340.11	11.06	1403.01	4.6511	0.9961	1.001

Table 2: Calculated irreversibility and mass flow rate for different subsystems

S. No.	Subsystem	Mass flow rate (kg/h)	Irreversibility (kW)
1	Low pressure compressor	7.524	0.252929
2	Solution heat exchanger-2	7.524	0.103517
3	Intercooler	7.524	0.084950
4	High pressure compressor	7.524	0.161218
5	Desuperheater	7.524	0.029443
6	Absorber	70.01	0.038737
7	Solution heat exchanger-1	70.01	0.087208
8	Expansion valve	70.01	0.009178
9	Desorber	70.01	0.097119
10	Solution pump	58.02	0.139271
11	State points 14-16	2.022	--
12	State points 17-20	2.883	--

For a typical set of operating parameters, the irreversibility rate (exergy loss) in the low pressure compressor is found to be the highest while it is found to be the lowers in the case of expansion valve. The effects of ambient air temperature on the irreversibility rates of different subsystems/components are shown on Figures 2 and 3 for a typical set of operating conditions given in Table 1. It can be seen from Figure 2 that the irreversibility rate in of the low and high pressure compressors as well as of the solution pump increase as the ambient air temperature increases. This is because of the fact that the availability decreases with the reference temperature which is ambient temperature in the present case. On the other hand, the irreversibility rates of the solution heat exchangers, intercooler, desuperheater, and the absorber is found to be decreasing function of the ambient temperature while its found to be reverse in the case of intercooler as can be seen from Figure 3. The effect of ambient temperature is found to be more pronounce for the desorber and less pronounce for the solution heat pump, while for other components it is found somewhere between the two extreme cases mentioned above.

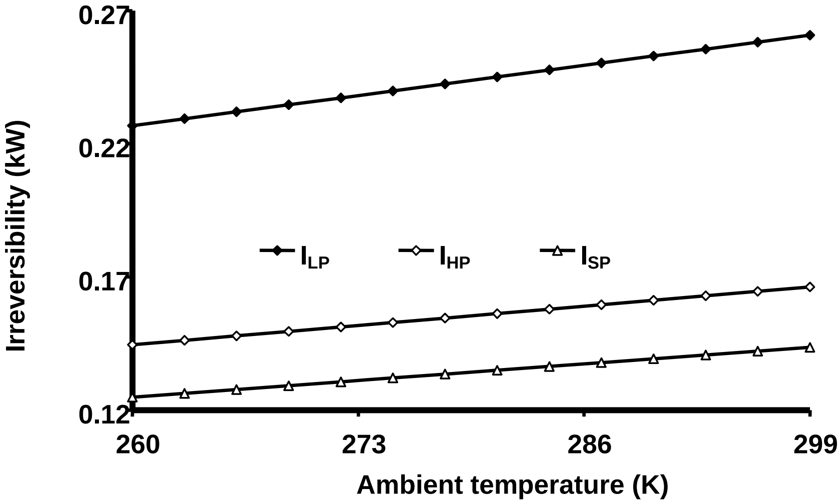

Figure 2: Variations of compressors and pump irreversibility against the ambient temperature

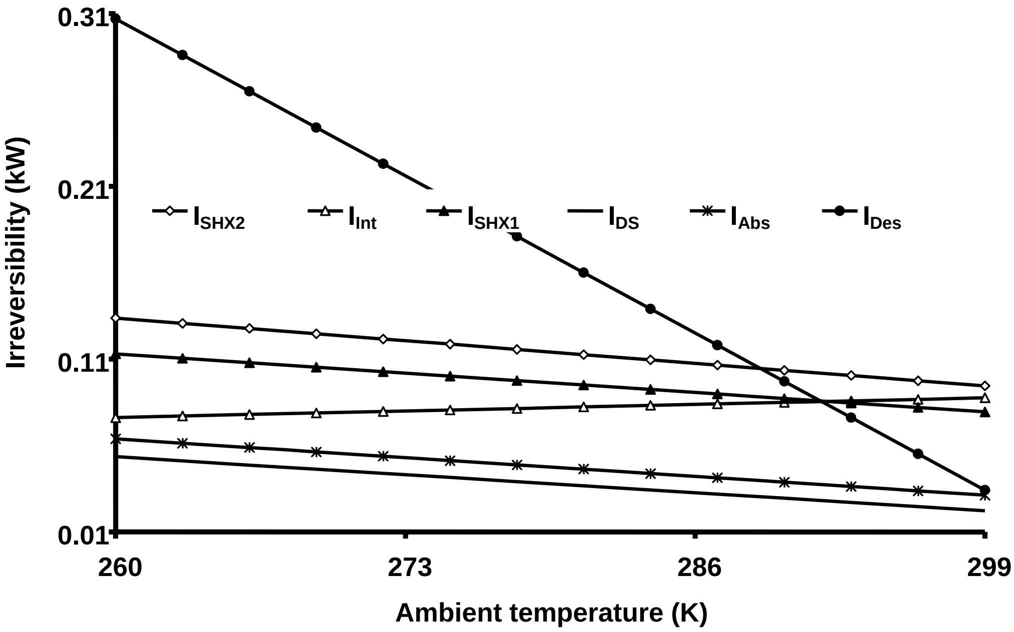

Figure 3: Variations of the component irreversibility against the ambient temperature

The exergy content of any system for given amount of energy/heat at a particular temperature is a function of reference temperature and changes with the reference temperature. The change in the irreversibility rate against the ambient temperature for some components is found to be different than that of the general trends. For example, the irreversibility rates in some components are found to be decreasing function of ambient temperature as can be seen in Figure 3. This gives a different picture for such component which may be due to other factors, such as the compositions and chemical reactions etc. and need to be studied in detail.

The heating and cooling COP of the experimental system have been calculated for particular operating condition. The heating COP was found to be 1.37 and the cooling COP was found to be 1.84. In general the heating COP is more than that of cooling COP but it reverse in the absorption cycle this is because of the higher temperature of the waste heat source. In the experimental setup the temperature of the heat source was maintained by some means. Although the performance of the present hybrid heat pump system is much less than that of a conventional heat pump system, yet it has a satisfactory result for a suitable range and can be further improved up to a certain extent using optimal design and operating parameters. Further modification of the present hybrid heat pump system is underway and the performance of the modified system will be presented in the coming time.

5 CONCLUSIONS

The theoretical and experimental investigations of a modified hybrid vapour compression/ absorption heat pump system using ammonia-water as the working fluid have been presented. The irreversibility rate (exergy loss) in the low pressure compressor is found to be the highest while it is found to be the lowers in the case of expansion valve. It is also found that the irreversibility rate in of the low and high pressure compressors as well as of the solution pump increase as the ambient air temperature increases.. On the other hand, the irreversibility rates of the solution heat exchangers, intercooler, desuperheater, and the absorber is found to be decreasing function of the ambient temperature while its found to be reverse in the case of intercooler. The effect of ambient temperature is found to be more pronounce for the desorber and less pronounce for the solution heat pump. The irreversibility rates in some components are found to be decreasing function of ambient temperature, which may be due to various reasons, such as the composition of the refrigerant mixture and chemical reactions. The performance of the present hybrid heat pump system is found to be much less than that of a conventional heat pump system. To obtain satisfactory results, further modifications with improved design of certain components in the present system are required.

6 ACKNOWLEDGEMENT

This work was jointly supported by Korean Foundation of Science and Technology (KOFST) through Brain Pool Research Scheme and by the grant from the Development Project of Energy and Resources Technique in the Korea Energy Management Corporation (KEMCO).

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