

Exergy Analysis and Parametric Study of a Sea Water Source Heat Pump System

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ABSTRACT: This article presents the second law based analysis of a sea water source cascade heat pump system using two heat pumps operating with two different refrigerants. The cascade heat pump system consists of two different heat pumps attached in series in a way that the condenser for the first stage heat pump is the evaporator for the second stage heat pump. This not only can enhance the exhaust temperature in the second condenser at the final stage but also can improve the performance of the overall system. The study has been made for a wide range of operating parameters. It is found that the performance based on the second law analysis is in the good agreement with the general results. The experimental performance is found to be different from the second law based performance. The system may achieve better performance if the irreversibility associated with different components can be minimized. The work for further investigations and modifications is underway to utilize the non conventional energy resources such as solar energy and biogas besides, the waste heat recovery for different applications and hence, the present system can be further modified for real life applications.

Keywords: sea water source, cascade heat pump, exergy analysis, irreversibility

NOMENCLATURE

CHX = Cascade heat exchanger

Cond = Condense

E = Exergy [kW]

Exp = Expansion valve

Evp = Evaporator

h = Specific enthalpy [kJ/kg]

I = Irreversibility

m = Mass flow rate [kg/s]

P = Pressure [kPa]

Q = Heat transfer rate [kW]

SWHX = Sea water heat exchanger

s = Entropy [kJ/kg-K]

T = Temperature [K]

W = Power input [kW]

Greek symbols

η = Efficiency

Δ = Difference/change

Subscripts

CHX = Cascade heat exchanger

Comp = Compressor

Cond = Condenser

Evap = Evaporator

Exp = Expansion valve

gen = Generation

II = Second law efficiency

HX = Heat exchanger

1 = First/one

2 = Second/two

i = Input

o = Output

1. INTRODUCTION

Thermal comfort plays a very important role on the mental and physical health and working efficiency of human beings. In an excessively cold climate it is necessary to increase the temperature besides the presence of suitable moisture content for thermal comfort. With the advancement of technologies, the electric systems such as room heater, heat pipes, heat pump systems are employed to produce heating. In some countries, where the temperature is very low and natural heating like solar energy is not sufficient, the heat pump 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.

Due to the limited resources of energy and the 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 the increasing pressure of energy demand, environmental degradation, global warming and depletion of ozone layer due to various reasons, most commonly the industrialization. The efficient use of energy is a hot topic of research especially after the Kyoto and the Montreal Protocols.

In some countries during the winter season the ambient air temperature is very low and the requirement of heating is high, hence, the performance of the heat pump decreases. As it is evident that during

summer the sea water temperature is lower and during winter it is much higher than the ambient air temperature. Thus the sea water can be used as heat sink during summer and as heat source during winter for the cascade heat pump system. In such a case, the heat pump system can be used to reject/extract the heat to and from sea water for heating/cooling applications. This not only can utilize the sea water heat for higher temperature applications but also can reduce a huge potential of the environment degradation (Tyagi et al. 2008, Zhou et al. 2007; Song et al. 2007; Okamoto, 2006, Kumar et al. 2005; D'Accadia & Sasso, 1998).

In order to study the importance, utility and performance of the cascade heat pump system, the exergy analysis has been made, as reported by different researchers (Tyagi et al. 2008, Wang et al. 2008; Tyagi et al. 2007, Kaygusuz and Ayhan 1993 and Kotas, 1985).

2. SYSTEM DESCRIPTION

The cascade heat pump cycle combines with two different heat pumps working on different refrigerants such as R410a and R134a respectively. The simplified diagram of the complete cascade heat pump system is shown in Fig.1. As shown in Fig.1, the sea water pumped to the heat exchanger at state point 1, where it released the heat during state points 2-3 and the heat is taken away by the circulating water between state points 4-5. This circulated water is then divided into three main parts and each part passes through a particular heat exchanger viz. low, medium and high temperature heat exchanger, depending on the weather conditions. For example, during winter, as only heating is required, the circulating water passed through the evaporator between state points 6-7. On the other hand, during summer, the circulating water passes through state points 12-13 through different heat exchangers. These heat exchangers can also be characterized as the evaporator heat exchanger, cascade heat exchanger and

condenser heat exchanger respectively. In order to obtain the optimum performance of the entire cascade heat pump system, the mass flow rate in each heat exchanger is controlled by some means depending on the weather conditions.

As it is clear from Fig.1, the evaporator is attached between state points 6-7, with the first heat pump operating between the evaporator (8-9) and the cascade heat exchanger (10-11) using R410A as the working fluid. During winter, the cascade heat exchanger is working as the heat sink for the first heat pump, while it is working as the heat source for the second heat pump. On the other hand, during summer a condenser is used in place of the cascade heat exchanger while the evaporator is attached with the space to be cooled rather than the sea water heat exchanger.

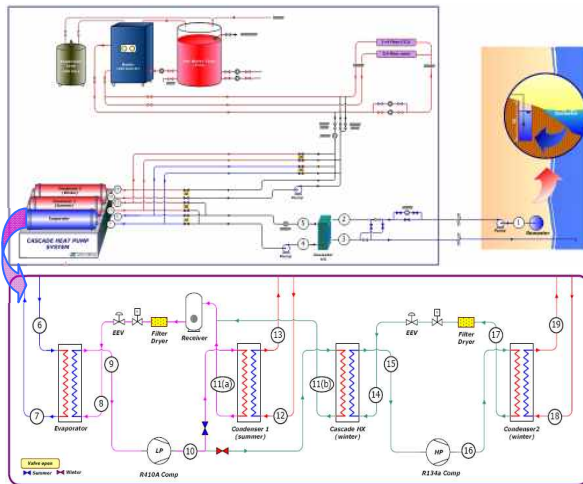


Figure 1: Schematic and line diagram of a sea water source cascade heat pump system.

The second heat pump is operating between cascade (14-15-16-17-14) heat exchanger and the condenser (18-19 and 16-17) heat exchanger using R134a as the working fluid. The second heat pump operates in auto off-on mode. In other words, when the temperature of the exhaust water from the high temperature heat exchanger reached at the set point temperature, it switch off and switch on automatically when it falls below the set point level. It reduces

not only the power consumption but also energy loss and hence, the irreversibility. Also the second heat pump works between the cascade heat exchanger and condenser during winter, while it works similar to the first heat pump in the summer.

3. EXERGY ANALYSIS

We have studied the cascade heat pump system using second law analysis based on the experimental data. For steady state flow the exergy balance for a thermal system is given as below:

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

where Ex_W represents the useful work done on/by the 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 air temperature, T_i is the temperature of the heat source/sink at which the heat is transferred. The component wise exergy balance of the cascade heat pump system is given below:

3.1. Pump1 (SWP)

The exergy flow equation for sea water pump becomes:

$$-W_{swp} = m_{sw}(ex_2 - ex_1) - T_a \Delta S_{gen,swp} \quad (2)$$

which gives:

$$T_a \Delta S_{gen,swp} = m_{sw}(ex_2 - ex_1) + W_{swp} \quad (3)$$

The entropy generation rate is:

$$\Delta S_{gen,swp} = m_{sw}(s_2 - s_1) \quad (4)$$

The irreversibility = exergy loss i.e.

$$I_{swp} = T_a \Delta S_{gen,swp} \quad (5)$$

The second law efficiency is:

$$\eta_{II,swp} = 1 - \frac{I_{swp}}{W_{swp}} = \frac{m_{sw}(ex_2 - ex_1)}{W_{swp}} \quad (6)$$

Similar expressions can be written for the second pump operating between state points 7-4.

3.2. Sea Water Heat-Exchanger (HX1)

The exergy flow equation for sea water heat exchanger becomes:

$$0 = m_{sw}(ex_2 - ex_3) - m_w(ex_4 - ex_5) - T_a \Delta S_{gen,HX1} \quad (7)$$

which gives:

$$T_a \Delta S_{gen,HX1} = \{m_{sw}(h_2 - h_3) - m_w(h_5 - h_4)\} - T_a \{m_{sw}(s_2 - s_3) - m_w(s_5 - s_4)\} \quad (8)$$

The entropy generation rate is:

$$\Delta S_{gen,HX1} = \{m_{sw}(h_2 - h_3) - m_w(h_5 - h_4)\} / T_a - \{m_{sw}(s_2 - s_3) - m_w(s_5 - s_4)\} \quad (9)$$

The irreversibility = exergy loss i.e.

$$I_{HX1} = T_a \Delta S_{gen,HX1} \quad (10)$$

The second law efficiency is:

$$\eta_{II,HX-1} = 1 - \frac{I_{HX1}}{m_{sw}(ex_2 - ex_3)} = \frac{m_w(ex_5 - ex_4)}{m_{sw}(ex_2 - ex_3)} \quad (11)$$

where m_{sw} and m_w are, the mass flow rates of sea water and circulating water respectively at a particular time.

3.3. Evaporator (Evap)

The exergy flow equation for the evaporator becomes:

$$0 = m_{wl}(ex_6 - ex_7) - m_{R1}(ex_8 - ex_9) - T_a \Delta S_{gen,Evap} \quad (12)$$

which gives:

$$T_a \Delta S_{gen,Evap} = \{m_{wl}(h_6 - h_7) - m_{R1}(h_9 - h_8)\} - T_a \{m_{wl}(s_6 - s_7) - m_{R1}(s_9 - s_8)\} \quad (13)$$

The entropy generation rate is:

$$\Delta S_{gen,Evap} = \{m_{wl}(h_6 - h_7) - m_{R1}(h_9 - h_8)\} / T_a - \{m_{wl}(s_6 - s_7) - m_{R1}(s_9 - s_8)\} \quad (14)$$

The irreversibility = exergy loss is:

$$I_{Evap} = T_a \Delta S_{gen,Evap}$$

The second law efficiency is:

$$\eta_{II,Evap} = 1 - \frac{I_{Evap}}{m_{wl}(ex_6 - ex_7)} = \frac{m_{R1}(ex_9 - ex_8)}{m_{wl}(ex_6 - ex_7)} \quad (15)$$

where m_{wl} and m_{R1} are respectively, the mass flow rates of circulating water and refrigerant in the evaporator.

3.4. Low Pressure Compressor (LP)

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

$$-W_{Com1} = m_{R1}(ex_9 - ex_{10}) - T_a \Delta S_{gen,LP} \quad (16)$$

which gives,

$$T_a \Delta S_{gen,LP} = m_{R1}(h_9 - h_{10}) - T_a(s_9 - s_{10}) + W_{LP} \quad (17)$$

and the entropy generation rate is:

$$\Delta S_{gen,LP} = m_{R1}(s_{10} - s_9) \quad (18)$$

The irreversibility = exergy loss is:

$$I_{LP1} = T_a \Delta S_{gen,LP} = m_{R1}(s_{10} - s_9) \quad (19)$$

The second law efficiency is:

$$\eta_{II,LP} = 1 - \frac{I_{LP}}{W_{LP}} = \frac{m_{R1}(ex_{10} - ex_9)}{W_{LP}} \quad (20)$$

3.4. Condenser-1 (Cond1)

The exergy flow equation for the first condenser (summer case) becomes:

$$0 = m_{R1}(ex_{10} - ex_{11a}) - m_{w2}(ex_{12} - ex_{13}) - T_a \Delta S_{gen,Cond1} \quad (21)$$

which gives:

$$T_a \Delta S_{gen,Cond1} = \left[\frac{\{m_{R1}(h_{10} - h_{11a}) - m_{w2}(h_{13} - h_{12})\}}{T_a} - \frac{\{m_{R1}(s_{10} - s_{11a}) - m_{w2}(s_{13} - s_{12})\}}{T_a} \right] \quad (22)$$

The entropy generation rate is:

$$\Delta S_{gen,Cond1} = \left[\frac{\{m_{R1}(h_{10} - h_{11a}) - m_{w2}(h_{13} - h_{12})\}}{T_a} - \frac{\{m_{R1}(s_{10} - s_{11a}) - m_{w2}(s_{13} - s_{12})\}}{T_a} \right] \quad (23)$$

The irreversibility = exergy loss is:

$$I_{Cond1} = T_a \Delta S_{gen,Cond1} \quad (24)$$

The second law efficiency is:

$$\eta_{II,Cond1} = 1 - \frac{I_{Cond1}}{m_{R1}(ex_{10} - ex_{11b})} = \frac{m_{w2}(ex_{13} - ex_{12})}{m_{R1}(ex_{10} - ex_{11a})} \quad (25)$$

3.5. Cascade-Heat-Exchanger (CHX)

There are two modes of operation for the cascade heat pump system, one for winter and the second for summer. As can be seen from Fig.1, during winter, the cascade heat exchanger works as the evaporator for the second heat pump system and the condenser works as the heat sink. On the other hand, during the summer there are two condensers those work as the heat source while the cascade heat exchanger and the evaporator works as the heat sink, as can be seen from Fig.1. The exergy flow equation for the cascade heat exchanger can be written as below:

$$0 = m_{R1}(ex_{10} - ex_{11b}) - m_{R2}(ex_{14} - ex_{15}) - T_a \Delta S_{gen,CHX} \quad (26)$$

which gives:

$$T_a \Delta S_{gen,CHX} = \left[\frac{\{m_{R1}(h_{10} - h_{11b}) - m_{R2}(h_{15} - h_{14})\}}{T_a} - \frac{\{m_{R1}(s_{10} - s_{11b}) - m_{R2}(s_{15} - s_{14})\}}{T_a} \right] \quad (27)$$

The entropy generation rate is:

$$\Delta S_{gen,CHX} = \left[\frac{\{m_{R1}(h_{10} - h_{11b}) - m_{R2}(h_{15} - h_{14})\}}{T_a} - \frac{\{m_{R1}(s_{10} - s_{11b}) - m_{R2}(s_{15} - s_{14})\}}{T_a} \right] \quad (28)$$

The irreversibility = exergy loss is:

$$I_{CHX} = T_a \Delta S_{gen,CHX} \quad (29)$$

The second law efficiency is:

$$\eta_{II,CHX} = 1 - \frac{I_{CHX}}{m_{R1}(ex_{10} - ex_{11b})} = \frac{m_{R2}(ex_{15} - ex_{14})}{m_{R1}(ex_{10} - ex_{11b})} \quad (30)$$

3.6. Expansion Valve (Exp-1)

The exergy flow equation for the first expansion valve can be given by:

$$0 = m_0(ex_{11} - ex_8) - T_a \Delta S_{gen,Exp-1} \quad (31)$$

which gives:

$$T_a \Delta S_{gen,Exp-1} = m_{R1}[(h_{11} - h_8) - T_a(s_{11} - s_8)] \quad (32)$$

The entropy generation rate is:

$$\Delta S_{gen,Exp-1} = m_{R1}[(h_{11} - h_8)/T_a - (s_{11} - s_8)] \quad (33)$$

The irreversibility = exergy loss:

$$I_{Exp-1} = T_a \Delta S_{gen,Exp-1} \quad (34)$$

The second law efficiency is:

$$\eta_{II,Exp-1} = 1 - \frac{I_{Exp-1}}{m_{R1}(ex_{11} - ex_8)} \quad (35)$$

3.7. High Pressure Compressor (HP)

The exergy balance for the compressor of first compressor is given by:

$$-W_{HP} = m_{R2}(ex_{15} - ex_{16}) - T_a \Delta S_{gen,HP} \quad (36)$$

which gives,

$$T_a \Delta S_{gen,HP} = m_{R2}(h_{15} - h_{16}) - T_a(s_{15} - s_{16}) + W_{HP} \quad (37)$$

and the entropy generation rate is:

$$\Delta S_{gen,HP} = m_{R2}(s_{16} - s_{15}) \quad (38)$$

The irreversibility = exergy loss is:

$$I_{HP} = T_a \Delta S_{gen,HP} = m_{R2}(s_{16} - s_{15}) \quad (39)$$

The second law efficiency is:

$$\eta_{II,HP} = 1 - \frac{I_{HP}}{W_{HP}} = \frac{m_{R2}(ex_{15} - ex_{16})}{W_{HP2}} \quad (40)$$

3.8. Condenser2 (Cond2)

Due to page limits, we have discussed only the winter case in this paper. The

exergy flow equation for the (second) condenser becomes:

$$0 = m_{R1}(ex_{16} - ex_{17}) - m_{w3}(ex_{18} - ex_{19}) - T_a \Delta S_{gen,Cond2} \quad (41)$$

which gives:

$$T_a \Delta S_{gen,Cond2} = \left[\frac{m_{R2}(h_{16} - h_{17}) - m_{w3}(h_{19} - h_{18})}{T_a} - \{m_{R2}(s_{16} - s_{17}) - m_{w3}(s_{19} - s_{18})\} \right] \quad (42)$$

The entropy generation rate is:

$$\Delta S_{gen,Cond2} = \left[\frac{m_{R2}(h_{16} - h_{17}) - m_{w3}(h_{19} - h_{18})}{T_a} - \{m_{R2}(s_{16} - s_{17}) - m_{w3}(s_{19} - s_{18})\} \right] \quad (43)$$

The irreversibility = exergy loss is:

$$I_{Cond2} = T_a \Delta S_{gen,Cond2} \quad (44)$$

The second law efficiency is:

$$\eta_{II,Cond2} = 1 - \frac{I_{Cond2}}{m_{R2}(ex_{16} - ex_{17})} = \frac{m_{w3}(ex_{19} - ex_{18})}{m_{R2}(ex_{16} - ex_{17})} \quad (45)$$

For summer conditions, similar expressions can also be written for the first condenser.

3.6. Expansion Valve (Exp2)

The exergy flow equation for the second expansion valve becomes:

$$0 = m_{R2}(ex_{17} - ex_{14}) - T_a \Delta S_{gen,Exp2} \quad (46)$$

which gives:

$$T_a \Delta S_{gen,Exp2} = m_{R2}[(h_{17} - h_{14}) - T_a(s_{17} - s_{14})] \quad (47)$$

The entropy generation rate is:

$$\Delta S_{gen,Exp2} = m_{R2}[(h_{17} - h_{14})/T_a - (s_{17} - s_{14})] \quad (48)$$

The irreversibility = exergy loss:

$$I_{Exp2} = T_a \Delta S_{gen,Exp2} \quad (49)$$

The second law efficiency is:

$$\eta_{II,Exp2} = 1 - \frac{I_{Exp2}}{m_{R2}(ex_{17} - ex_{14})} \quad (50)$$

4. RESULTS AND DISCUSSION

To carry out the comparative study of the cascade heat pump system, the real time data was measured and the calculations were made using REPROF database software. The temperature, pressure and mass flow

rate were measured using sensor based thermocouple, pressure gauge and flow meter, respectively, at different state points. The entropy generation rates, the exergy loss, irreversibility and other parameters were calculated using a simple Excel sheet. The effect of ambient air temperature on the irreversibility of each component is shown in Figures.(2-4). On the other hand, the comparative exergetic loss in components is shown in Fig.5 at the steady state operation. It can be seen from these figures that the irreversibility rate in all the heat exchangers, expansion devices decreases as the ambient air temperature increases while it is reverse in the case of compressors and pumps. This is because of the reason that the availability decreases with the increase in the reference temperature i.e. with the ambient air temperature. As a result the overall loss in all the heat exchangers and expansion devices decreases, decreasing the irreversibility rates in these components. On the other hand, the irreversibility due to over heating of the compression/pumping devices and increment in the temperature of the cooling fluid causes the higher loss in the useful energy in these devices. As a result, the irreversibility rates in the compressors and pumps increase as the ambient air temperature increases.

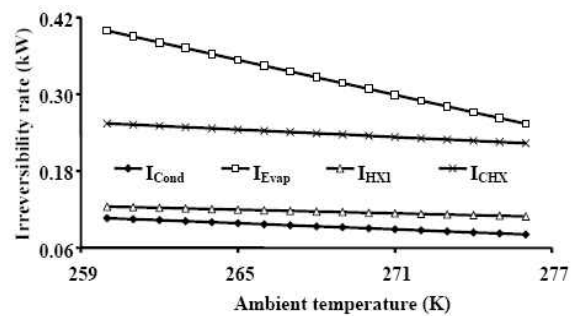


Figure 2: Irreversibility vs ambient temperature for various heat exchangers.

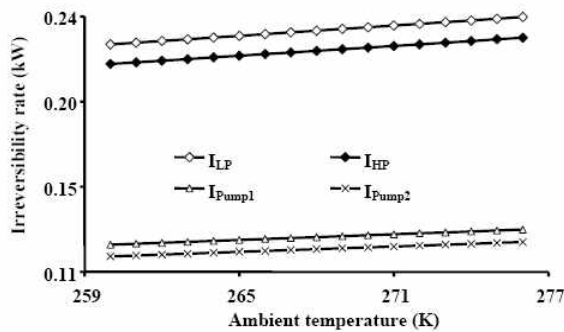


Figure 3: Irreversibility vs ambient temperature for compressors and pumps.

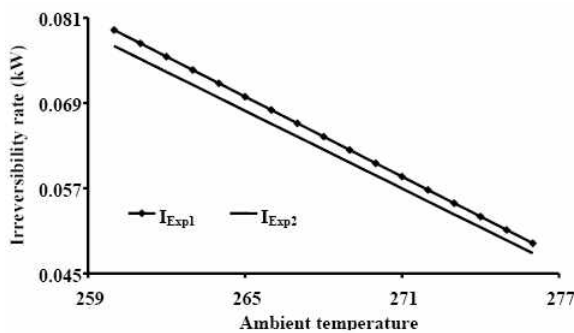


Figure 4: Irreversibility vs ambient temperature for expansion valves

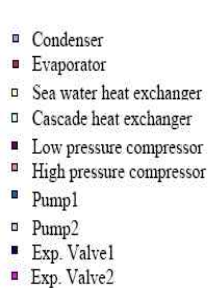


Figure 5: Component wise irreversibility rate in the cascade heat pump cycle

The comparative irreversibility rate in each components of the cascade heat pump cycle system is shown in Fig.5 for a typical set of operating parameters. It is seen from Fig.5 that the irreversibility rate for evaporator is the highest while for the expansion valves, it is the lowest. On the other hand, the irreversibility rates for other components lies somewhere within the two extreme cases.

These results are similar in the trend of these analyzed by number of researchers and are available in the literature. Basically, the

irreversibility rates in the sources side are due to the fact that the quality of energy affects the performance of any system and which is the heat source in the case of a heat engine while it is an evaporator in the case of refrigeration cycles. Also the rate of irreversibility for the low pressure compressor (LP) is found to be higher than that of the high pressure compressor (HP). Similarly, the irreversibility rate for the sea water pump is found to be more than that of the water circulating pump. Again, the overall irreversibility rates are for the first heat pump cycle is found to be higher than that of the second heat pump. This may be due to various reasons such as the properties of the working fluid.

5. CONCLUSIONS

The exergetic investigation of a sea water source cascade heat pump system using different working fluid has been presented. Based on the second law analysis, the following conclusions are made:

- The irreversibility rates in the heat exchanger and expansion devices are found to be decreasing function of the ambient air temperature.
- The effect of ambient air temperature is found to be more pronounced for the irreversibility rates in the evaporator, while it is found to be less pronounced for the expansion devices.
- The irreversibility rate (exergy loss) in the evaporator is found to be the highest while it is found to be the lowest in the case of expansion valves. For other components the irreversibility losses are found to be somewhere between these two extreme cases.
- The rate of irreversibility for the low pressure compressor (LP) is found to be higher than that of the high pressure compressor (HP). Similarly, the rate of irreversibility the sea water pump is found to be more than that of the water circulating pump. Again, the overall

irreversibility rates are for the first heat pump cycle is found to be higher than that of the second heat pump.

- One of the main facts is that the availability depends on the reference temperature which is the ambient air temperature in the present case. Since the availability decreases with the increase of reference temperature, and hence, the energy loss also decreases. As a result, the irreversibility loss in heat exchanger and expansion devices decrease while it is reverse for the compression and pumping devices.

Further investigations and modifications in the cascade heat pump system are under way and the detail results with the useful findings will be presented during the Conference.

ACKNOWLEDGEMENTS

This work was supported by the Korean Foundation of Science and Technology (KOFST) through Brain Pool Research Program. The technical and financial support from through MKE 2005-E-BD02-05-0-000 is also highly appreciated.

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