Exergy Analysis of On/Off Controlled Heat Pump

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Key Words : Exergy, On/Off control, Reference state

Abstract

A multi-type heat pump controls the mass flow rate of the working fluid to cope with variable heat loads when it is under dynamic load condition. This paper describes the exergy analysis associated with the unsteady response of a heat pump. First, a basic heat pump cycle is examined at a steady state to show the general trends of exergy variations in each process of the cycle. Entropy generation issue for the heat exchangers is discussed to optimize the heat pump cycle. Secondly, the performance of the inverter-driven heat pump is compared to that of the conventional one when the heat load is variable. Thirdly, the exergy destruction rate of the heat pump with On/Off operation is calculated by simulating the thermodynamic states of the working fluid in the condenser and the evaporator. The inefficiency of On/Off operation during the transient period is quantitatively described by the exergy analysis.

Nomenclature

A : Heat transfer area
COP : Coefficient of performance
ex : Specific exergy
U : Overall heat transfer coefficient
h : Heat transfer coefficient or specific enthalpy
L : Length
$\dot{m}$: Mass flow rate
P : Pressure
Q_m : Heat exchanged in the condenser
s : Specific entropy
T : Temperature
v : Specific volume
W : Compressor work

Subscripts

air : Air

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cond : Condenser
evap : Evaporator
o : Reference equilibrium state

1. Introduction

There is a growing interest in the air-conditioning system, which uses heat pump for cooling and heating of the complex medium/small size buildings.

A heat pump has an advantage of small entropy generation and high total energy efficiency because it accomplishes both cooling and heating as a sole mechanical equipment. The heat transfer occurs between the refrigerant and the indoor/outdoor air with relatively small temperature difference.(1)

The characteristic of the multi-type heat pump is that the independent air-conditioning for the separate space can be accomplished by distributing the refrigerant to each space from the compressor unit. The medium/small size building can be a typical application of the multi-type heat pump.

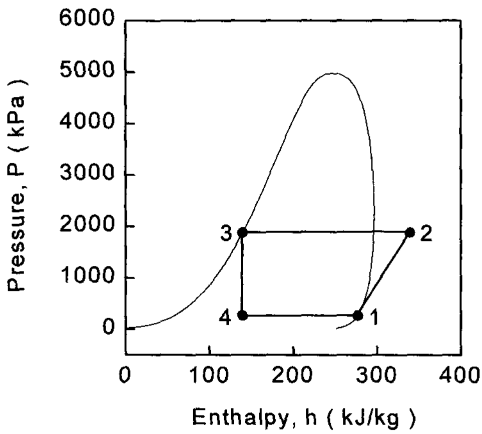

Fig.1 Pressure-enthalpy diagram of the basic heat pump cycle.

The multi-type heat pump driven by an inverter is particularly an energy saving device that modulates the flow rate of the refrigerant by controlling the rotational speed of the compressor.(2,3)

In the present study, the representative base-line cycle of the heat pump is considered at steady state and the exergy loss is calculated for each component of the heat pump. And then the heat pump cycles with an inverter and without an inverter are compared to examine the capability of the modulations for the various heat loads.

The exergy loss in the On/Off operation without an inverter is also calculated for the existing heat pump and the inefficiency is represented quantitatively. Moreover, the entropy generation associated with the heat transfer and the pressure drop in the condenser and the evaporator is examined for the optimization of the heat pump.(4)

2. Exergy Analysis for Steady State

2.1. Base-line cycle

For a given heat load of 5000 W, the optimized base-line cycle is constructed by simulating each state with the real thermodynamic properties of the refrigerant.

<Assumptions for the base-line cycle>

① Refrigerant : R-22
② Inlet of the expansion valve : saturated liquid
③ Inlet of the compressor : saturated gas
④ Pressure ratio in the compressor : 7.0
⑤ Isentropic efficiency of the compressor : $\eta_{COMP} = 0.8$
⑥ Temperature of the inlet air in the condenser : 20 ℃

⑦ Temperature of the outlet air in the condenser : 40 ℃
⑧ Temperature of the inlet air in the evaporator : 0 ℃
⑨ UA in the condenser : 300.0 W/ ℃
⑩ UA in the evaporator : 300.0 W/ ℃
⑪ Heat transfer in the condenser (Heat load) : 5000 W
⑫ Negligible pressure drop due to the flow of the refrigerant

The base-line thermodynamic cycle is constructed by the above conditions after the iterative computation. Fig.1 is the P-h diagram for this base-line cycle.

The refrigerant is compressed from the state 1 to the state 2 in the compressor and condenses to the state 3 after the heat transfer to the air in the condenser. The refrigerant expands to the state 4 through the expansion device and completes the cycle returning to the state 1 after the heat transfer from the air in the evaporator.

2.2. Exergy analysis

The specific exergy value at each state of the base-line cycle is calculated. The exergy is defined as the maximum reversible work which the refrigerant can do until it reaches to the equilibrium with the environment. The definition of the exergy in the flow process is therefore as follows.[5]

$$ex = (h - h_o) - T_o(s - s_o) \quad (1)$$

where

ex : specific exergy
T_o : reference equilibrium temperature
h : specific enthalpy at each state
h_o : specific enthalpy at reference equilibrium state
s : specific entropy at each state
s_o : specific entropy at reference equilibrium state

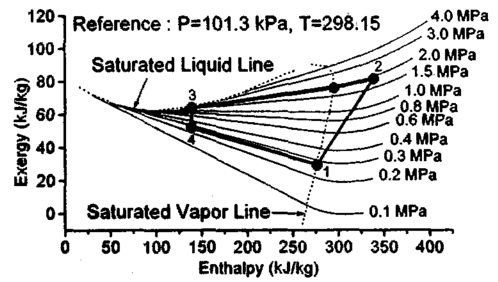

Fig.2 Exergy-enthalpy diagram with the reference temperature of 25 ℃.

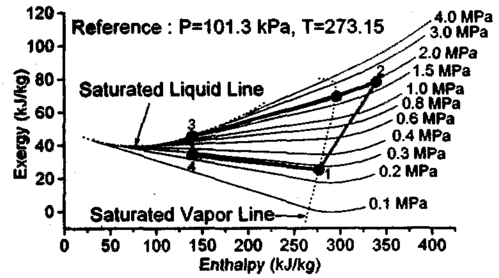

Fig.3 Exergy-enthalpy diagram with the reference temperature of 0 ℃.

The exergy is a thermodynamic property which depends on the reference equilibrium state. In the usual exergy analysis, the reference equilibrium state is chosen as the state of 1 atm and 25 ℃. In this study, however, the change of exergy is also considered for the case that the reference equilibrium state is 0 ℃ and 1 atm (typical outdoor winter condition of the base-line cycle in the present study).

Figures 2 and 3 show that the exergy which is generated by the work in the compressor vanishes gradually in the condenser, the expansion device and the evaporator.[6,7] It is noticeable that the ratio of the exergy loss is differ-

Table 1 Change of exergy in heat pump (Reference state : 25 ℃, 1 atm).

Component	Exergy of refrigerant (kJ/kg)		Change of the exergy for the refrigerant (kJ/kg) (outlet - inlet)	Change of the exergy for the air (kJ/kg) (outlet - inlet)
	inlet	outlet		
Compressor	29.5	81.8	52.3	
Condenser	81.8	64.5	-17.3	0.3
Expansion device	64.5	52.3	-12.2	
Evaporator	52.3	29.5	-22.8	1.4

Table 2 Change of exergy in heat pump (Reference state : 0 ℃, 1 atm).

Component	Exergy of refrigerant (kJ/kg)		Change of the exergy for the refrigerant (kJ/kg) (outlet - inlet)	Change of the exergy for the air (kJ/kg) (outlet - inlet)
	inlet	outlet		
Compressor	25.2	78.3	53.1	
Condenser	78.3	45.7	-32.6	1.98
Expansion device	45.7	34.5	-11.2	
Evaporator	34.5	25.2	-9.3	0.28

ent for the same thermodynamic cycle according to the choice of the reference equilibrium state. The exergy loss in the condenser is apparently underestimated if the reference state is 25 ℃.

Table 1 and 2 show the detailed numerical values. In the exergy analysis of the heat pump, it is appropriate that the outdoor condition should be selected as the reference equilibrium state. Therefore, 0 ℃ is more adequate than 25 ℃ as the reference equilibrium state because the heat pump is usually operated during the winter.

As shown in Table 1 and 2, the exergy loss in the expansion device is a little more than 20 % of the total exergy loss. It is due to the irreversibility accompanied with the isenthalpic process. The exergy loss in the condenser is almost 60 % of the total exergy loss of the system according to Table 2. The exergy losses in the condenser and the evaporator are greatly dependent on the size of the heat exchanger. The exergy value of the refrigerant returns to the original one after one cycle, but the exergy of the air which exchanges heat with the refrigerant increases in the condenser and the evaporator after one cycle. Because the decrease of the exergy for the refrigerant is larger than the increase of exergy for the air in the condenser and the evaporator, the exergy is lost in these heat exchanger processes. Since this exergy loss represents the entropy generation in the heat exchangers, it is very important to design the heat exchangers of the heat pumps to minimize the entropy generation. The phase transition of the refrigerant occurs in the condenser and the evaporator. Therefore, the pressure drop is very significant compared to the other heat exchangers where the heat exchange occurs only in single phase. The pressure drop should be included in the calculation of the entropy generation. The entropy generation due to the pressure drop is comparable to that due to the heat transfer with finite temperature difference. Fig.4 and 5 show the calculation results for the entropy generation of the condenser and the evaporator of the base-line heat pump cycle. The heat exchanger is assumed to be made with a single copper tube. The thermodynamic analysis does not consider any economical aspect that is associated with the material of the heat exchanger.

$S_{gen,\Delta T}$ is the entropy generation rate due to the irreversibility in the process of heat transfer, and $S_{gen,\Delta P}$ is the entropy generation rate due to the pressure drop.

As the heat exchanger size increases with the entropy generation due to the pressure drop. The entropy generation due to the heat exchanger process shows the opposite tendency. As shown in Fig.4 and 5, there should exist an optimal length, L_{opt} that minimizes the total entropy generation. The heat exchangers with an optimal length in the heat pump shall make the whole system more efficient. Since the specific volume of the refrigerant in the evaporator is much larger than that in the condenser, the optimal length of the evaporator is shorter than that of the condenser. The pressure drop in the evaporator is larger than that in the condenser for the same mass flow rate of the refrigerant. Therefore, the entropy generation due to the pressure drop is larger in the evaporator than in the condenser.

2.3. Variation of the heat load and the heat pump cycle

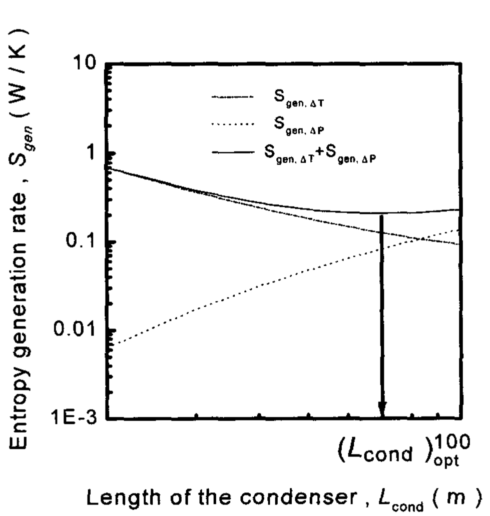

Fig.4 Variation of entropy generation for different length of the condenser.

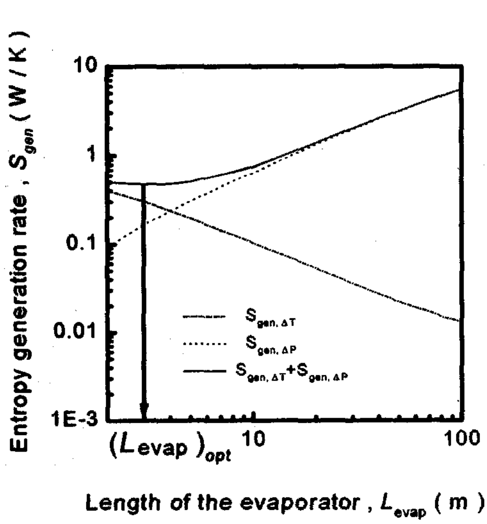

Fig.5 Variation of entropy generation for different length of the evaporator.

The characteristic of the heat pump operation for the variable heat load is important because the load is passively determined when the temperature of the outdoor air changes. The heat load can be accommodated by controlling the flow rate of the air through the indoor heat exchanger, i.e. the condenser in winter.

This section will describe the change of the heat pump cycles for the variable heat load:

(i) without inverter control

(ii) with inverter control

The following conditions, which assumed in the base-line cycle, apply to the calculation of cycle both without and with the inverter control.

- ▶ Isentropic efficiency of the compressor : $\eta_{COMP} = 0.8$
- ▶ Temperature of the inlet air in the condenser : 20 ℃
- ▶ Temperature of the outlet air in the condenser : 40 ℃
- ▶ Temperature of the inlet air in the evaporator : 0 ℃
- ▶ Nominal UA value in the condenser : 300.0 W/ ℃
- ▶ Nominal UA value in the evaporator : 300.0 W/ ℃
- ▶ Negligible pressure drop due to the flow of the refrigerant

Assumption for the overall heat transfer coefficient and the heat transfer area, UA in the heat exchanger

The values of UA in the condenser and in the evaporator are calculated under the following assumptions when the heat load changes.

① The heat transfer coefficient for the refrigerant in the heat exchanger does not strongly influence the variation of the value of UA since it is much larger than that for the air side. The thermal resistance by conduction through the heat exchanger tube also is negligible.

$$U_oA_o \approx h_oA_o \; (\Leftarrow h_i \gg h_o) \tag{2}$$

② The inside and outside area of the heat exchanger are almost same.

$$U_oA_o \approx U_iA_i \approx UA \tag{3}$$

③ The value of UA does not depend on the temperature of the refrigerant or the air.

$$U_o = U_o(\dot{m}_{air}, T) \approx U_o(\dot{m}_{air}) \tag{4}$$

④ The heat transfer coefficient of the air varies with the relation to the mass flow rate of the air as follows.

$$U_o \propto (\dot{m}_{air})^{0.8} \tag{5}$$

Variation of the heat load without inverter control

In case of the heat pump without an inverter, the flow rate of the refrigerant and the pressure ratio in the compressor are constant even if the heat load changes.

Figure 6 and 7 show the modified thermodynamic cycle when the heat load changes under the condition that the flow rate of the refrigerant and the air in the evaporator are fixed to the values for the base-line cycle.

As shown in Fig.6 and 7, the shape of the cycles with the variation of the heat load are largely different from the base-line cycle. This is due to the fact that each state in the cycle should change to satisfy the varying heat load under the condition that the flow rate of the refrigerant is constant.

The state 3 and 1 of the refrigerant are limited by the inlet temperature condition of the air in the counterflow heat exchangers. Fig.6 shows the cycle for the condition just before the temperature of the state 3 is lower than the inlet air temperature in the condenser(20 ℃) when the heat load is reduced to 4700 W. Fig.7 shows the cycle for the condition just before the temperature of the state 1 is higher than the inlet air temperature in the evaporator(0 ℃) when the heat load increases to 5100 W.

As shown in Fig.6 and 7, the thermodynamic cycle for the heat pump without an inverter largely changes if the heat load varies. Therefore, the cycle without an inverter has a very narrow operating range. In this study, the heat pump without an inverter is limited for the heat load between 4700 W and 5100 W.

Variation of the heat load with inverter control

In case of the inverter-driven operation, the flow rate control of the refrigerant is possible because of the capability of varying the rotational speed of the compressor using an inverter. The flow rate of the air in the evaporator should be also changed together with the flow rate of the refrigerant to keep the following conditions assumed in the base-line cycle.

- ▶ Inlet of the expansion valve : saturated liquid
- ▶ Inlet of the compressor : saturated gas

When the rotational speed of the compressor varies by means of the inverter, the pressure ratio in the compressor changes in addition to the mass flow rate of the refrigerant. The pressure ratio is assumed to change proportionally to the mass flow rate of the refrigerant.[8]

Figures 8 and 9 show the calculated heat pump cycle in case of the inverter-driven operation when the heat load changes between 4000 W and 6000 W. As can be seen in the figures, the thermodynamic cycle for the heat pump with an inverter does not change much compared to that without an inverter. This is attributed to the fact that the mass flow rate of the refrigerant can be modulated to satisfy

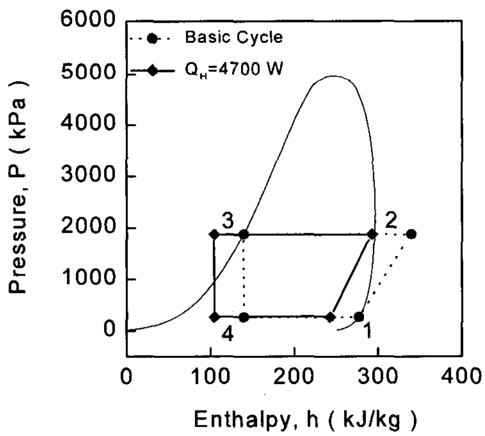

Fig.6 Cycle for different heating load without inverter (Q_H = 4,700 W).

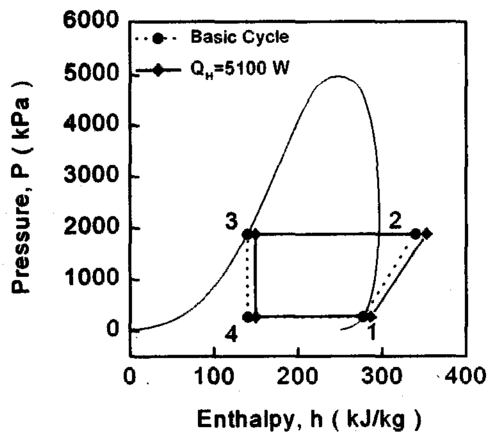

Fig.7 Cycle for different heating load without inverter (Q_H = 5,100 W).

the saturation conditions in the state 3 and 4. Therefore, the capability of the heat load variation is better than that of the case without an inverter. The flow rate of the air in the evaporator should vary widely to accomodate the changing refrigerant temperature and the exit air temperature.

Figure 10 shows the comparison of the thermodynamic performance between the case with an inverter and the case without an inverter. It is noticeable in Fig.10 that COP without an inverter is lower than that with an inverter above 5000 W.

On the other hand, COP without an inverter is higher than with an inverter below 5000 W. However, this cycle is very unrealistic because the compression process is partially inside the saturation dome. Even though the thermodynamic cycle formation is possible, it is unlikely to have such an operation due to the mechanical problem. Therefore, it is very difficult for the heat pump to cope with the varying heat load continuously without utilizing the inverter. If the mass flow rate of the refrigerant cannot change during the operation of the compressor, it has to be On/Off to match the heat load condition.

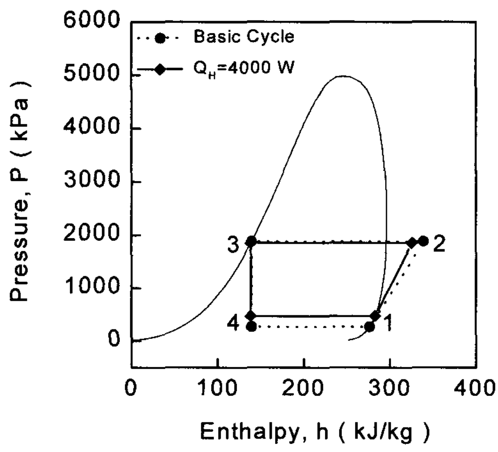

Fig.8 Cycle for different heating load with inverter (Q_H = 4,000 W).

3. Exergy loss during the ON/OFF operation

3.1. Modeling for the analysis

To calculate the exergy of the refrigerant in the condenser and the evaporator for the steady state, it is assumed that most of the refrigerant in the system exists in the condenser and the

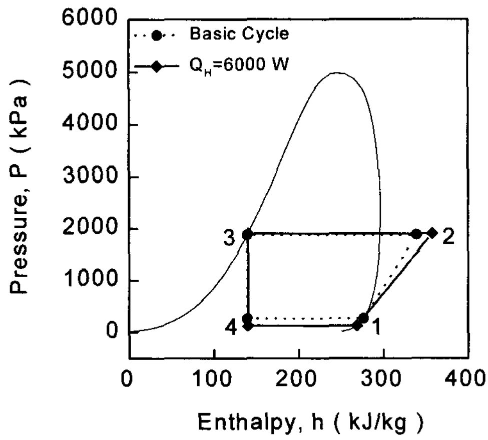

Fig.9 Cycle for different heating load with inverter (Q_H = 6,000 W).

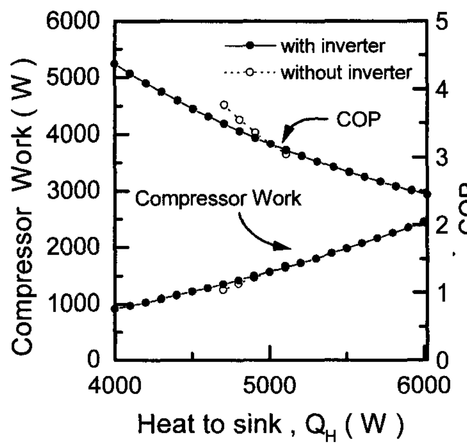

Fig.10 Variation of the compressor work and COP for different heat load.

evaporator. As shown in Fig.11, total exergy is calculated by summing up the exergy in the infinitesimal length over the total length. The exergy loss in the heat pump during the On/Off operation is obtained by comparing the accumulated exergy for the On state with that for the Off state. Whenever the heat pump is On and Off, there is an exergy loss associated with the dynamic response of the heat pump.[9]

3.2. Comparison of the exergy for the On/Off states

In this study, total exergy of the refrigerant in the heat pump system is calculated for the base-line cycle with the heat load of 5000 W. The condenser or the evaporator consists of the single tube that has the inner diameter of 11.1 mm and the outer diameter of 12.7 mm. As shown in Fig.4 and 5, 70 m and 4 m are the optimal length for the condenser and the evaporator respectively. However, in the present study, 50 m is selected as the heat exchanger length for both of the condenser and the evaporator. The following assumptions are made to calculate the exergy of the Off state:

① The refrigerant in the condenser and the evaporator exists in the uniform state respectively.
② The temperature of the refrigerant in the condenser is the same as that of the indoor air.
③ The temperature of the refrigerant in the evaporator is the same as that of the outdoor air.
④ The pressures in the condenser and the evaporator are the same.

The Off state calculation from these four conditions is presented in Fig.12. As shown in Fig.12, the pressure of the Off state is between the condensing pressure and the evaporating pressure of the On state. At this balanced pressure, which is 0.5 MPa, the refrigerant is thermally in equilibrium with the indoor and outdoor air respectively.

The numerical value of the total exergy diff-

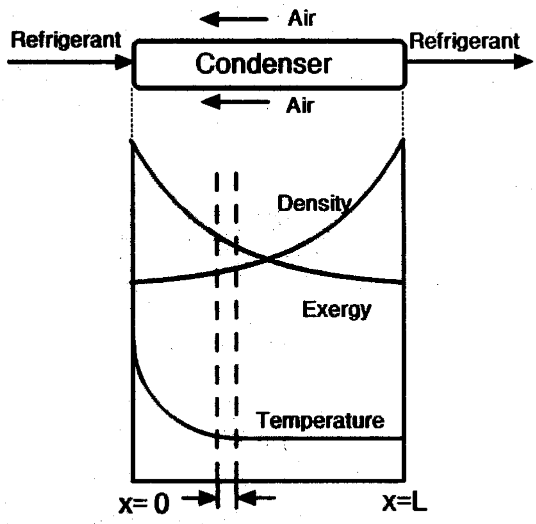

Fig.11 Schenmatic diagram for the exergy of the refrigerant in the condenser.

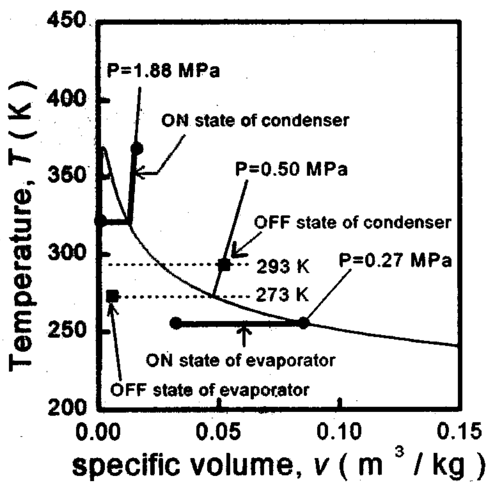

Fig.12 Temperature-specific volume diagram for ON/OFF states.

erence is listed in Table3. Note that the large amount of exergy in the condenser which is generated by the compressor through the electrical power is reduced to a small value in the Off state. This is the main exergy loss occurring between the On and Off states.

As a result of this calculation, it is evident that the heat pump in On/Off operations is always accompanied by the large exergy loss at each Off action. The switching action to the On state from the Off state may require the electrical power in the compressor corresponding to the exergy loss. This is why the heat pump with an inverter has superior performance to the heat pump without an inverter. There shall be no exergy loss associated with On/Off operations in the inverter driven unit.

4. Conclusion

From the results of the present study on the multi-type heat pump with variation of the he at load, the following conclusions can be drawn:

(1) Exergy of each state in the heat pump cycle is quite different according to the reference equilibrium state. It is reasonable that the values of exergy should be calculated on the basis of the outdoor state(0 ℃, 1 atm) in winter. The results of the calculation for each state in the cycle is presented in the exergy-enthalpy diagram, and the variation of the exergy along the cycle can be easily noticed from that diagram.

(2) When the heat load changes, the heat pump cycle with an inverter can be constructed in a relatively wide range without deviating extraordinarily from the optimized base-line cycle. On the other hand for the heat pump without an inverter, the heat pump cycle must change dramatically to cope with the variable heat load if it has to operate continuously. Since the efficiency decreases and there is a limit in the operational range, the On/Off control of the heat pump should be executed if the heat pump does not have an inverter. In case of the multi-type heat pump, an inverter is essential part for the efficient operation because of its wide range of the heat load.

Table 3 Exergy for On/Off states.

	On state		Off state	
	Condenser	Evaporator	Condenser	Evaporator
Mass of refrigerant	0.76 kg	0.09 kg	0.09 kg	0.76 kg
Exergy	43.3 kJ	3.0 kJ	3.7 kJ	30.1 kJ
Total exergy	46.3 kJ		33.8 kJ	

∴ Exergy loss between On/Off states = 46.3 - 33.8 = 12.5 kJ

(3) Exergy loss in the process of On/Off state is quantitatively examined through the computation of the exergy in the heat pump system. Frequent On/Off operations cause more exergy loss due to the more consumption of the electrical power. Part of the compressor work is wasted to restore the system to a steady state. An efficient operation can be achieved with an inverter when the heat load widely changes in that the exergy does not have to be lost by the unnecessary On/Off process.

(4) In the present study, the thermodynamic cycles with variable heat loads are considered by the computation of exergy and entropy at each state. Although the pressure drop, the superheat, the subcooling of the cycle, and the change of the property with variation of the outdoor temperature are not considered, this study

presents more accurate methodology for the exergy analysis of the heat pump.

Acknowledgements

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