DEVELOPMENT OF A PREDICTIVE MODEL FOR POWER CONSUMPTION OF AIR-TO-WATER HEAT PUMPS FOR RESIDENTIAL HOUSE

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ABSTRACT

A calculation model for air-to-water heat pump for space heating in residential house was developed. This model is assumed to be used for labelling or energy standards in Japan and parameters used in the model are assumed to be identified in the facility test by the manufactures. In order to simplify the test and decrease the identified parameters, the model used heat pump cycle model where the efficiency could be calculated by the evaporation temperature and the condensation temperature. Based on the steady-state experiments, the authors clarified that the energy consumption could be calculated by modifying the heat pump cycle model with three parameters: heat loss from refrigerant piping, compressor efficiency and auxiliary power consumption.

INTRODUCTION

For energy saving in residential sector, it is important to evaluate not only the performance of building envelope but also that of equipment and appliances. In Japan, the annual primary energy calculation programme[1] was developed in 2009 for the energy efficiency standard and labelling.

This programme consists of several calculation components, and the energy performance of air-to-water heat pumps for space heating in residential house can be calculated by using one of them (Figure 1). This calculation method was developed based on several experiments (Miura et al., 2009). With this method, power consumption, efficiency and maximum thermal output can be calculated based on experimental regression equations with the following parameters: thermal output, outdoor temperature and humidity, and supply water temperature[2] ((a) Experimental regression model in Figure 2).

In this model, the parameters should be identified by the tests with changing outdoor temperature and humidity, thermal output and water supply temperature separately in test laboratory, and thus many tests in different conditions should be carried out.

Two national institutes, namely BRI (Building Research Institute) and NILIM (National Institute for Land and Infrastructure Management) have carried out these tests in the laboratory and decided the coefficients in the equations in the model. However, these tests need much time and cost since many test cases are necessary to identify the parameters. In order to evaluate the performance of new-released heat pumps with higher efficiency or with different capacity hereafter, it is essential to develop the simple test with less reference measurement points, which can be easily carried out by the manufactures.

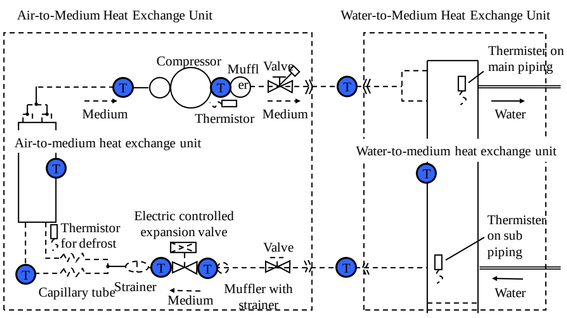
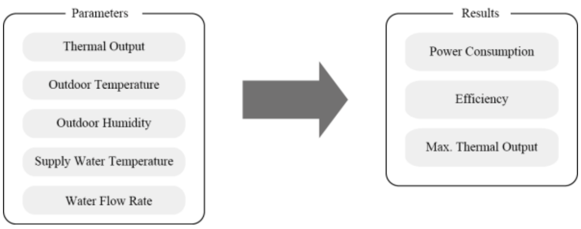
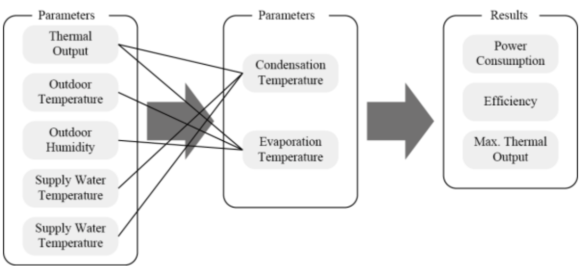


Figure 1 Circuit of air-to-water heat pump.
In Japan, it is common that the heat sources for heating and for DHW are equipped respectively.



(a) Experimental regression model



(b) Model with heat pump cycle model

Figure 2 Calculation models

Many models for heat pumps have been developed in many countries (e.g. IEA/ECBCS Annex 28 and DOE Programme), and some of them are detail models with enough accuracy. However, these models have many identified parameters and are therefore too complex to apply in the labelling although they are useful for research. On the other hand, there are simple models for labelling as seen in CEN standards or IPLV (Integrated Part Load Value Method). However, the operating condition assumed is different from Japanese operation, i.e. in Japan there are no relation between thermal output (capacity) and outdoor temperature since we usually operate heating equipments intermittently only while we stay the room. Therefore, a little detailed model with reasonable accuracy for labelling, which can be identified by some manufactures' tests, is needed, for evaluating the performance under various external conditions and various thermal outputs (capacities) separately.

For this purpose, we developed the calculation model of heat pump with high accuracy and with less tested parameters of the equations in the model.

In this paper, a series of experiments of an air-to-water heat pump in the artificial climate chamber is described and the developed calculation model is discussed.

OUTLINE OF CALCULATION MODEL

The power consumption is calculated by the factors of outdoor temperature and humidity, thermal output, and supply water temperature. Heat pump cycle model was applied and the calculation process was divided into three estimation steps ((b) Model with heat pump cycle model in Figure 2).

- The first step is the estimation of evaporation temperature and condensation temperature by thermal output, supply water temperature and outdoor temperature and humidity. This estimation mainly depends on the performance of the heat exchangers.
- The second step is the estimation of theoretical efficiency of heat pump cycle by the refrigerant temperatures at the evaporator and the condenser estimated in the first step. In this estimation, we assumed that heat pump cycle consists of four processes, which are adiabatic compression (isentropic process), condensation process at a constant pressure, isentropic expansion process and evaporating process at a constant pressure. We also assumed that there are thermal gains or losses only at the heat exchangers and there is no pressure loss of refrigerant medium.
- The last step is the calculation of the electric power consumption. The theoretical efficiency is compensated taking into account the mechanical loss of compressor. Then, the electric power consumption of the compressor is calculated by the compensated efficiency, thermal output and heat loss from refrigerant piping. Finally, the total electric power consumption is calculated by adding the auxiliaries' power consumption to the compressor's power consumption.

In this way, the power consumption can be calculated only based on the performance of the heat exchangers, the heat loss from the refrigerant medium piping, the compressor performance, and auxiliaries' power consumption, which can be identified by small numbers of tests.

EXPERIMENTS

We carried out a series of experiments in the artificial chamber in BRI. We installed an air-to-water heat pump and the heat load control system as shown in Figure 3, by which we can control the thermal output and the water flow rate. Table 1 shows the measured items. The measured interval was 1 second. The refrigerant temperatures were measured by thermocouples attached on the surface of the refrigerant pipe and wrapped by insulating materials.

Table 2 shows the specification of the tested heat pump which was bought in September 2008.

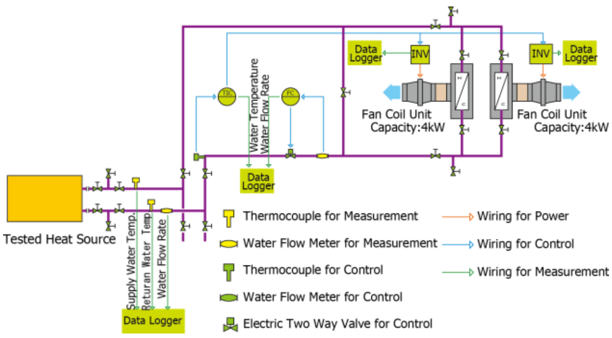


Figure 3 Heat load control system

The thermal output can be controlled by changing the number of the fan coil units with capacity of 4 kW and by changing the rotation of the attached inverter fans. Water flow rate also can be controlled by the magnetic valves with PID control.

Table 1 Measured items.

Items	Instruments [Specification / Manufacture]
Data Logger	Data Acquisition Unit [MW100/Yokogawa]
	Universal Input Module [MX110-UNV-M10 / Yokogawa]
	Pulse Input Module [S-M10 / Yokogawa]
Water Temperature	Platinum resistance thermometer [Pt100, Class-1 / Chino]
Refrigerant and air temperature	Thermocouple [type-T, Class-A / Chino]
Water Flow Rate	MagneW Neo+ [MTG11A / Yamatake]
Electric Power Consumption	Power High Tester [3332 / Hioki]
Humidity	Temperature and Humidity Sensor Unit [CHS-UPS／TDK]

Table 2 Specification of the tested heat pump.

Capacity for heating*1	6.0 kW
Electric Energy Consumption*1	1.5 kW
Efficiency*2	4.0
Refrigerant	R410A
*1 The value measured in the rated condition that the outdoor temperature is 7ºC, the supply water temperature is 40 ºC, the return water temperature is approx. 22 ºC and the water flow rate is 5L/min. *2 The value is calculated value.	

Two types of experiments were carried out in various outdoor conditions, in one of which we changed the thermal output with constant water flow rate, while in the other one, we changed the supply water flow rate.

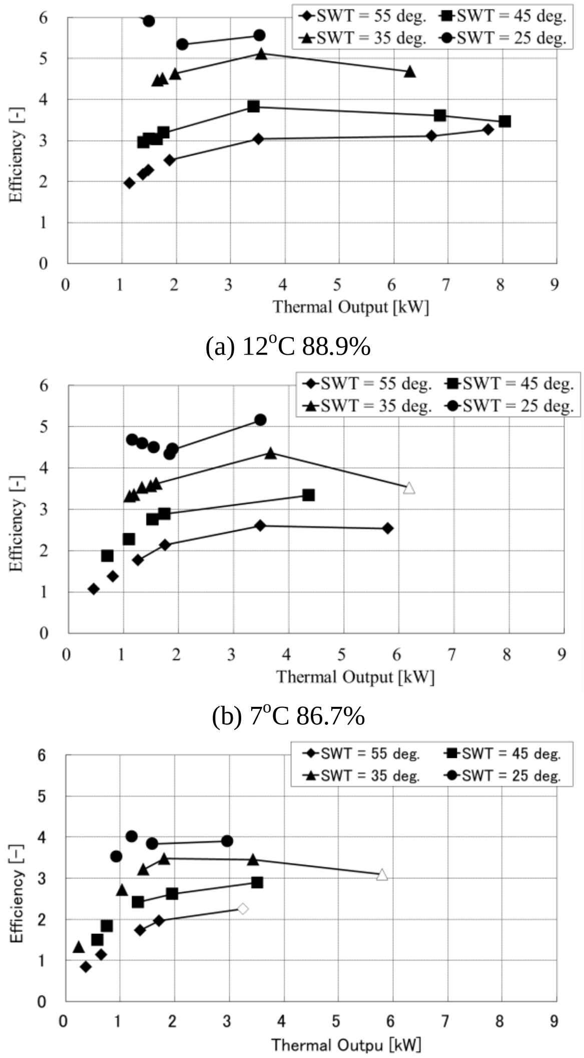
In the experiments with changing thermal output, supply water temperature was set by the controller of tested heat pump, water supply flow rate was kept at a constant value of 5L/min by water flow valves, and return water temperature was kept constant by controlling the inverter of the fan coil units. These experiments were conducted in the range of the thermal output where the tested heat pump operated continuously. In the range of the thermal output below the lower limit of continuous operation, we fixed the rotation of the fan coil units and we carried out the semi-steady-state experiments (where the thermal output was changing with a cycle, but the average thermal output in a period did not change). All experiments were carried out at three conditions: Temperature / Relative humidity is 12ºC / 88.9%, 7ºC / 86.7% and 2ºC / 83.7%.

In the experiments with changing water flow rate, we changed the flow rate in the range of ±40% to 5L/min with the external condition of 7ºC / 86.7%, and the supply water temperature of 35 ºC and 45 ºC.

The thermal output was calculated by the supply and return water temperature difference and the water flow rate. In the experiments where the tested heat pump operated stably, we used the average or integrated data over 30 minutes. In the experiments where the tested heat pump operated with defrosting with a cycle, we used the data in a period where we observed one or more cycles.

EXPERIMENTAL RESULTS

Figure 4 shows the results with water flow rate to be 5L/min. A set of the points not connected by straight lines is the result of the intermittent operation. Outline plots on white background represent the results with defrosting. In every operating condition, the efficiency is higher in the case with lower supply water temperature or higher outdoor temperature. The efficiency in the intermittent operation is lower than that in the continuous operation. The defrosting operation appeared in the experiments when the outdoor temperature was lower and the thermal output is higher. The higher the supply water temperature is, the lower the lower limit for the continuous operation is, and in the range of the thermal output below the approximately 20% of the rated thermal output, the intermittent operation appeared.



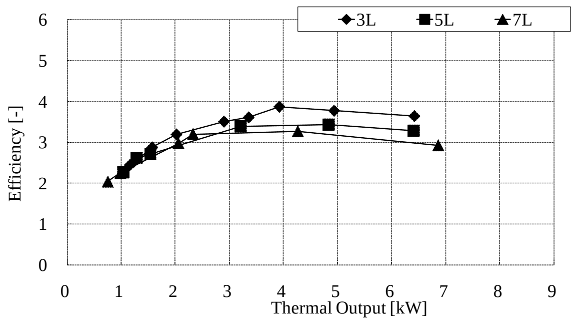
(a) 12^{o}C 88.9%

(b) 7^{o}C 86.7%

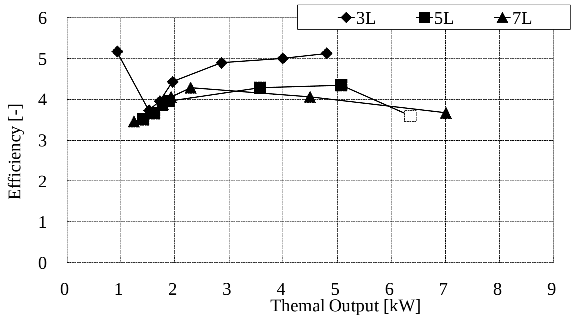
(c) 2^{o}C 83.7%

Figure 4 The thermal output and the efficiency with the supply water flow rate of 5L/min. (SWT is the sypply water temperature.)

Figure 5 shows the efficiency and the thermal output with changing water flow rate. When comparing the results with the same thermal output, the lower the water flow rate is, the higher the efficiency is. The reason is that the lower the water supply flow rate is, the lower the condensation temperature is, because of the lowering of the return water temperature, and generally, the efficiency was higher with lower condensation temperature. Decrease of the electric power consumption for water supply pump may be also the reason.



(a) Targeted supply water temperature = 45°C



(b) Targeted supply water temperature = 35°C

Figure 5 The thermal output and the efficiency with changing the supply water flow rate.

ESTIMATION FOR REFRIGERANT TEMPERATURE AT CONDENSER

We needed to measure the condensation temperature for developing the calculation method, because condensation temperature and evaporation temperature are important in the heat pump cycle model. It is easy to measure it in case of air-to-air heat pump such as room air conditioner (e.g. by measuring the surface temperature of the refrigerant pipe). However, in case of the air-to-water heat pump, it is difficult to measure the temperature at the condenser because plate heat exchanger is normally used. Therefore, we estimated the condensation temperature by other measured temperatures[3].

There are two cases of the refrigerant medium at the outlet of the condenser are liquid and gas-liquid equilibrium respectively. We described two estimation methods for each case. For the estimation, we set the assumption as described below:

A1: At the centre of the evaporator, the refrigerant medium is gas-liquid equilibrium.

A2: At the inlet of the compressor and condenser, the refrigerant medium is gas.

A3: There is no pressure loss in the refrigerant piping, i.e. the pressures at the inlet and outlet of the evaporator and the inlet of the compressor are equivalent, and the pressures at the inlet and outlet of the condenser and the outlet of the compressor are equivalent.

A4: All amount of the energy of the electric power consumed at the compressor transmits to the refrigerant medium.

Table 3 shows the measurement points used in this estimation.

Table 3 Measured points

Item	Symbol
Inlet refrigerant temp. of compressor	$T_{comp,in}$
Outlet refrigerant temp. of compressor	$T_{comp,out}$
Inlet refrigerant temp. of condenser	$T_{cnd,in}$
Outlet refrigerant temp. of condenser	$T_{cnd,out}$
Refrigerant temp. at the centre of evaporator	T_{evp}
Power consumption of compressor	P_{comp}
Thermal output of the heat pump	Q_{out}

Case 1: Refrigerant is liquid at outlet of condenser.

Power consumption of compressor P_{comp} is calculated by the inlet and outlet enthalpy of the compressor $h_{comp,in}$ and $h_{comp,out}$, and refrigerant flow rate v_{ref} in Equation (1).

$$P_{comp} = \left(h_{comp,out} - h_{comp,in}\right) \times v_{ref} \tag{1}$$

Thermal output Q_{out} is calculated by the inlet and outlet enthalpies of condenser $h_{cnd,in}$ and $h_{cnd,out}$, and refrigerant flow rate v_{ref} in Equation (2).

$$Q_{out} = \left(h_{cnd,in} - h_{cnd,out}\right) \times v_{ref} \tag{2}$$

Equation (3) is given by Equation (1) and (2).

$$\frac{P_{comp}}{Q_{out}} = \frac{h_{comp,out} - h_{comp,in}}{h_{cnd,in} - h_{cnd,out}} \tag{3}$$

The enthalpy $h_{comp,in}$ is calculated by the inlet pressure $p_{comp,\ in}$ and inlet temperature $T_{comp,in}$ of compressor by Equation (4). Here let the pressures at the inlet of the compressor and at the evaporator are equivalent based on A3, which we can calculate by temperature at the evaporator T_{evp} (Equation (5)).

$$h_{comp,in} = h_g\left(p_{comp,in}, T_{comp,in}\right) \tag{4}$$

$$p_{com,in} = p_{evp} = p_{gl}\left(T_{evp}\right) \tag{5}$$

Here, the function h_g stands for the relation between pressure, enthalpy and temperature when refrigerant is liquid. The function p_{gl} represents the relation between temperature and pressure when refrigerant is gas-liquid equilibrium.

Enthalpies $h_{comp,out}$, $h_{cnd,in}$ and $h_{cnd,out}$ can be calculated by pressure p_{cnd} and temperatures $T_{comp,out}$, $T_{cnd,in}$ and $T_{cnd,out}$ by Equation (6), (7) and (8). Pressure p_{cnd} can be calculated by temperature T_{cnd} by Equation (9). However, temperature T_{cnd} is not measured. Therefore, now we assume a given temperature.

$$h_{comp,out} = h_g\left(p_{cnd}, T_{comp,out}\right) \quad (6)$$

$$h_{cnd,in} = h_g\left(p_{cnd}, T_{cnd,in}\right) \quad (7)$$

$$h_{cnd,out} = h_{ll}\left(T_{cnd,out}\right) \quad (8)$$

$$p_{cnd} = p_{gl}\left(T_{cnd}\right) \quad (9)$$

Here, the function h_{ll} stands for the relation between temperature and enthalpy when the refrigerant is liquid.

Equation (10) is given by Equation (3), (6), (7), (8) and (9).

$$\frac{P_{com}}{Q_{out}} = \frac{h_g\left(p_{gl}\left(T_{cnd}\right), T_{comp,out}\right) - h_g\left(p_{gl}\left(T_{evp}\right), T_{comp,in}\right)}{h_g\left(p_{gl}\left(T_{cnd}\right), T_{cnd,in}\right) - h_{ll}\left(T_{cnd,out}\right)} \quad (10)$$

Since we can calculate all the terms in Equation (10) excepting T_{cnd}, we consequentlly decide T_{cnd} which satisfy Equation (10).

Here, since we assume that refrigerant at the outlet of the condenser is liquid, $T_{cnd,out} < T_{cnd}$. Obviously, $T_{cnd} < T_{cnd,in}$. Therefore, T_{cnd} should satisfy Equation (11).

$$T_{cnd,out} < T_{cnd} < T_{cnd,in} \quad (11)$$

If this equation is not satisfied, the assumption is wrong and we should adopt another assumption that refrigerant at the outlet of the condenser is gas-liquid equilibrium, described in the next paragraph.

Case 2: Refrigerant is gas-liquid equilibrium at outlet of condenser.

In case that refrigerant at the outlet of the condenser is gas-liquid equilibrium, $T_{cnd} = T_{cnd,out}$.

Hereinafter, we will describe how to check whether the refrigerant at the outlet of the condenser is gas-liquid equilibrium or not.

Inlet enthalpy of compressor $h_{comp,\ in}$ is calculated by Equation (4) and (5).

Pressure at the condenser p_{cnd} is calculated by the temperature T_{cnd}, which equals to $T_{cnd,out}$.

$$p_{cnd} = p_{gl}\left(T_{cnd,out}\right) \quad (12)$$

Outlet enthalpy of the compressor $h_{com,out}$ is calculated by p_{cnd} and $T_{com,out}$ by Equation (13).

$$h_{comp,out} = h_g\left(p_{cnd}, T_{com,out}\right) \quad (13)$$

Inlet enthalpy of the condenser $h_{cnd,in}$ is calculated by p_{cnd} and $T_{cnd,in}$ by Equation (14).

$$h_{cnd,in} = h_g\left(p_{cnd}, T_{cnd,in}\right) \quad (14)$$

Equations (3), (12), (13) and (14) give Equation (15).

$$\frac{P_{com}}{Q_{out}} = \frac{h_g\left(p_{gl}\left(T_{cnd,out}\right), T_{comp,out}\right) - h_g\left(p_{gl}\left(T_{evp}\right), T_{comp,in}\right)}{h_g\left(p_{gl}\left(T_{cnd,out}\right), T_{cnd,in}\right) - h_{cond,out}} \quad (15)$$

Since we can calculate all terms in Equation (15) excepting $h_{cnd,out}$, we consequentlly decide $h_{cond,out}$ which satisfy Equation (15).

Here, based on the assumption that refrigerant at the outlet of the condenser is gas-liquid equilibrium, the calculated enthalpy $h_{cnd,out}$ is greater than and equals to the enthalpy when the refrigerant is liquid at $T_{cnd,out}$:

$$h_{cnd,out} > h_{ll}\left(T_{cnd,out}\right) \quad (16)$$

If enthalpy at the outlet of the condenser does not satisfy Equation (16), the assumption is wrong and refrigerant should be calculated under the assumption that refrigerant at the outlet of the condenser is liquid described in the preceding paragraph.

Decision of phase of refrigerant medium

In the paragraphs above, the estimation for the condensation temperature is described both in the cases that the refrigerant is liquid and gas-liquid equilibrium. Then, the applicable condition for condensation temperature is described in Equation (11) and (16). If the both cases satisfy these conditions, we decided that the refrigerant medium is liquid, because the efficiency is higher and controllability for electric expansion valve is better. Moreover, considering the recent technology for controlling the electric expansion valve and inverter compressor, the assumption that heat cycle is operated with supercooling is more proper.

BASIC EQUATION FOR ENERGY CONSUMPTION CALCULATION

We define compressor efficiency e_{comp} by:

$$e_{comp} = \frac{Q_{cnd}}{P_{comp}} \quad (17)$$

where P_{comp} is the power consumption of the compressor and does not include the auxiliary power consumption, and Q_{cnd} is condensation output. The condensation output Q_{cnd} include thermal output Q_{out} and heat loss from the refrigerant piping Q_{loss}. Therefore, Q_{cnd} is represented by:

$$Q_{cnd} = Q_{out} + Q_{loss} \quad (18)$$

We denote the compressor efficiency e_{comp} by theoretical heat pump cycle efficiency e_{th} and coefficient ε in Equation (19). Theoretical heat pump cycle efficiency e_{th} can be calculated by the condensation temperature T_{cnd} and the evaporation temperature T_{evp} under theoretical heat pump cycle, which heat pump compression is assumed to be adiabatic compression. We calculated e_{th} with the assumption that the degree of supercooling is 5 ^{o}C and the degree of superheating is 0 ^{o}C. Efficiency ε represents various losses. Under actual heat pump cycle, mechanical loss at compressor and pressure loss of refrigerant medium piping exist. We denote these losses by ε (0< ε <1) and given by:

$$e_{comp} = \varepsilon \times e_{comp,th} \quad (19)$$

Power consumption P is represented by:

$$P = P_{comp} + P_{aux} \quad (20)$$

where P_{aux} is auxiliary power consumption for the fan in the air-to-medium heat exchange unit and water supply pump etc.

Energy consumption P is consequently calculated by Equation (21) via Equation (17), (18), (19) and (20).

$$P = \frac{Q_{out} + Q_{loss}}{\varepsilon \times e_{comp,th}} + P_{aux} \quad (21)$$

Eventually, in order to calculate energy consumption P by a given thermal output Q_{out}, we need to identify Q_{loss}, ε and P_{aux}. We do not need to identify the theoretical heat pump cycle efficiency e_{th} because it is calculated theoretically by evaporation and condensation temperatures, T_{evp} and T_{cnd}.

IDENTIFICATION FOR PARAMETERS IN BASIC EQUATION

Heat loss from refrigerant medium piping Q_{loss}

Figure 6 shows heat losses and gains from refrigerant medium piping and the heat exchanged at evaporator and condenser. Since condensing heat is the sum of $Q_{loss,1}$, $Q_{loss,2}$, $Q_{loss,3}$, $Q_{loss,4}$ and $Q_{loss,5}$, heat loss Q_{loss} is the sum of $Q_{loss,1}$, $Q_{loss,2}$, $Q_{loss,4}$ and $Q_{loss,5}$. $Q_{loss,1}$ is included as the degradation of the compressor efficiency and $Q_{loss,5}$ can be neglected because the surface area on the piping is very small. Therefore, heat loss Q_{loss} is given by:

$$Q_{loss} = Q_{loss,2} + Q_{loss,4} \quad (22)$$

Q_{loss} is assumed to be proportional to the difference between refrigerant temperature and external temperature. Here, we represent the refrigerant temperature by T_{cnd}. Then, Q_{loss} is given by:

$$Q_{loss} = k \times \left(T_{cnd} - T_{ext}\right) \quad (23)$$

Figure 7 shows the relation between the calculated Q_{loss} in Equation (23) and the measured Q_{loss}. The measured Q_{loss} is the sum of $Q_{loss,2}$ and $Q_{loss,4}$ and we calculated $Q_{loss,2}$ by the refrigerant temperature at the inlet of the condenser and the refrigerant temperature at the outlet of the compressor. We could not calculate $Q_{loss,3}$ by the measured data because we did not measured the refrigerant temperature at the inlet of the expansion valve. Therefore, we firstly compared (a) the temperature difference between refrigerant temperature at the inlet of the condenser and the external temperature and (b) the temperature difference between refrigerant temperature at the outlet of the condenser and the external temperature. Since the temperature difference (b) is half of (a) and the piping length between the outlet of compressor and the inlet of the condenser is approximately the same as the piping length between the outlet of condenser and the inlet of the expansion valve, we assumed that $Q_{loss,4}$ is half of $Q_{loss,2}$.

Figure 7 shows that heat loss Q_{loss} is approximately proportional to the difference between the condensation temperature and the external temperature.

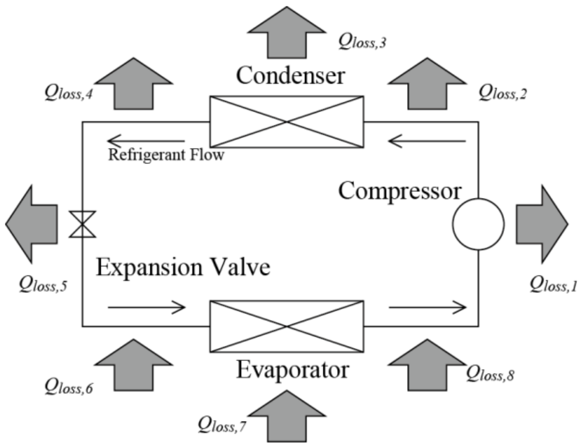


Figure 6 Heat losses and gains from refrigerant piping and heat exchanged at condenser and evaporator

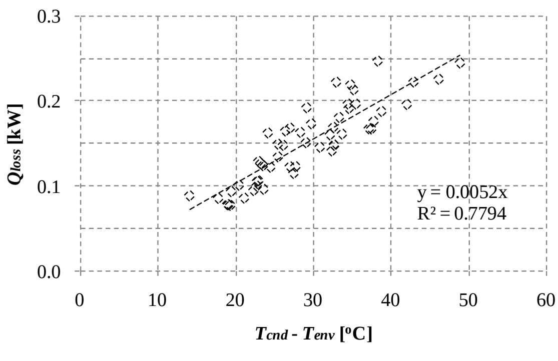


Figure 7 Calculated and measure Q_{loss}

Auxiliary power consumption

Auxiliary components are composed of supply water pump, fan at the air-to-medium heat exchange unit and other controls. We measured the energy consumption of the supply water pump and other controls when test-drive operation. Since we could not measure the fan at the air-to-water heat exchanger, we gave it the catalogue value of the motor for the fan. Eventually, we let the auxiliary power consumption be constant of 0.11[kW].

Coefficient of degradation of heat pump cycle ε

Figure 8 shows the relation between theoretical heat pump efficiency e_{th} and measured compressor efficiency e_{comp}. The theoretical efficiency e_{th} is calculated by the measured evaporation temperature and the calculated condensation temperature. The e_{comp} is calculated by:

$$e_{comp} = \frac{Q_{out} + Q_{loss}}{P - P_{aux}} \tag{24}$$

In Figure 8, there is good correlation between e_{th} and e_{comp}. However, e_{th} is higher than e_{comp} because the losses are not included. We represented these losses by coefficient of degradation of heat pump cycle ε. Large part of these losses is the losses at the compressor, which consist of adiabatic efficiency, mechanical efficiency and volumetric efficiency of the motor. We assumed these efficiencies depend on the inlet and outlet refrigerant medium pressure. In this paper, for simplicity, we assumed that efficiency ε depends mainly on condensation temperature T_{evp}, and given by:

$$\varepsilon = a \times T_{evp} + b \tag{25}$$

Figure 9 shows ε identified by the measurement data and condensation temperature T_{evp}. It is shown that ε has approximately linear relationship to T_{evp}.

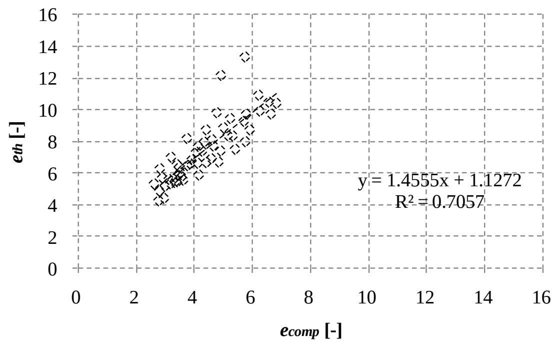


Figure 8 Relation between theoretical heat pump efficiency and measured compressor efficiency

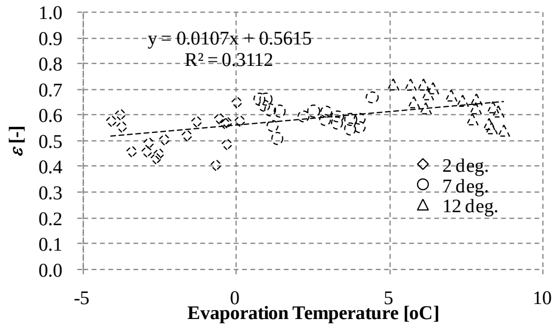


Figure 9 Relation between evaporation temperature and coefficient ε

RESULTS

Figure 10 shows the calculated and measured energy consumption. We gave the parameters in the equations in order that calculated values have good correlation to the measured one. However, the calculated one has good relationship to the measured one. Therefore, we can conclude that the developed model can be useful for calculating the energy consumption, as long as the parameters can be identified properly.

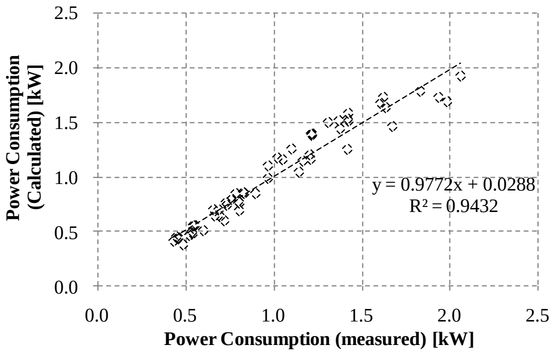


Figure 10 Calculated and measured energy consumption

ESTIMATION FOR REFRIGERANT TEMPERATURE

Since the final goal is the development of the energy calculation by only the thermal output, external temperature (and humidity for defrosting operation), and water supply temperature, we have to develop the estimation method that we can calculate the refrigerant temperatures. Figure 11 and 12 show the result of the refrigerant temperatures. The performance of the heat exchanger units has great effect on the relation between refrigerant temperatures and the parameters such as external temperature and supply water temperature. Since we tested only one type of the heat exchangers of heat pump system, we need to correct the data of various types of heat exchanges for developing the estimation method in further research.

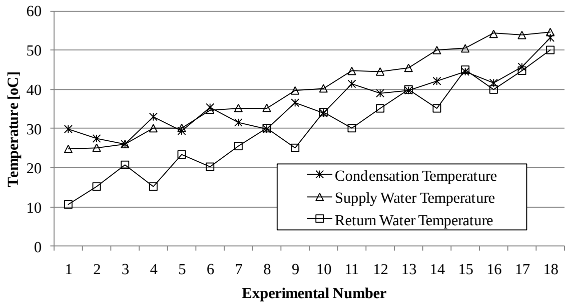


Figure 11 Condensation Temperature

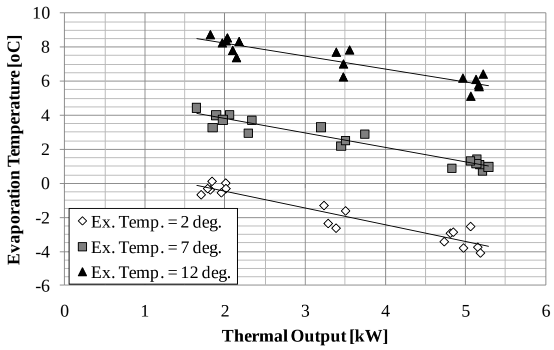


Figure 12 Evaporation Temperature

CONCLUSION

We developed the estimation method for water-based heat pump for space heating. In the developed estimation, the energy consumption can be calculated by four parameters: heat loss from the refrigerant piping Q_{loss}, coefficient for the efficiency degradation of the compressor ε and the auxiliary power consumption P_{aux} and the theoretical heat pump efficiency e_{th} which can be calculated by the condensation temperature and evapolation temperature. We showed that the developed model could be useful for calculating the energy consumption, as long as the parameters could be identified properly.

Finally, we need to develop the practical tests for the identification of the parameters as follows:

i) Air-to-water heat exchanger unit (condenser) performance test

ii) Water-to-water heat exchanger unit (evaporator) performance test

iii) Heat loss from the refrigerant piping identification test

iv) Compressor efficiency test for identifying ε

v) Auxiliary power consumption test

NOMENCLATURE

e_{comp}	Compressor efficiency, [-]
e_{th}	Theoretical heat pump cycle efficiency, [-]
$h_{cnd,in}$	Inlet enthalpy of condenser, [J/kg]
$h_{cnd,out}$	Outlet enthalpy of condenser, [J/kg]
$h_{comp,in}$	Inlet enthalpy of compressor, [J/kg]
$h_{comp,out}$	Outlet enthalpy of compressor, [J/kg]
p_{cnd}	Pressure of condenser, [Pa]
$p_{comp,in}$	Inlet pressure of compressor, [Pa]
p_{evp}	Pressure of evaporator, [Pa]
P	Power consumption of heat pump, [W]
P_{aux}	Auxiliary power consumption, [W]
P_{comp}	Power consumption of compressor, [W]
Q_{cnd}	Condensation output, [W]
Q_{loss}	Heat loss from refrigerant piping, [W]
Q_{out}	Thermal output of the heat pump, [W]
T_{cnd}	Refrigerant temperature at the centre of condenser, Condensing temperature, [^{o}C]
$T_{cnd,in}$	Inlet refrigerant temperature of condenser, [^{o}C]
$T_{cnd,out}$	Outlet refrigerant temperature of condenser, [^{o}C]
$T_{comp,in}$	Inlet refrigerant temperature of compressor, [^{o}C]
$T_{comp,out}$	Outlet refrigerant temperature of compressor [^{o}C]
T_{evp}	Refrigerant temperature at the centre of evaporator, Evaporation temperature, [^{o}C]
v_{ref}	Refrigerant flow rate, [kg/s]
ε	Coefficient representing the degradation of heat pump cycle efficiency, [-]

Function

h_g	Function for calculate enthalpy by pressure and temperature for gas refrigerant
p_{gl}	Function for calculate pressure by temperature for gas-liquid equilibrium refrigerant
h_{ll}	Function for calculate enthalpy by temperature for liquid refrigerant

NOTE

1 The programme can be found on the website http://ees.ibec.or.jp. By this programme, we can calculate the annual energy consumption for all energy usage: heating, cooling, ventilation, DHW, lighting, PV and cogeneration, excluding the electric home appliances and cooking.

2 The calculation is bellow:

Efficiency e is given by:

$$e=\left(\left(c_o+c_1\times\Delta\theta\right)\times\left(1-r\right)^2+\left(c_2+c_3\times\Delta\theta\right)\times\left(1-r\right)+1\right)$$
$$\times\frac{q_{max}}{P_{max}}\times e_{rtd}$$

where,

r : partial output ratio

(= output / maximum output)

$\Delta\theta$:the temperature difference between supply water temperature and outdoor temperature [^{o}C]

P_{max} : The maximum power [W]

c0 ~ c3 was defined by the experiments as below:

c0	c1	c2	c3
1.120656	-0.03703	-.036786	0.012152

3 If we measure the pressure of the refrigerant, we can estimate the refrigerant temperature in the condenser. However, in this research, we did not measure the pressure because we assumed that it was difficult to make the hole for measuring the pressure on plate heat exchanger unit.

REFERENCES

IEA/ECBCS Annex 28 http://www.annex28.net/

Miura, et al. 2009. Explanation for Energy Efficiency Standard, IBEC http://ees.ibec.or.jp (2011).