On-site Performance Test and Simulation of a 10 RT Air Source Heat Pump

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Key words: On-site, Performance test, Simulation, Air source heat pump, Heat and mass transfer

ABSTRACT: In this study, on-site performance test of an air source heat pump which has a rated capacity of 10 RT is carried out. Since indoor and outdoor air conditions can not be controlled to satisfy the standard test conditions, experiments are done with the inlet air conditions as they exist. To estimate the performance of the heat pump for other conditions, the heat pump is modeled with a small number of characteristic parameters. The values of the parameters are determined from the few measurements measured on-site during steady operation. A simulation program is developed to calculate cooling capacity and power consumption at any other arbitrary operating conditions. The simulation results are in good agreement with the experiment. This study provides a method of an on-site performance diagnosis of an air source heat pump.

Nomenclature

A : area [m^2]
$\hat{a}$: ratio of enthalpy variation to temperature variation of saturated moist air [kJ/kg·K]
c_p : specific heat [kJ/(kg·K)]
dT_{sub} : subcooling [K]
dT_{sup} : superheating [K]
e : enthalpy [kJ/kg]
e^* : enthalpy of saturated moist air [kJ/kg]
h : heat transfer coefficient [kW/(m^2·K)]
h_m : mass transfer coefficient [kg/(m^2·s)]
$\dot{m}$: mass flow rate [kg/s]
P : pressure [kPa]
Q : heat transfer rate [kW]
Q_{loss} : compressor heat loss [kW]
RH : relative humidity [%]
SHF : sensible heat factor [%]
T : temperature [K]
V : volumetric flow rate [m^3/s]
v : specific volume [m^3/kg]
W : compressor power [kW]
w : humidity ratio [kg/kg$_{DA}$]

Greek symbols

ε : clearance volume ratio
ε' : experimental constant, Eq. (3)
η_m : mechanical efficiency
η_s : isentropic efficiency; fin efficiency
η_v : volumetric efficiency
ρ : density [kg/m^3]

Subscripts

a : air
c : condenser
dis : discharge
e : evaporator

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EA : exhaust air
i : tube inner wall
in : inlet
o : tube outer wall
OA : outdoor air
out : exit
r : refrigerant
RA : return air
sat : saturation
suc : suction
w : tube wall

1. Introduction

Due to the concern about the depletion of natural resources and environmental problem, considerable attention on energy saving has grown up and a great number of research related to this field has been carried out. Recently, government has enormously invested on ESCO (Energy Service Company) for promoting energy saving projects effectively. With the ESCO policy, cost for applying energy saving technologies is loaned from the government and this loan is repaid by the diminished cost of energy consumption, resulting in win-win strategy for all parties involved—government, ESCO and building owners. Though ESCO has a wide range of business, HVAC (heating, ventilation and air-conditioning) field is most favorable because for a typical office building, energy spent on HVAC accounts for about 47% of all energy consumption. The objective of ESCO business in HVAC systems is to suggest energy saving strategies such as optimum operating methods or system remodeling based on the on-site energy consumption diagnosis and analysis. Therefore, above all, actual energy consumption data of the target building or system should be collected for proper energy saving strategies. Among various HVAC (heating, ventilating and air-conditioning) system, refrigeration system is one of the most energy consuming components. Therefore accurate on-site diagnosis of refrigeration system is very important. But at present, diagnosis of a refrigeration system is roughly done with a few measurement of capacity and power consumption, which are collected in a short period of time. The performance of a refrigeration system is affected by outdoor/indoor conditions and operation methods, thus a few experimental data which are collected briefly can not be used as performance index. Moreover, energy saving strategies based on such data can not be reliable. Performance should be measured based on standard test method and experimental conditions. For small systems such as domestic air-conditioners, standard test method and conditions to acquire objective performance index are described in KS B 6369,(1) KS C 9306,(2) ANSI/ASHRAE Standard 37(3) and ANSI/ARI Standard 210/240.(4) These standards specify experimental conditions which include evaporator and condenser inlet air conditions, and recommend at least one test room to realize them. But the medium-sized heat pumps installed in the field, which can be regarded as a primary target of ESCO business, can not be brought into the test room, nor can indoor/outdoor conditions be controlled. Therefore, the performance of target systems should be evaluated through a proper system modelling based on the experimental results by on-site arbitrary test conditions. Numerous research on the system modelling and simulation method for a refrigeration system has been carried out by many researchers. Kim et al.(5) and Kim et al.(6) carried out system simulations with a thermodynamic modelling for an optimal system design. Lee et al.(7) developed an automotive air-conditioning system performance analysis program which was based on the experimental results of a compressor and a heat exchanger, and verified their program with experimental results. But most of related research were on a design point of view and included a lot of equations for more accurate modelling. Therefore, a few on-site experimental data is not enough to utilize the fore-mentioned complex modelling. On-site experiments have less information about the system and more uncertainties in measurement than experiments in the laboratories. For example, accurate specification of a heat exchanger or a compressor performance curve are rarely available for a system operating in the field. And in many cases, desired locations for measurement may not be accessible. Therefore, for the development of performance prediction modelling based on the on-site experimental results, a simple model with minimum measurement and information seems to be more appropriate than a complex model with complicated heat and mass transfer correlations. In this study, on-site test method and performance prediction modelling technique of an air source heat pump with a reciprocating compressor is proposed. The performance prediction by the simulation model was also verified with experimental results from on-site performance measurement.

2. Component modelling

2.1 Compressor

The most convenient and accurate way to estimate the performance of a compressor is to refer to the characteristic curve. But, this is possible only if the performance curve is available. In most cases, characteristic curve of compressors used for a long time in the field is not available and besides they may have different characteristics from what the performance curve indicates. Thus for best prediction, a simple thermodynamic model with proper characteristic parameters should be used. Compressor performance can be presented by discharged gas mass flow rate, temperature and power consumption. For a reciprocating compressor, discharged gas mass flow rate is given by Eq. (1), and volumetric efficiency can be presented as Eq. (2) with a constant α which accounts for the leakage and pressure drop at the suction port.

$$\dot{m} = \rho_{suc}\, \eta_v\, V \tag{1}$$

$$\eta_v = \alpha \cdot \left[1 - \varepsilon\left(\frac{v_{suc}}{v_{dis}} - 1\right)\right] \tag{2}$$

Assuming α as 1, and with experimentally determined ε, Eq. (2) can be used with satisfactory results.[8] Therefore, volumetric efficiency can be approximated as Eq. (3) with experimental constant ε'. The relation among isentropic efficiency, power consumption and discharged gas temperature can be presented as Eq. (4) and (5).

$$\eta_v \fallingdotseq 1 - \varepsilon'\left(\frac{v_{suc}}{v_{dis}} - 1\right) \tag{3}$$

$$\eta_s = \frac{\dot{m}_r \cdot (e_{P=P_c, s=s_{suc}} - e_{suc})}{W} \tag{4}$$

$$W = \frac{\dot{m}_r \cdot (e_{dis} - e_{suc})}{\eta_m} \tag{5}$$

Approximating air flow near the compressor as cross flow across the cylinder, heat transfer coefficient on the shell is proportional to exponent of 0.6 of the air velocity.[9] Compressor heat loss can then be approximated as Eq. (6) using constant B with assumptions of fixed compressor surface area and negligible thermodynamic property variation within one season. Mechanical efficiency can then be represented as Eq. (7).

$$Q_{loss} \fallingdotseq B\, \dot{m}_{OA}^{0.6} (T_{dis} - T_{OA}) \tag{6}$$

$$\eta_m = \frac{W - B\, \dot{m}_{OA}^{0.6} (T_{dis} - T_{OA})}{W} \tag{7}$$

2.2 Heat exchanger

There are a lot of heat exchanger models such as three region model, finite element model, parametric model, and so on.[10] Especially, tube-by-tube method proposed by Domanski[11] is widely used for finned-tube heat exchanger due to its accuracy and simplicity. Since most air source heat pump uses finned-tube heat exchanger, tube-by-tube method is recommended. The principle of the this method is as follows. First, divide the whole heat exchanger into single tubes, and analyze them respectively as a single tube cross flow heat exchanger with an assumption of refrigerant-side mixed and air-side unmixed. Yoon et al.[12] also confirmed that his simulation model using tube-by-tube method predicts heat transfer rates within ±5% error. But, this method requires detailed geometric specifications of a heat exchanger. The detailed information about the heat exchangers in the field is not always available, so this method can not be employed for on-site performance test. Besides for system in field operation, non-uniformity of frontal air velocity and refrigerant maldistribution always exist,[13] which are deviations from the assumptions that most analyses are based on. For the condenser, heat transfer rate Q_c is given by Eq. (8) with saturated condensation temperature $T_{c,sat}$ as a representative refrigerant temperature and $T_{c,w}$ as a representative tube wall temperature. A heat transfer coefficient for an internal flow is proportional to exponent of 0.8 of mass flow rate. Then Q_c can be approximated as Eq. (9) using constant C_1.

$$Q_c = h_i A_i (T_{c,sat} - T_{c,w}) \tag{8}$$

$$Q_c \fallingdotseq C_1 \dot{m}_r^{0.8} (T_{c,sat} - T_{c,w}) \tag{9}$$

Q_c can be approximated as Eq. (10) with empirical constants C_2 and C_3.

$$Q_c \fallingdotseq \left(\frac{1}{C_1 \dot{m}_r^{0.8}} + \frac{1}{C_2 \dot{m}_{OA}^{0.6}} + C_3 \right)^{-1} \times \left\{ \frac{(T_{c,sat} - T_{a,c,in}) - (T_{c,sat} - T_{a,c,out})}{\ln\left(\frac{T_{c,sat} - T_{a,c,in}}{T_{c,sat} - T_{a,c,out}} \right)} \right\} \tag{10}$$

A similar analogy can be applied to an evaporator. Due to the condensation of water vapor, however, heat transfer rate is proportional to the difference between saturated moist air enthalpy which is calculated at the corresponding temperature. $\hat{a}$ in Eq. (11), which represents the enthalpy variation with respect to temperature variation of saturated moist air, can be computed by Eq. (12). In the same manner, heat transfer rate of an evaporator can be approximated as Eq. (13) with empirical constants D_2 and D_3.

$$Q_e = h_i A_i (T_{e,w} - T_{e,sat}) \fallingdotseq \frac{D_1 \dot{m}_r^{0.8}}{\hat{a}} (e^{*_{e,w}} - e^{*_{e,sat}}) \tag{11}$$

$$\hat{a} = \left(\frac{e^{*_{e,w}} - e^{*_{e,sat}}}{T_{e,w} - T_{e,sat}} \right) \tag{12}$$

$$Q_e \fallingdotseq \left(\frac{\hat{a}}{D_1 \dot{m}_r^{0.8}} + \frac{c_{p,air}}{D_2 \dot{m}_{RA}^{0.6}} + D_3 \right)^{-1} \times \left\{ \frac{(e_{a,e,in} - e^{*_{e,sat}}) - (e_{a,e,out} - e^{*_{e,sat}})}{\ln\left(\frac{e_{a,e,in} - e^{*_{e,sat}}}{e_{a,e,out} - e^{*_{e,sat}}} \right)} \right\} \tag{13}$$

Average humidity of the air at the exit of evaporator can be calculated by Eq. (14), which is from the equation of mass transfer between tube surface and air. By using heat and mass transfer analogy, Eq. (13) including D_2 can be derived from Eq. (14). But, this resulted in incorrect prediction of the exit states of the air, which seems to be caused by the on-site characteristics such as droplet carry over.[13,14] Thus with empirical constant E, Eq. (14) can be arranged as Eq. (15), where $\overline{w}_w$ is saturated moist air humidity calculated at the averaged temperature of the evaporator tube wall.

$$\overline{w}_{out} = \overline{w}_w + (w_{in} - \overline{w}_w) \times \exp\left[\frac{-h_m A_o \eta_s}{\dot{m}_{RA}} \right] \tag{14}$$

$$\overline{w}_{out} \fallingdotseq \overline{w}_w + (w_{in} - \overline{w}_w) \times \exp[-E \cdot \dot{m}_{RA}^{-0.4}] \tag{15}$$

3. On-site experiment

On-site experiments are required to determine experimental constants included in the modelling mentioned above. In the experiments, flow rates, temperatures and humidities of air, condenser and evaporator pressures, degree of superheating and subcooling, compressor power, and discharge gas temperature of refrigerant should be measured respectively. The compressor power can be measured by portable type digital power meter. Since thermal resistance of tube wall is assumed negligible, type T thermocouples that were tightly attached to the tube wall were used for measuring refrigerant temperatures. The medium-sized heat pump generally has its own pressure gauges and offers service ports by which refrigerant pressures can be measured. If neither pressure gauges nor service ports are available, refrigerant pressures are to be estimated from tube wall temperatures measured at proper positions, where condensation or evaporation seems to occur. For air temperature and humidity measurements, sampling equipments should be used to yield averaged values for the cross sectional area. For air flow rate measurements, multinozzles are often used in standardized test. But this becomes quite difficult for on-site measurements. Hot-wire type anemometers or pitot tubes can be substitutes. If air flow in duct is fully developed, air flow rate can be determined by the center velocity measurement only with pre-determined relation between center velocity and air flow rate as Hartnett et al.[15] suggested. But, multiple measurement points for air velocity are usually required because air flow is not always fully developed. For a rectangular duct, 16 to 64 measurement points are recommended in T.A.B. Standard for HVAC Applications by SAREK.[16]

The system investigated in this study is an air source heat pump as shown in Fig. 1. The system has hermetic reciprocating compressor

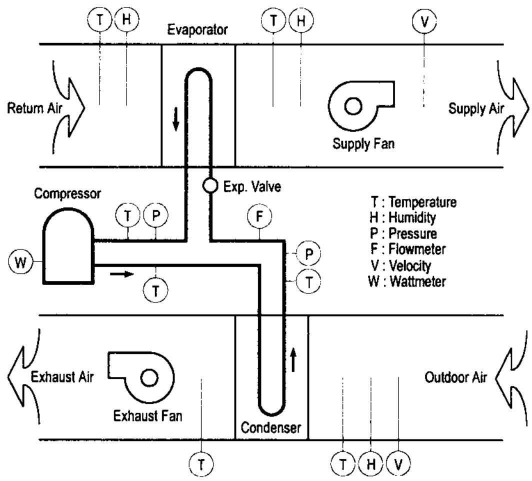

Fig. 1 Schematic diagram of an experimental apparatus.

made by Maneurop and uses HCFC-22 as the working fluid. The evaporator and condenser are finned-tube type cross flow heat exchanger with corrugated aluminum fins. Thermostatic expansion valves, accumulators, filter dryers, suction strainers and sight glasses are also parts of the refrigerant line. The power, temperatures, humidities, pressures, and flow rates were measured for the determination of the experimental constants. Temperatures were measured by type T thermocouples with ± 0.1℃ accuracy. Humidity transmitter with accuracy of $\pm 2\%$ was also used. Condenser and evaporator outlet pressures were measured by strain-gauge-type pressure transducers with accuracy of $\pm 0.1\%$. Pitot tube and micro-manometer with accuracy of $\pm 0.1\%$ were used for air velocity measurement. All instruments were scanned by data acquisition system and the data were transferred to a computer through GPIB and NI Labview program. Since the heat pump was installed in the field, experimental conditions could not be controlled and experiments were done with the inlet air conditions as they existed. Experiments conditions are shown in Table 1. Uncertainty analysis is carried out by the method of Kline and McClintock.[17] Assuming

Table 1 Experimental conditions

T_{OA} [℃]	21.8~31.8
T_{RA} [℃]	21.3~23.2
RH_{RA} [%]	54.3~72.0
P_e [kPa]	462~503
P_c [kPa]	1324~1951
Q_e [kw]	19.6~25.6
W [kW]	7.25~8.38
SHF [%]	65~84
dT_{sub} [℃]	7.8~11.0
dT_{sup} [℃]	7.3~9.6

the uncertainty of flow rate, temperature, power measurement, and fluid properties are 3%, 0.3℃, 3%, and 3% respectively, average and maximum uncertainty for cooling capacity are 11.5 % and 12.3% and those for COP are 11.9% and 12.6%.

4. Results and discussion

Constants V, ε', η_s, B, C_1, C_2, C_3, D_1, D_2, D_3 and E should be determined from the experimental results. If sufficient experimental results are available, least square method which yields minimum error for each experiment, is recommended essentially. But, as on-site experiment is time consuming and difficult, fewer experiment is preferred if possible. To fulfill this requirement, only a few experimental results were used to determine the constants in this study. Table 2 shows two set of experimental data of which the first one is presented in P-h diagram of Fig. 2. Table 3 shows experimental constants determined by measurements. Fouling resistance and contact resistance are neglected with an assumption that they are relatively smaller than other thermal resistances. Namely, C_3 and D_3 are regarded as 0. η_s, B, and E are averaged values from each experimental data and the other constants are acquired by solution of simultaneous equation using both data set. η_m has a value of 0.95 on the average. Once the constants are determined, system performance for other conditions can be predicted by a system simulation program developed in this study. Its input variables are

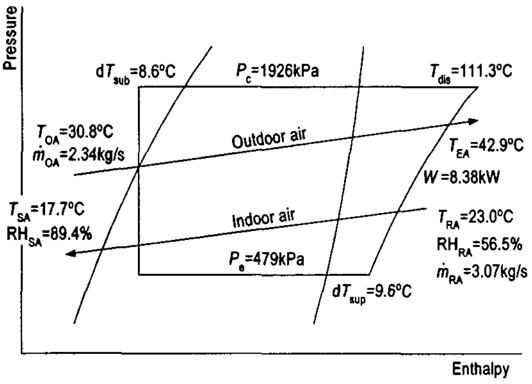

Fig. 2 Experimental data #1 in P-h diagram.

Table 2 Experimental data

	Data #1	Data #2
2000. 6. 21.	14 : 33	06 : 16
W [kW]	8.38	8.08
T_{dis} [℃]	111.3	99.5
P_c, P_e [kPa]	1926, 479	1721, 499
dT_{sup}, dT_{sub} [℃]	9.6, 8.6	8.4, 10.8
$\dot{m}_{OA}$, $\dot{m}_{RA}$ [kg/s]	2.34, 3.07	2.42, 3.06
T_{OA}, T_{EA} [℃]	30.8, 42.9	25.3, 38.3
T_{RA} [℃], RH_{RA} [%]	23.0, 56.5	22.5, 72.0
T_{SA} [℃], RH_{SA} [%]	17.7, 89.4	18.3, 71.1

Table 3 Experimental constants

Par.	Value
V	0.01
η_s	0.562
ε'	0.175
B	0.00284
C_1	39.93
C_2	2.38
D_1	15.9
D_2	0.45
E	0.26
C_3, D_3	0, 0

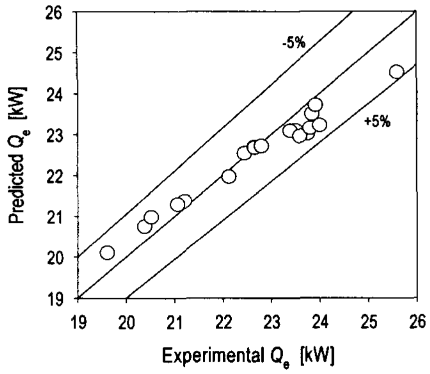

Fig. 3 Comparison of experimental Q_e with predicted Q_e.

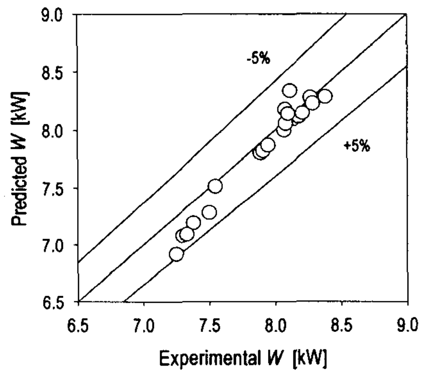

Fig. 4 Comparison of experimental W with predicted W.

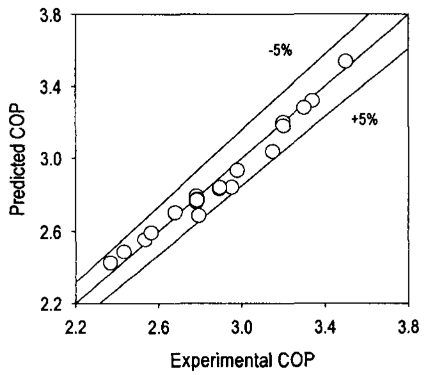

Fig. 5 Comparison of experimental COP with predicted COP.

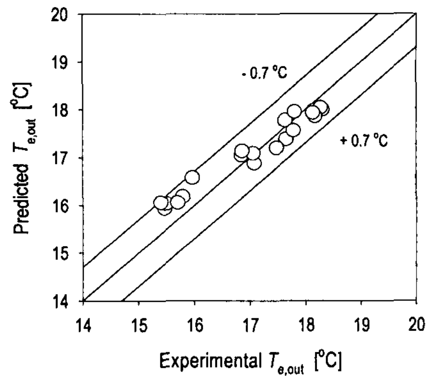

Fig. 6 Comparison of experimental $T_{e,out}$ with predicted $T_{e,out}$.

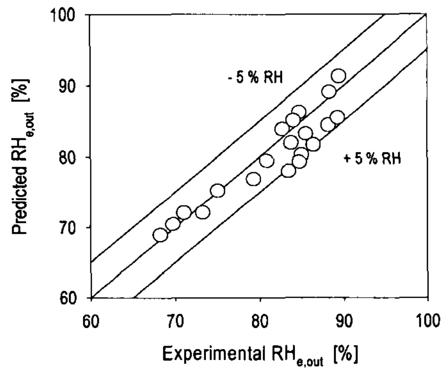

Fig. 7 Comparison of experimental $RH_{e,out}$ with predicted $RH_{e,out}$.

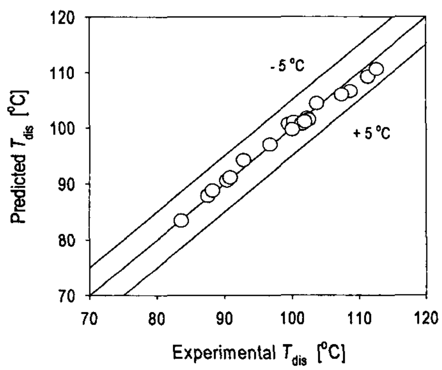

Fig. 8 Comparison of experimental T_{dis} with predicted T_{dis}.

inlet air flow rates, temperatures and humidities, and superheating and subcooling. An average of experimental superheating and subcooling measurement is used in this study. The steps of the simulation are as follows. First, assume condenser and evaporator pressures.

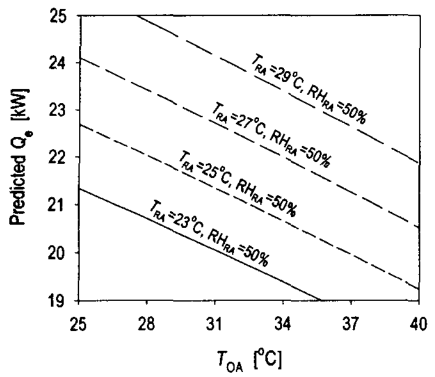

Fig. 9 Variation of predicted Q_e with respect to T_{OA} and T_{RA}.

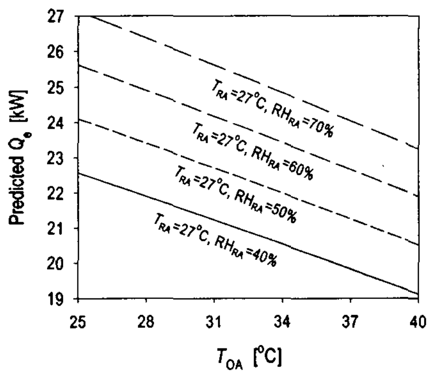

Fig. 10 Variation of predicted Q_e with respect to T_{OA} and RH_{RA}.

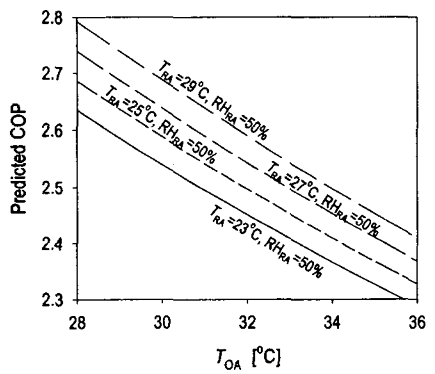

Fig. 11 Variation of predicted COP with respect to T_{OA} and T_{RA}.

Calculate refrigerant mass flow rate by Eqs. (1)~(7) and states of refrigerant and air at the condenser exit by Eq. (10). Compare subcooling between measured and calculated. If they are not within error tolerance, guess a better condenser pressure and calculate again. Else, calculate evaporator exit refrigerant state by Eq. (13) and compare it with experimental superheating. If it is satisfactory, calculate evaporator exit air state by Eq. (15) or else reassume evaporator pressure and repeat the above process. Simulation results include cooling capacity, compressor power, supply and exhaust air temperatures and humidities, condensation and evaporation pressure, discharged gas temperature, and the amount of condensate. Figures 3~8 show calculated cooling capacity, power, COP, states of supply air, and discharged gas temperature with 20 experimental results. The simulation results are in good agreement with those of experiment within acceptable errors. The system performance variations for other conditions are predicted by simulation program. Figure 9 and 10 show predicted cooling capacity variations with respect to temperature and humidity variations of the air. The predicted COP variations with respect to temperature variations of the air are also shown in Fig. 11. The simulation results show physically reasonable trends. Cooling capacity and COP decrease as outdoor temperature increases for the same indoor condition. Indoor humidity is also a major factor that affects the system performance. According to Figs. 9~11, the system investigated in this study shows the performance of cooling capacity of 21.7 kW and COP of 2.41 for standard cooling test condition[1] as given in Table 4.

Table 4 Standard rating conditions for cooling

Indoor unit air entering	DB T [℃]	27±1
	WB T [℃]	19.5±0.5 (≒50%RH)
Outdoor unit air entering	DB T [℃]	35±1
	WB T [℃]	24±0.5 (≒40%RH)

5. Conclusions

In this study, on-site test method and performance prediction modelling technique of medium-sized air source heat pumps installed in the field were presented. The results are verified with experimental results from on-site performance tests. This study provides a method of predicting the performance of an air source heat pump in which on-site performance test is necessary. Verification of the method needs to be done with a wide range of experiments for various sites in the future.

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