A Study on an Eco-friendly and High-performance Cooling System using Evapo-transpiration

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Abstract

According to International Energy Agency, buildings contribute nearly about 40 % total world energy consumption. Most of the consumption is used during their operational phase, especially for air conditioning, which causes peak-load in electricity demand in summer. Urban heat island recently has become a big issue for human's lives. Tokyo temperature has increased about 3.5 ^{o}C in the past one hundred years, while global temperature increases about 0.7 ^{o}C.

Recognizing that heat released from outdoor unit of the air conditioning is one of the causes of heat island in the city, the objective of the study is to create a cooling system that does not exhaust heat to the environment. From the thermodynamic viewpoint, temperature is free from energy conservation. As the leaves of trees, they can keep the temperature or even make the air cooler while absorbing solar insolation, by evapo-transpiration. Learning from that phenomenon, an air-conditioning system that not only cools the indoor space but also can create comfortable space outdoor by using evapo-transpiration is proposed in this study. With the fact that latent heat of water vaporization gives high potential of cooling capacity, e.g., 1 g water evaporation in 1 second can absorb about 2.43 kW heat at a temperature of 30 ^{o}C, it is expected that exhaust air from outdoor unit can reach to wet-bulb temperature, which is lower than that of ambient or dry-bulb temperature by 7 ^{o}C at dry-bulb temperature of 30 ^{o}C and the relative humidity of 50 %.

Evapo-transpiration is applied to the condenser in the outdoor unit of the air conditioning system. Proposed condenser is copper-tubing covered by porous ceramics with tiny open holes, which can automatically spread water

by the capillary phenomenon. The heat transfer coefficient is higher than that of the conventional air-cooled condenser from 3.5 up to 10 times. In addition, it also helps to reduce the condensing temperature; in consequence, reduce the work for compressor and energy consumption for air conditioning.

Experiments of an existing air conditioning system using air-cooled condenser, an air conditioning system with water-cooled condenser, and an air conditioning system with the proposed evapo-transpiration condenser have been done. Existing system is used as a baseline; while air conditioning system with water-cooled condenser is used to test the ability to reduce condenser temperature; and finally, the proposed system is to confirm the possibility of apply evapo-transpiration principle to air-conditioning system.

The result confirms that condenser temperature is reduced to ambient air from the water-cooled condenser. From simulation result, it is expected that any system, which has condenser temperature near to outdoor temperature, can increase its Coefficient of Performance (COP) up to 30 %.

For the prototype air conditioning system with evapo-transpiration condenser, it shows that the air outlet from the outdoor unit has almost the same temperature as that of the outdoor air. Since there is no spray of water in the system and the air just passes the wet surface of ceramics, the relative humidity of the outlet air is just slightly higher than that of the ambient, at an average of 5%. Therefore, the proposed outdoor unit does not release any heat to the environment. Hence, the problem of heat island can be reduced.

In addition, for the energy consumption at the specific case of experiment, the proposed system was confirmed to reduce the energy consumption up to 30 %. Last but not least, water condensed indoor can be utilized to cool the condenser outdoor.

In conclusion, the study has figured out the originality to create an air-conditioning system which can cool a space without release higher heat to the environment. The proposed system can help to reduce urban heat island problem by using the evapo-transpiration phenomenon. Moreover, it is also confirmed the possibility to save energy consumption using the proposed system compared to conventional air conditioning system.

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List of Symbols

Symbols	Description	Unit
A	Area	m^2
c_p	Specific heat capacity	$J \cdot kg^{-1} \cdot K^{-1}$
$\dot{E}$	Exergy	W
H	Enthalpy	$J \cdot kg^{-1}$
h_{fg}	Latent heat of evaporation	$J \cdot kg^{-1}$
h_f	Enthalpy of saturated water at Temperature of air-vapor mixture	$J \cdot kg^{-1}$
h_g	Enthalpy of saturated steam at Temperature of air-vapor mixture	$J \cdot kg^{-1}$
$\dot{m}$	Mass flow rate	$kg \cdot s^{-1}$
RH	Relative humidity of air	-
P	Pressure	kPa
P	Power	W
p_s	Water-vapor pressure	kPa
p_t	Atmospheric pressure	kPa
p_w	Partial pressure of water	kPa
q, Q	Heat rate	W
t, T	Temperature	K, ^{o}C
U	Overall heat transfer coefficient	$W \cdot m^{-2} \cdot K^{-1}$
V	Velocity	$m \cdot s^{-1}$
w, W	Net work	W
H	Correction factor	-
P	Density	$kg \cdot m^{-3}$
Ω	Humidity ratio	(kg water/kg air)

Subscripts

Symbols	Description
a	Air
cond	Condensing
e	Evaporation
f	Fan
h	Hot
I	Inlet
Lm	Logarithmic mean
n	Nominal condition
o	Outlet, outside
ref	Refrigerant
sys	System
sat	Saturated
theo	Theory
w	Water

Chapter 1 Introduction

I. Research Background

Buildings' energy consumption status

Currently, most of the energy consumption of buildings is used during their operational phase, rather than for construction. The International Energy Agency (IEA) estimates that residential, commercial and public buildings account for 30 % to 40 % of the world's energy consumption. The sector's contribution to the world CO_2 emissions is estimated at 25 % to 35 % (Schwarz 2009). Buildings consumption in European countries of total final energy consumption in 2010 is approximately 41 % (Bosseboeuf 2012), which is the same percentage as in the United States in the same year.

In developing countries, energy consumption and greenhouse gas (GHG) emissions related to buildings increase quickly as countries develop because of more urbanization and higher living standard. In the case of Vietnam, globally, about 50 % of the produced electricity is used in the household and service sector (MONRE 2005) with the annual electricity increases about 15 % (Eurocham 2010). Also, residential and commercial/institutional sectors compose of 13 % Vietnam energy GHG emission in the year of 2000.

Climate change issue

1. The Earth's temperature

The Earth's surface temperature has been successively warmer in the last three decades as in the report from Intergovernmental Panel on Climate

Change 2013 (IPCC 2013). It has increased about 0.6 °C since the year of 1900. Globally, the 12 hottest years on record have all come in the last 15 years (UNFCCC 2014).

In a World Bank report, even with the current mitigation commitments and pledges fully implemented, there is roughly a 20 % likelihood of exceeding 4 °C by the year 2100 (PIK 2012). Russia, tropical South America, central Africa, and all tropical islands in the Pacific are likely to regularly experience heat waves of unprecedented magnitude and duration. In Mediterranean, North Africa, the Middle East, and the Tibetan plateau, most of summer months seem to be warmer as well. Temperature in summer has also increased in more than 100 years in Japan, according to the data recorded of Japan Meteorology Agency (JMA), see Fig. 1 for Tokyo temperature in August, which is the hottest month in a year. In addition, number of days that air temperature is over 35 °C has increased in recent years.

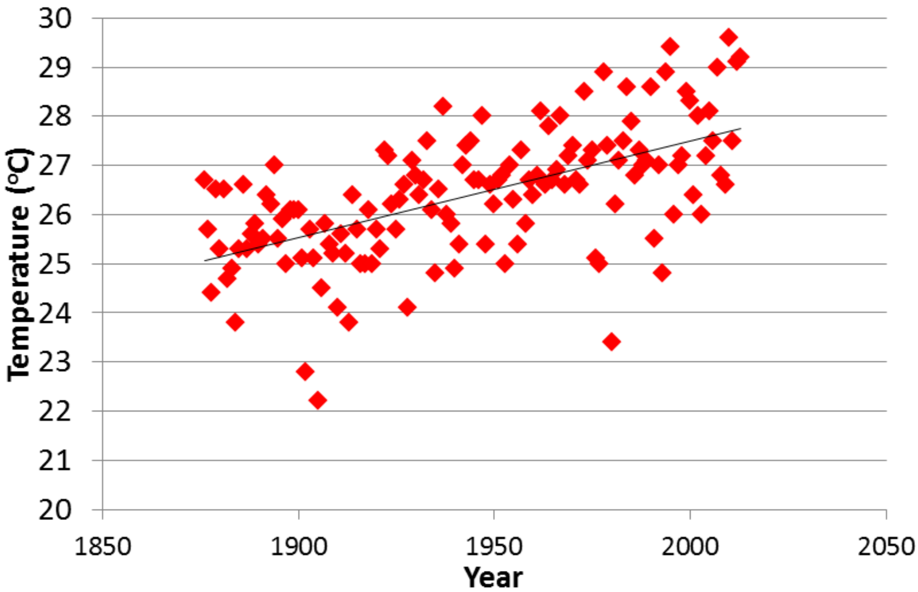


Fig. 1 Average temperature of Tokyo in August from 1876 to 2013

[Source: JMA]

Average U.S. temperature has increased by about 0.8 °C since 1895; more than 80 % of this increase has occurred since 1980. The warmest year ever recorded in the United States was 2012. In that year, about one-third of all Americans experienced 10 days with temperature higher than 37.8 ^{o}C.

In Vietnam, based on the trend of data recording at all the stations in each sub-section of the country, it is estimated that temperature has increased annually from 0.6 ^{o}C to 0.9 ^{o}C (Thang 2010).

2. Urban Heat Island (UHI)

The term "heat island" describes the area that is hotter than nearby areas. Urban heat island, which means that the city is hotter than its surrounding rural areas, can affect communities by increasing peak energy demand, air conditioning costs, heat-related illness and mortality, and water quality.

According to The United States Environmental Protection Agency (USEPA 2008), the annual mean air temperature of a city with 1 million people or more can be 1-3 °C warmer than its nearby rural areas. In Vietnam, the heat island effect reveals that temperature of the inner-core of Ho Chi Minh City is up to 10 ºC higher than typical temperatures of the surrounding rural areas (Storch 2008).

In Tokyo, Japan, annual temperature has increased about 3.5 ^{o}C since the year of 1876 to the year of 2013, as in Fig. 2, which is much higher than the elevation of global temperature.

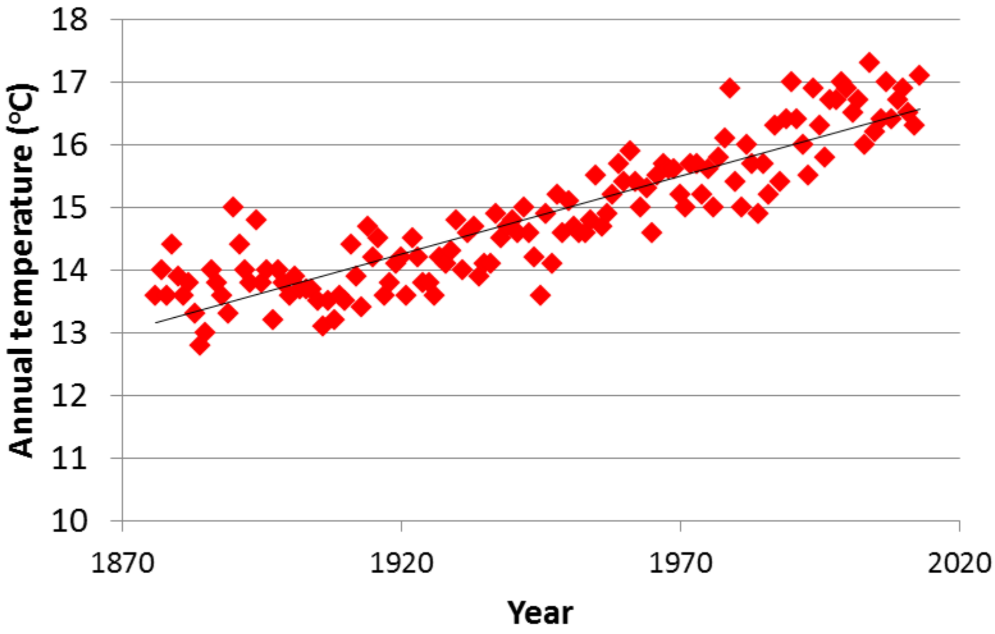


Fig. 2 Annual temperature of Tokyo from 1876 to 2013 [Source: JMA]

Heat island can be created in the city because of following factors:

a. ***Reduced vegetation***: In rural areas, vegetation and open land typically dominate the landscape. Trees and vegetation provide shade, which helps lower surface temperatures. It also helps reducing air temperatures through a process called *evapo-transpiration*, in which plants release water to the surrounding air, dissipating ambient heat. In contrast, in the city, more vegetation is lost, and more surfaces are paved or covered with buildings as the city develops. Therefore, it evaporates less water (Fig. 3); in consequence, surface and air temperatures in urban area increases.

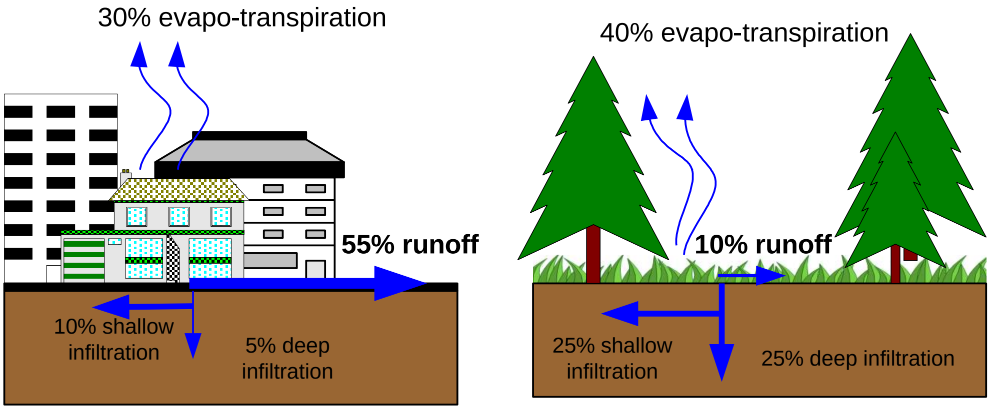


Fig. 3 Impervious Surfaces and Reduced Evapo-transpiration [redraw from The United States Environmental Protection Agency (USEPA 2008)]

b. ***Absorption of solar energy on the surface of the material used*** in the construction, such as conventional roofs, sidewalks, roads, parking lots…

c. ***Heat emissions*** accumulated by the citizen's activities:

Rapid and dense urban expansion has affected the increasing of energy demands for indoor cooling. Recently, split small-scale air-conditioner with air-cooled outdoor-unit is used very popular in residential and commercial application, especially in developing countries, like in Fig. 4. In Vietnam, the energy consumption for cooling of buildings due to the effect of UHI increases more than 50% compared to that in rural areas (Storch 2008).

Fig. 4 Air conditioners outside office building and apartment in Ho Chi Minh City, Vietnam

d. ***Urban infrastructure geometry*** affects the amount of radiation received and emitted

What is Evapo-transpiration phenomenon?

In nature, evapo-transpiration (or evapotranspiration) is the combination of two processes whereby water is lost from the soil surface by evaporation and from the leaf of the tree by transpiration, as in Fig. 5. However, there is not easy to distinguish between the two processes. In both processes, energy is required to change the state of water from liquid to vapour. Normally, direct solar radiation and the ambient temperature of the air provide this energy. Hence, evapo-transpiration gives cooling effect to the ambient air, i.e. 1 g water evaporation in 1 second can absorb about 2.43 kW heat at a temperature of 30 ^{o}C.

Besides, amount of evaporation is also affected by rains, irrigation, water transported upwards in soil, wind velocity, degree of shading… Likewise, transpiration, which occurs at the small openings on the plant leaf, also depends on energy supply, vapour pressure gradient and wind. Therefore, both evaporation and transpiration are affected by radiation, air temperature, air humidity and wind.

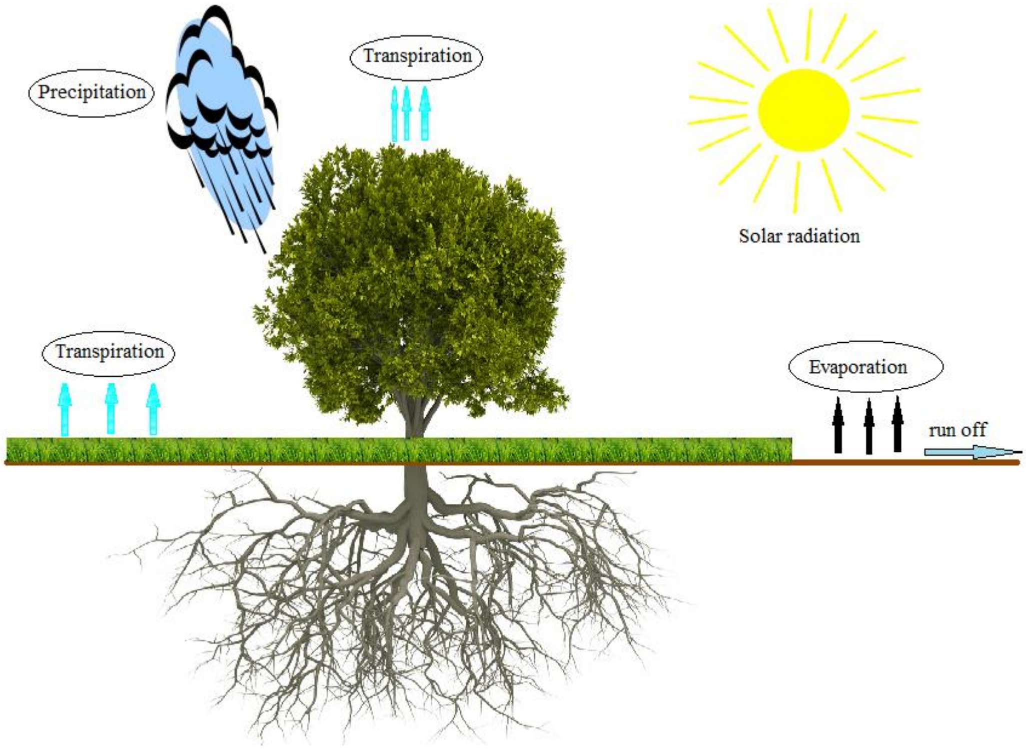


Fig. 5 Evapo-transpiration in nature

Remarks

As mentioned above, global warming is the most urgent problem to be solved by human beings, and eliminating CO_2 emission is the most important measure as well. The earth receives thermal energy from the sun and thermal radiation from itself, in which thermal radiation is several times greater than the amount of solar insolation all over the surface of the globe. Therefore, heating the surface air of the globe is also the key issue that could make the earth warmer and warmer. For this reason, both eliminating CO_2 emission, accompanied with avoiding heat release to the air might be the most effective measure to the global warming and urban heat island problem.

From a thermodynamic viewpoint, temperature is free from the energy conservation. As mentioned above, evapo-transpiration absorbs solar energy and heat of ambient air to its process but still keeping the temperature. Applying this principle to the existing air-conditioning system (cooling mode), one of the most important objectives of the study is to create environmentally friendly cooling system from engineering side, which is designed to keep a nature of the globe. The proposed cooling system has:

(1) no heat released to the outside environment

(2) possibility to have higher Coefficient of Performance, COP, compared to the existing system due to lower temperature of condenser, that also means lower energy consumption for the same cooling capacity required;

II. Research Objectives

An air conditioning system of the small domestic one is selected to be studied, but it is possible to be applied into business scales, which is quite similar to cooling tower. Aware of the problems of global warming and heat island in cities due to lack of evapo-transpiration, research objective is to apply evapo-transpiration phenomenon to the outdoor unit of the air-conditioning system to minimize its waste-heat release to the surroundings and increases the system's performance. In addition, it can contribute to a better comfort space surrounds the outdoor unit.

III. Research Scope and Methodology

Scope of the study is to develop a new condenser for the small-scale air conditioning system in cooling mode by performing different experiments.

Besides, a program for evaluating those experiments is also developed to simulate the performances of the heat exchanger and the cooling system.

IV. Dissertation Organization

To demonstrate the overview of this study, this dissertation is divided into six chapters. Chapter I introduces the overview of this study including research background, objective, scope, methodology, and dissertation organization. Theoretical review of vapour compression cycle and review on condensers of vapour-compression cycle air-conditioning system is presented in chapter II. Chapter III is about experimental work including: a heat exchanger using porous ceramics, water-cooled air-conditioning system using cooling tower, and finally an air conditioner with a prototype of evapo-transpiration heat exchanger. Then, in chapter IV, all the calculation and algorithm to evaluate the heat exchanger and the system are presented. Chapter V shows the experimental results and discuss issues related. Summary as well as findings of the study will be demonstrated in the last chapter.

Chapter 2 Theory and Literature Review

I. Vapor compression cycle

The vapor-compression cycle is the most widely used refrigeration cycle in practice. In this cycle a vapor is compressed, then condensed to a liquid, following which the pressure is dropped so that fluid can evaporate at a low pressure.

I.1. Carnot refrigeration cycle

The Carnot cycle is one whose efficiency cannot be exceeded when operating between two given temperatures. The Carnot cycle operating as a heat engine is familiar from the study of thermodynamics. The Carnot refrigeration cycle performs the reverse effect of the heat engine, because it transfers energy from a low level of temperature to a high level of temperature. The refrigeration cycle requires the addition of external work for its operation. The diagram of the equipment and temperature-entropy diagram of the refrigeration cycle are shown in Fig. 6. All processes in the Carnot cycle are thermodynamics reversible. Processes 1-2 and 3-4 are consequently isentropic.

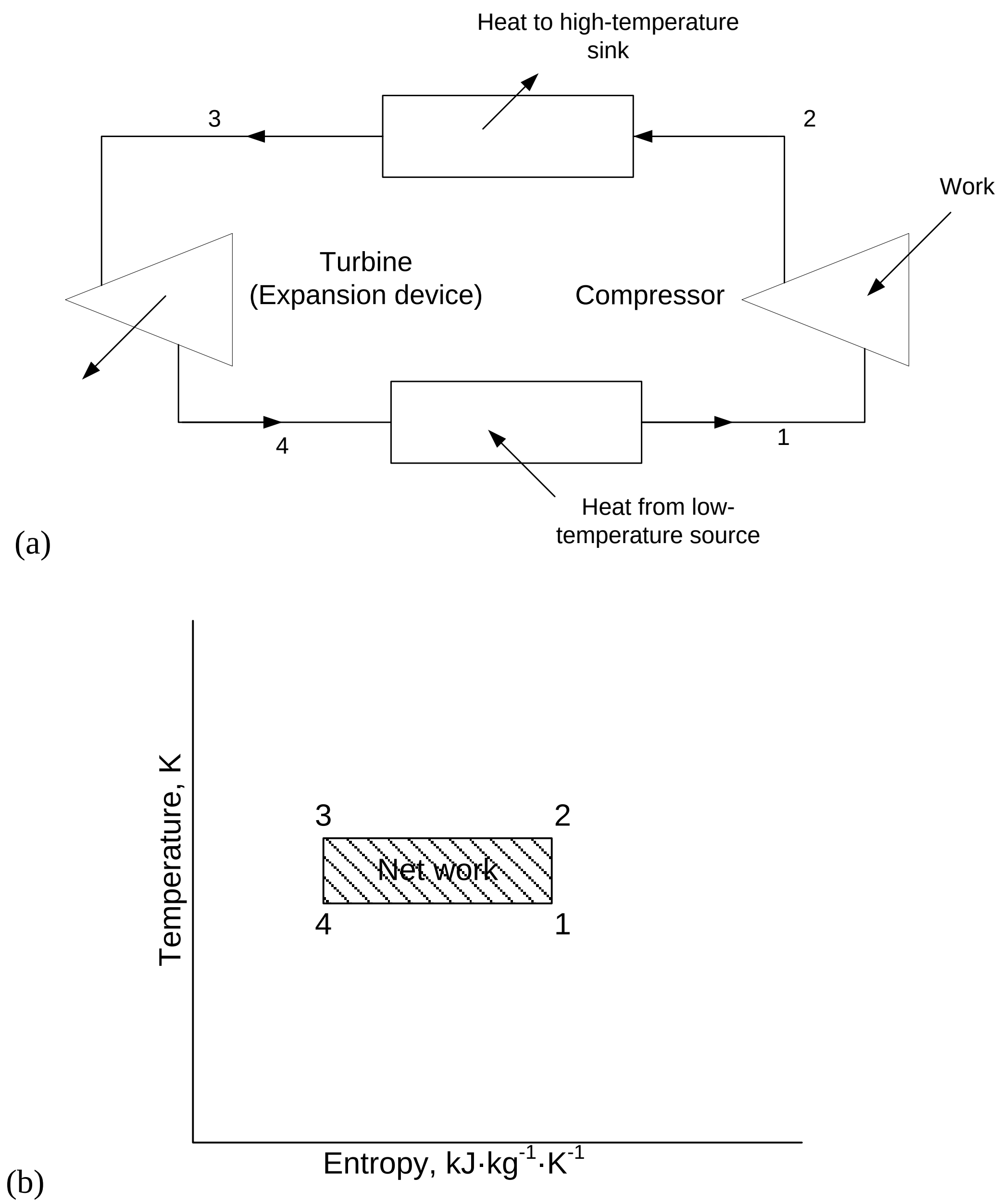


Fig. 6 (a) Carnot refrigeration cycle; (b) temperature-entropy diagram of the Carnot refrigeration cycle

The withdrawal of heat from the low-temperature source in process 4-1 is the refrigeration step and is the entire purpose of the cycle. All the other processes in the cycle function so that the low-temperature energy can be discharged to some convenient high-temperature heat sink.

I.2. Coefficient of Performance (COP)

Before any evaluation of the performance of a refrigeration system can be made, an effectiveness term must be defined. The index of performance is not called efficiency, however, because that term is usually reserved for the ratio of output to input. The ratio of output to input would be misleading applied to a refrigeration system because the output in process 2-3 is usually wasted. The concept of the performance index of the refrigeration cycle is the same as efficiency, however, in that it represents the ratio

$$\frac{\text{Magnitude of demand}}{\text{Magnitude of expenditure}}$$

The performance term in the refrigeration cycle is called the coefficient of performance, defined as

$$\text{Coefficient of performance} = \frac{\text{useful refrigeration}}{\text{net work}}$$

The two terms, which make up the coefficient of performance, must be in the same units, so that the coefficient of performance is dimensionless.

*** Condition for highest coefficient of performance**

We can express the coefficient of performance of the Carnot cycle in terms of the temperatures that exist in the cycle. Areas beneath reversible processes on the temperature-entropy diagram therefore represent transfers of heat. Areas shown in Fig. 7 can represent the amount of useful refrigeration and the net work. The useful refrigeration is the heat transferred in process 4-1, or the area beneath line 4-1. The area under line 2-3 represents the heat reject from the cycle. The difference between the heat rejected from the cycle and heat added to the cycle is the net heat which for a cyclic process equals the net work. The area enclosed in rectangle 1-2-3-4 represents the net work. An expression for the coefficient of performance of the Carnot refrigeration cycle is therefore:

$$\text{Coefficient of performance} = \frac{T_1(s_1-s_4)}{(T_2-T_1)(s_1-s_4)} = \frac{T_1}{T_2-T_1} \qquad (1)$$

where s is entropy and T is temperature.

The coefficient of performance of the Carnot cycle is entirely a function of the temperature limits and can vary from zero to infinity.

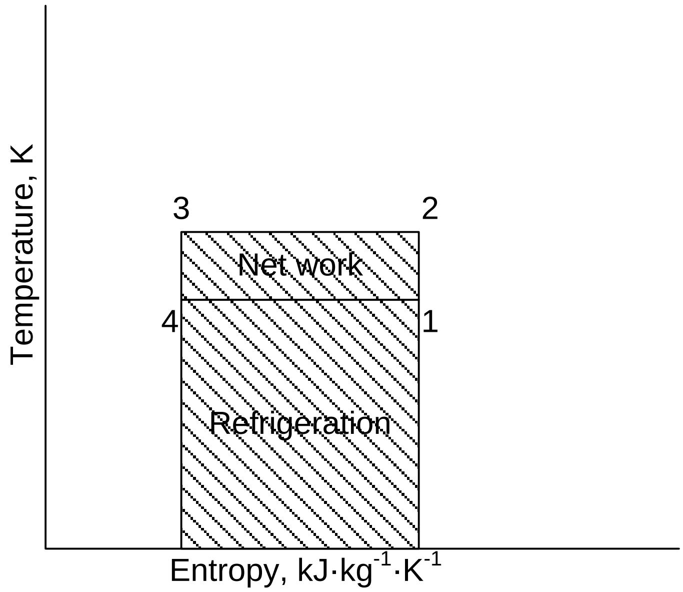


Fig. 7 Useful refrigeration and net work of the Carnot cycle shown by area on the temperature-entropy diagram.

A low value of T_2 will make the coefficient of performance high. A high value of T_1 increases the numerator and decreases the denominator, both of which increase the coefficient of performance.

To summarize, for a high coefficient of performance (1) operate with T_1 high and (2) operate with T_2 low. It means that the smaller temperature difference between condenser and evaporator, the higher COP achieves.

*** Temperature limitation**

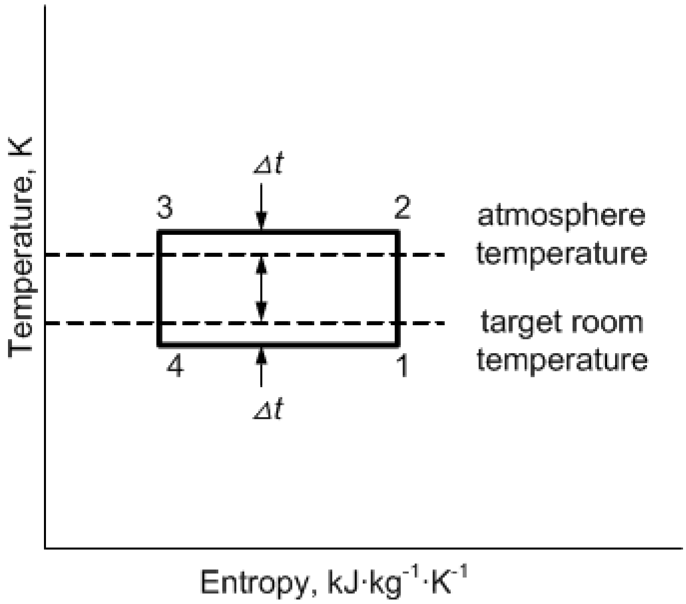


Fig. 8 Temperature requirements imposed upon a refrigeration cycle

In Fig. 8, temperature T_2 should be kept low, but it cannot be reduced below atmosphere temperature, if the atmosphere air is used to cool it. On the other hand, temperature T_1 should be kept high, but it can be increased no higher than the target temperature of the room. Because of that, it can concentrate on keeping the Δt as small as possible.

In heat transfer, reduction of Δt can be accomplished be increasing A or U in the equation:

$$q = UA\,\Delta t \tag{2}$$

where q = heat rate, W

U = overall heat-transfer coefficient, $W \cdot m^{-2} \cdot K^{-1}$

A = heat-transfer area, m^2

Δt = temperature change, K

In order to decrease Δt to zero, either U or A would have to be infinite. Since infinite values of U and A would require an infinite cost, the actual selection of equipment always stops short of reducing Δt to zero.

I.3. Standard vapour-compression cycle

The standard vapour-compression cycle is shown on the temperature-entropy diagram in Fig. 9. The processes constituting the standard vapour-compression cycle are:

1-2. Reversible and adiabatic compression from saturated vapor to the condenser pressure

2-3. Reversible rejection of heat at constant pressure, causing desuperheating and condensation of the refrigerant

3-4. Irreversible expansion at constant enthalpy from saturated liquid to the evaporator pressure

4-1. Reversible addition of heat at constant pressure causing evaporation to saturated vapor

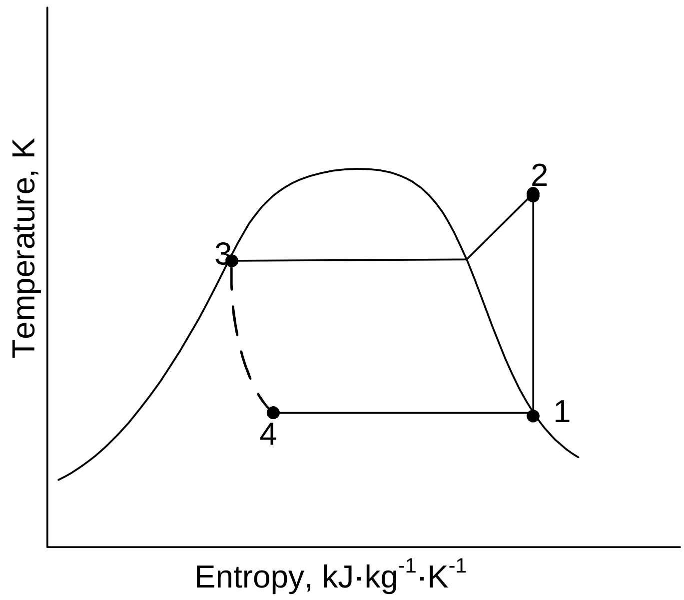
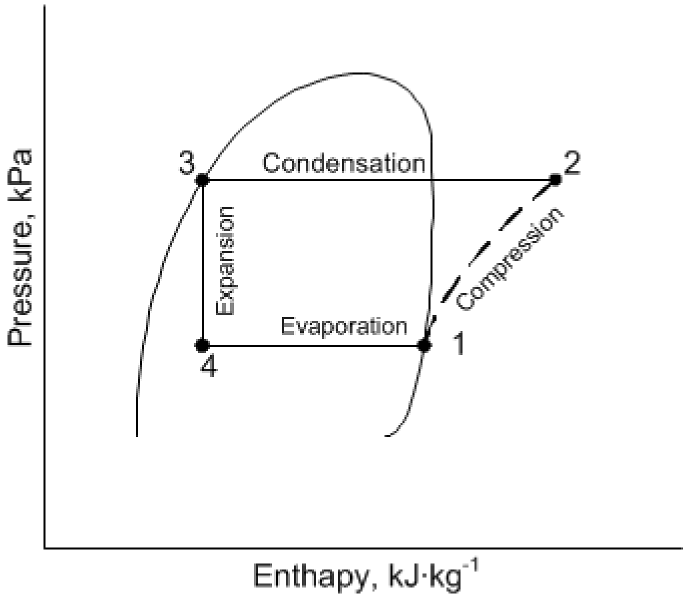


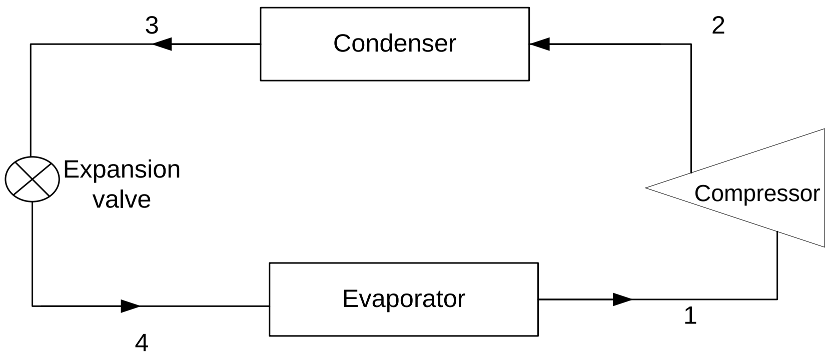
Fig. 9 The standard vapor-compression cycle

*** Performance of the standard vapor-compression cycle**

With the help of the pressure-enthalpy diagram, the significant quantities of the standard vapor-compression cycle will be determined. These quantities are the work of compression, the heat-rejection rate, the refrigerating effect, the coefficient of performance, the volume rate of flow per kilowatt of refrigeration, and the power per kilowatt of refrigeration.



(a)



(b)

Fig. 10 (a) The standard vapor-compression cycle on pressure-enthalpy diagram; (b) flow diagram.

The work of compression in kilojoules per kilogram is the change in enthalpy in process 1-2 of Fig. 10 (a) or $h_1 - h_2$. This relation derives from the steady-flow energy equation:

$$h_1 + q = h_2 + w \tag{3}$$

where changes in kinetic and potential energy are negligible. Because in the adiabatic compression the heat transfer q is zero, the work w equals $h_1 - h_2$. The difference in enthalpy is a negative quantity, indicating that work is done on the system. Even though the compressor may be of the reciprocating type, where flow is intermittent rather than steady, process 1-2 still represents the action of the compressor. At a short distance in the pipe away from the compressor, the flow has smoothed out and approaches steady flow.

The heat rejection in kilojoules per kilogram is the heat transferred from the refrigerant in process 2-3, which is $h_3 - h_2$. This knowledge also comes from the steady-flow energy equation, in which the kinetic energy, potential energy, and work terms drop out. The value of $h_3 - h_2$ is negative, indicating that heat is transferred from the refrigerant. The value of the heat rejection is used in sizing the condenser and calculating the required flow quantities of the condenser cooling fluid.

The refrigerating effect in kilojoules per kilogram is the heat transferred in process 4-1, or $h_1 - h_4$. Knowledge of the magnitude of the term is necessary because performing this process is the ultimate purpose of the entire system.

The coefficient of performance of the standard vapor-compression cycle is the refrigerating effect divided by the work of compression:

$$Coefficient\ of\ performance = \frac{h_1 - h_4}{h_2 - h_1} \tag{4}$$

*** Actual vapor-compression cycle**

The actual vapor-compression cycle suffers from inefficiencies compared with the standard cycle. There are also other changes from the standard cycle, which may be intentional or unavoidable. Some comparisons can be drawn be superimposing the actual cycle on the pressure-enthalpy diagram of the standard cycle, as in Fig. 11.

The essential differences between the actual and standard cycle appear in the pressure drops in condenser and evaporator, in the subcooling of the liquid leaving the condenser, and in the superheating of the vapor leaving the evaporator. The standard cycle assumes no drop in pressure in the condenser and evaporator. Because of friction, however, the pressure of the refrigerant drops in the actual cycle. The result of these drops in pressure is that the compression process between 1 and 2 requires more work than in the standard cycle. Subcooling of the liquid in the condenser is a normal occurrence and serves the desirable function of ensuring that 100 % liquid will enter the expansion device. Superheating of the vapor usually occurs in the evaporator and is recommended as a precaution against droplets of liquid being carried over into the compressor. The final difference in the actual cycle is that the compression is no longer isentropic and there are inefficiencies due to friction and other losses.

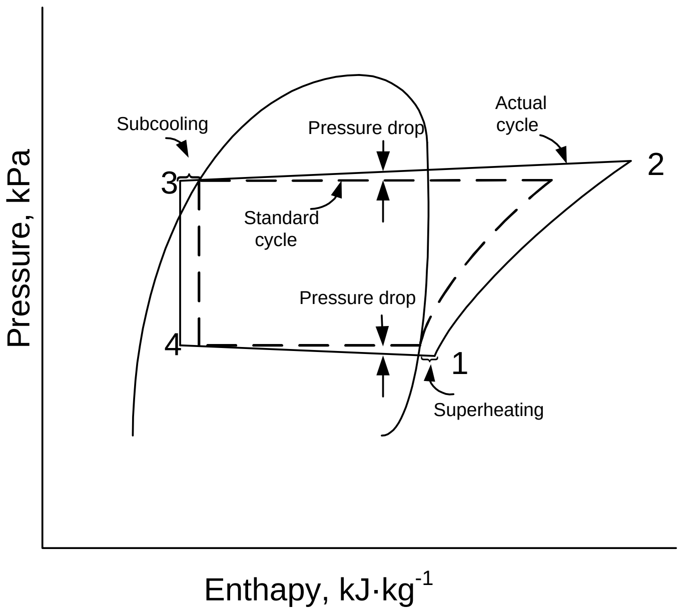


Fig. 11 Actual vapor-compression cycle compared with standard cycle

I.4. Commercial split air-conditioner

A commercial split small-scale air-conditioner is composed of an evaporator, an expansion device, a condenser and a compressor, as in Fig. 12.

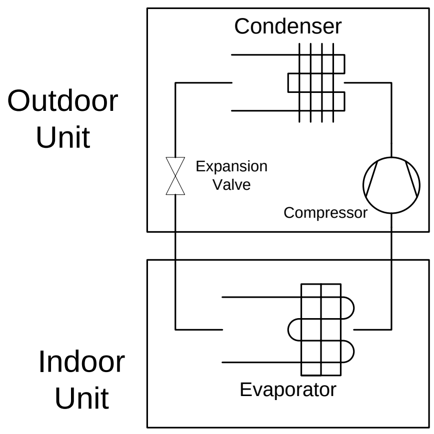


Fig. 12 Split air-conditioner

II. Review of outdoor heat exchanger of air-conditioning system

In cooling mode, outdoor heat exchanger functions as a condenser. For split domestic air conditioning system, as vapour-compression air conditioning system, depending on the type of cooling medium, condensers can be classified as air-cooled, water-cooled and evaporative condensers.

II.1. Air-cooled condenser

Air-cooled condensers have contributed large amount of domestic using due to its advantages of easy maintenance, and convenient size. Shibata (Shibata 2007) has mentioned that for heat exchangers in air conditioning system, thermal resistance on the airside is 100 times compared to refrigerant side. Therefore, many researchers have most efforts in order to increase performance of air-side. Different kinds of fin have been investigated by experiments as well as numerical analysis. Diameter of heat-exchanger tube has been smaller little by little. Some other researchers use other kinds of tube, such as oval (Leu 2001) or flat tube (Awad 2007). Although these developments give the higher performances than circular tube, they are not popular. Air conditioner manufacturers still prefer using circular tube due to its availability in the market.

However, as in Eq. (2), cooling by using sensible heat from air has low effective (low U), causes high ΔT; hence, high condensing temperature. In some cases, condensing temperature is said to be up to 15-20˚C above that of the ambient air (Hosoz 2004).

In addition, outlet air temperature from air-cooled outdoor-unit is higher than that of ambient air. As in a study of Sato (Naito 2008) with commercial split air conditioner, air temperature from outdoor unit is higher than that of the ambient about 2 to 10 ˚C. As mention in Chapter 1 about heat islands problem, air conditioning is one of the heat sources that make urban air temperature higher.

Last but not least, as in the studies of Chow et al. (Chow 2002) and Hajidavalloo (Hajidavalloo 2007), they mentioned that the coefficient of performance (COP) of an air conditioner decreases about 2–4 % by increasing each ˚C in condenser temperature. Therefore, it consumes high energy in summer weather. As higher condensing temperature requires higher pressure ratio across the compressor, power cost increases, compressor life decreases and Coefficient Of Performance, COP, also decreases. Moreover, for multi-storey buildings, hot air from lower outdoor-unit can reduce performance of the upper one. This problem has not taken care enough from manufacturing as well as research viewpoint.

II.2. Water-cooled condenser

In water-cooled condensers, the heat is absorbed by the water at the condenser. The cooling water is usually rejected to the ambient air by means of a cooling tower. However, commercial cooling tower power consumption is higher than common residential air-conditioning system. Therefore, water-cooled condenser system is applicable for medium and large scale systems.

Because of the above reason, a cooling tower has been simplified and combine inside an outdoor unit as in the study of Hu (Hu 2005). They developed a water-cooled technology for residential split-type air conditioner of 3.5 kW cooling capacity. Performance of the water-cooled condenser is elevated by improving air–water cooling design using cellulose pad. The steady state COP of this system can improve from 3.0 to

3.5. Extra power for cooling tower consumes minimum of 98 W, which is considerable for residential scale system. However, the size of this outdoor unit is rather big since it requires space of cooling pad for evaporation.

II.3. Evaporative-cooled condenser

Evaporative condenser is compact combining the functions of an air-cooled condenser, a water-cooled condenser and a cooling tower. In a typical evaporative-cooled condenser, hot refrigerant vapour is pumped by the compressor flows through a bank of tubes, whose outside is continually kept wet by a water-distributing system. In order to improve the performance, fin combined with packing material have been used (Ettouney 2001). Although the application of evaporative cooling in large industrial refrigeration systems were investigated, but there is little work to investigate the application of evaporative cooling on small size refrigeration system.

A new design with high commercialization potential for incorporating of evaporative cooling in the condenser of window-air-conditioner is introduced and experimentally investigated by Hajidavalloo (Hajidavalloo 2007). A real air conditioner is used to test the innovation by putting two cooling pads in both sides of the air conditioner and injecting water on them in order to cool down the air before it passing over the condenser. The experimental results show that thermodynamic characteristics of new system are considerably improved and power consumption decreases by about 16% and the coefficient of performance increases by about 55 % at outdoor dry-bulb temperature of 45-46 ˚C. However, for direct cooling, fouling of condenser surface can affect to the overall performance (Qureshi 2006).

Goswami et al. (Goswami 1993) employed an evaporative cooling on existing 2.5 ton air conditioning system by using media pad. They put four media pad around condenser and inject water from the top by a small water pump. They reported the electric energy saving of 20 % for the retrofitted system when ambient air temperature was 34 ˚C. The retrofit system has Return On Investment (ROI) of 2 years.

Hwang et al. (Hwang 2001) has applied the innovative design of evaporatively-cooled condenser for residential-size heat pump system. The condenser is immersed in a cooling water tank, which is cooled by the rotating disks driven by a motor. The experimental results after optimizing short-tube length, amount of refrigerant charged and rotating speed showed that Coefficient of Performance (COP) increases by 11 to 22 %.

There is an evaporative-cooling condenser in a split air-conditioning system that has been in commercial with the name Freus (Freus 2013). Depending on different climate zone, the system is proved to have reduction in total energy use from 0.2 % to 49 %. Since water is necessary to spray directly on the condenser coil, additional pump is required and condenser surface is required some additional routine simple maintenance checks.

Cooling pad

Evaporative pad materials have also already investigated to improve the performance of systems in cooling tower, evaporative cooler and also evaporative condenser. Cellulose pad seems to be popular for using as cooling pad (Kittas 2003; Hu 2005; Dagtekin 2009). Gunhan (Gunhan 2007) has evaluated the suitability of some local materials like: pumice stones, volcanic tuff and greenhouse shading net to compare with the material called CELdek, which has been used in commercial. Also, in the study of Al-Sulaiman (Al-Sulaiman 2002), local fibers have been used: date palm fibers (stem), jute and luffa, and then compare with a commercial wetted pad as a reference. However, in those studies, they have not considered about how these materials effective by reducing circulation pump power and those cooling pad materials normally require much space of the system.

Comments

- In general, the most advantage of water-cooled and evaporate condensers is that they have lower condensing temperature and give higher energy efficiency but requires more maintenance than air-cooled system (Yik 2001; Hosoz 2004; Yu 2006).
- From the above improvement for small-scale air conditioning system, either additional power, e.g., disk motor, pump, is required; or the size of the unit is not considered.

Chapter 3 Experimental Systems Development

The main content in this chapter describes experiments that have been done to support the study: experiment with air-conditioning system connected with cooling tower, experiment with proposed heat exchanger and experiment with a prototype air-conditioning system using evapo-transpiration condenser.

All experiments have been performed at roof top of the library of Yagami Campus, Keio University in Yokohama, Japan.

Fig. 13 Experiment site at Yagami Campus, Keio University in Yokohama, Japan

Fig. 14 Air conditioning room for experiment

I. Experiment of proposed heat exchanger

A prototype of a new heat-exchanger has been set-up and tested in the summer of the year 2009 in Yokohama, Japan. In 2009, mean air temperature and mean relative humidity of Yokohama is as in Fig. 15. From this figure, as annual climate of Japan, summer in the year 2009 seems to start from the middle of June and end in the middle of September with the hottest month in August. In addition, relative humidity is also highest in those months of summer compared to the rest of the year. For high air temperature and high humidity, in the summer, the weather is not comfortable for people in the summer and cooling requirement is necessary. Therefore, the cooling system of the experiment has been done in summer for actual weather condition.

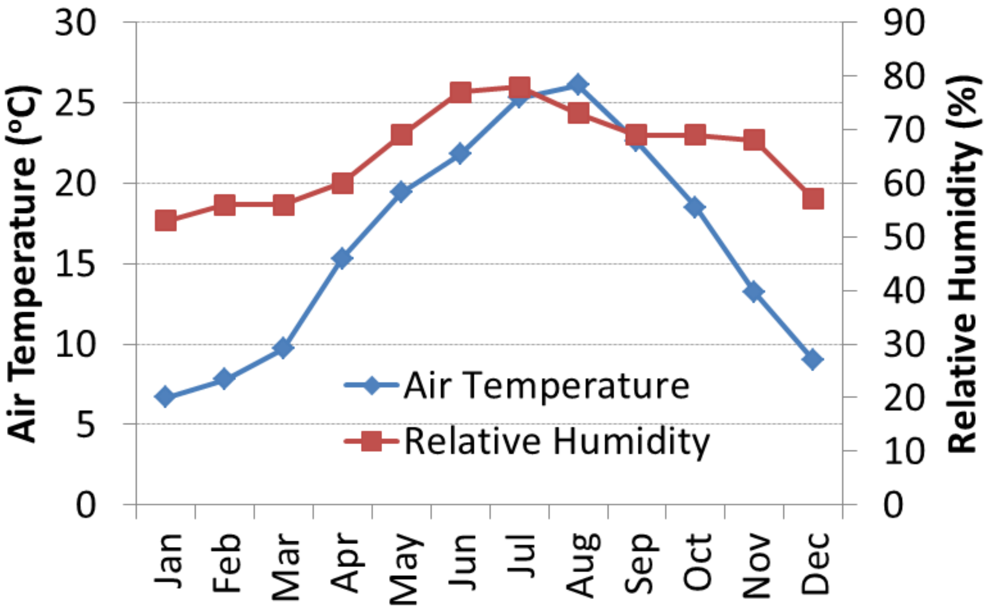


Fig. 15 Air temperature and relative humidity in Yokohama, 2009

[Source: JMA]

The experiment is sketched in Fig. 16. The heat-exchanger consists of copper tubes covered with porous ceramics. Porous ceramics used in this experiment has main component of Al_2O_3, that used in planting to keep the soil always wet. Moreover, diameter of vacancy inside porous ceramics is so small that porous ceramics can protect copper surface from outside condition.

However, it is manufactured with special shape for this experiment to cover copper tube surface. Since it is not flexible for experiment if ceramics is stick permanently on the round surface of copper tube; in addition, it is better that porous ceramics contact directly to copper surface without any intermediate layer. Therefore, a much easier way is that ceramic is manufactured in shape of half-round, then it is not difficult to attach on copper tube by rubber tie outside these two halves. Also, in this experiment, in order to select the best porosity for the heat exchanger, two types of porous ceramics are tested: 20 µm and 90 µm in porosity diameter. Besides, there are 2 possible ways to arrange these 2 half-round pieces on the copper tube, experiment for both of them have done to find the best arrangement, as in Fig. 17.

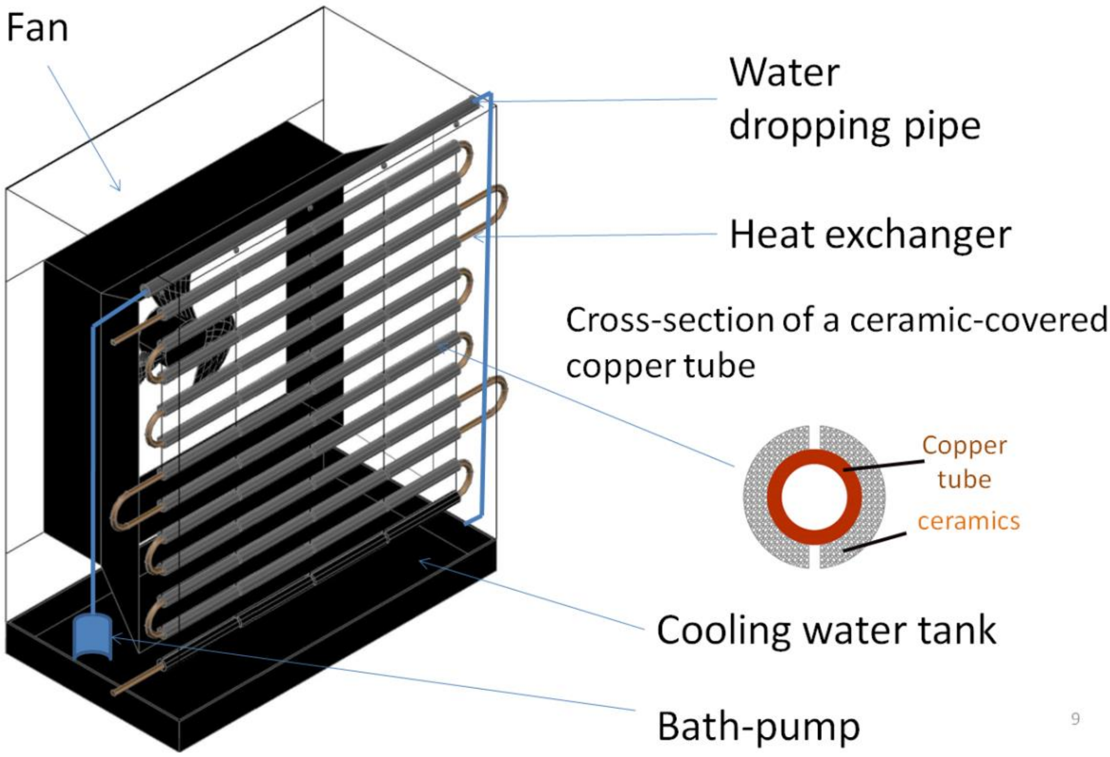


Fig. 16 Sketch of evaporative heat exchanger

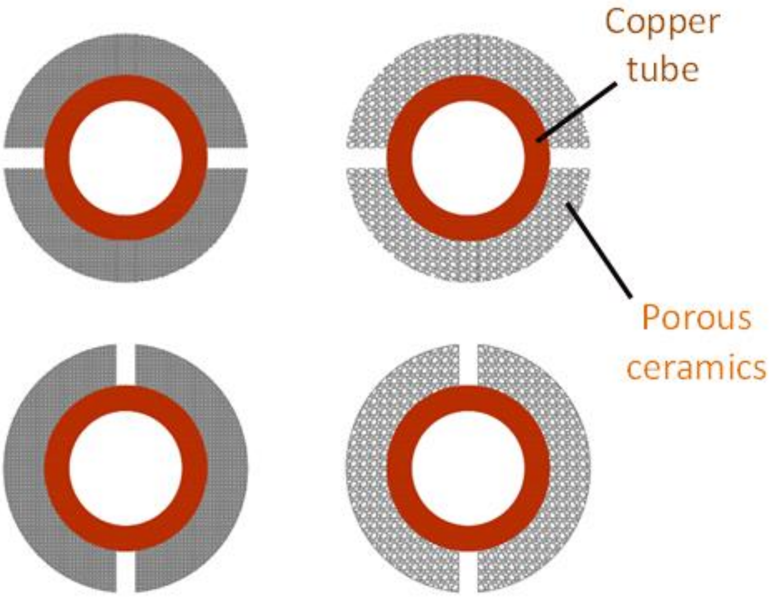


Fig. 17 Two types of porous ceramics and 2 types of arrangement on copper surface: horizontal (upper), vertical (lower); 20-µm-porosity ceramics (left-hand side) and 90-µm-porosity ceramics (right-hand side).

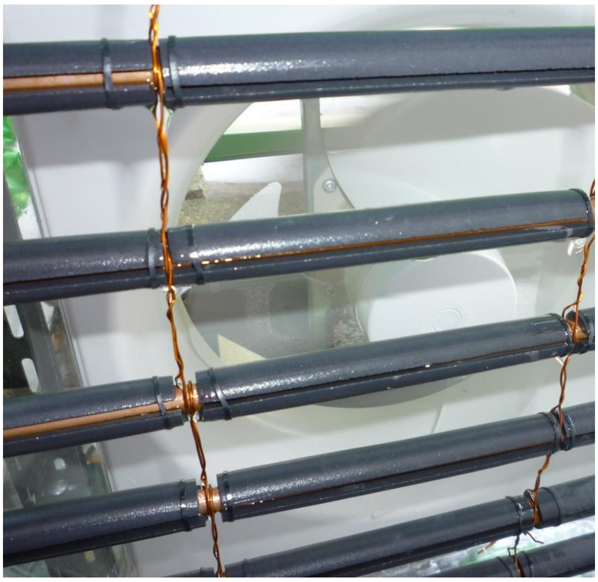


Fig. 18 Actual arrangement of porous ceramics with 20-µm-porosity ceramics in horizontal arrangement

Fig. 19 End of heat exchanger is insulated

- Since the distance of rows in heat exchanger is quite high, copper wire is also used to fix the distance between rows and also direct the flow of water drop from the top, shown in Fig. 18. In addition, connection at the end of heat exchanger is insulated by insulation material to reduce heat loss from hot water inside, Fig. 19.
- Hot water flows inside copper tube and is circulated by a circulation pump. Hot water is heated by an electric heater, as in Fig. 20.

Fig. 20 Hot water tank with electric heater

- Cooling water drops from the top to outside of heat exchanger, wetting ceramic surface. The water is circulated from bottom tank by a 13-W bath-pump.
- A fan of 1740 $m^3 \cdot h^{-1}$ flow-rate is put in front of the heat-exchanger, showed in Fig. 21. Air flow of the fan is used to enhance the evaporation of water on the ceramic surface. This airflow is ducted by an acrylic duct to maximize the flow velocity of the air. At the cross-section of heat exchanger, with dimension as in Fig. 22, the area is about 0.293 m^2; hence, average air flow velocity is about 1.63 m/s.

Fig. 21 The fan used in the experiment

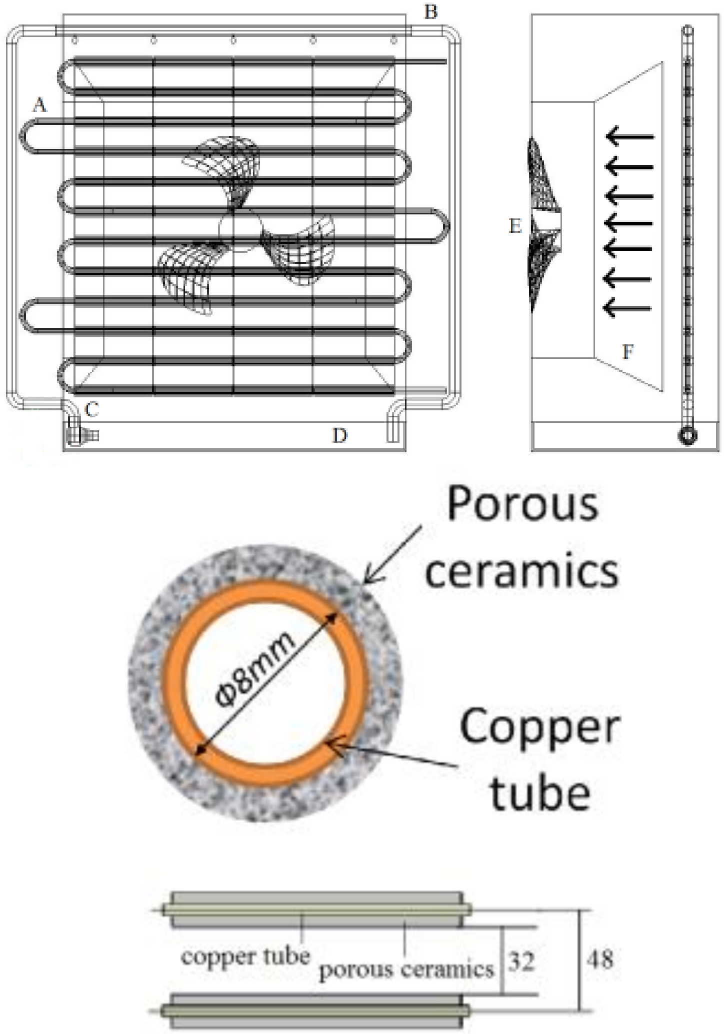


Fig. 22 Components of the heat-exchanger.

A: heat exchanger (copper tube and ceramics), B: water-drop pipe,

C: bath-pump, D: cooling-water tank, E: fan, F: duct

Besides, type-T-thermocouples are used to measure temperatures at different points in the system. Temperatures at inlet and outlet of hot water; top, middle and bottom of copper-tube and ceramics surfaces; inlet and outlet of air flow; of water in the bottom tank; and of air in the ambient are recorded. Data is connected to computer and recorded by data logger CADAC, with the uncertainty of 0.02 % of reading +0.03 ˚C.

II. Experiment of air conditioning systems

In the experiment with air conditioning system, 3 experiments with 3 air conditioning systems have been done for the same cooling space of 2.63 m × 1.79 m × 2.09 m:

- ***Baseline system***: is the conventional air conditioning system, which has been commercialized.
- ***"Water" system:*** is the air conditioning system whose condenser in the outdoor unit is water-cooled condenser connected to a cooling tower.
- ***Proposed system***: is the air conditioning system whose condenser is evapo-transpiration condenser.

All of the experiments have been performed in the year of 2010 and 2011, whose temperature and relative humidity are as in the figures of Fig. 23 and Fig. 24. Among the 3 years of 2009, 2010 and 2011, summer in the year of 2010 has highest air temperature, with highest mean air temperature of August of 28.6 ^{o}C.

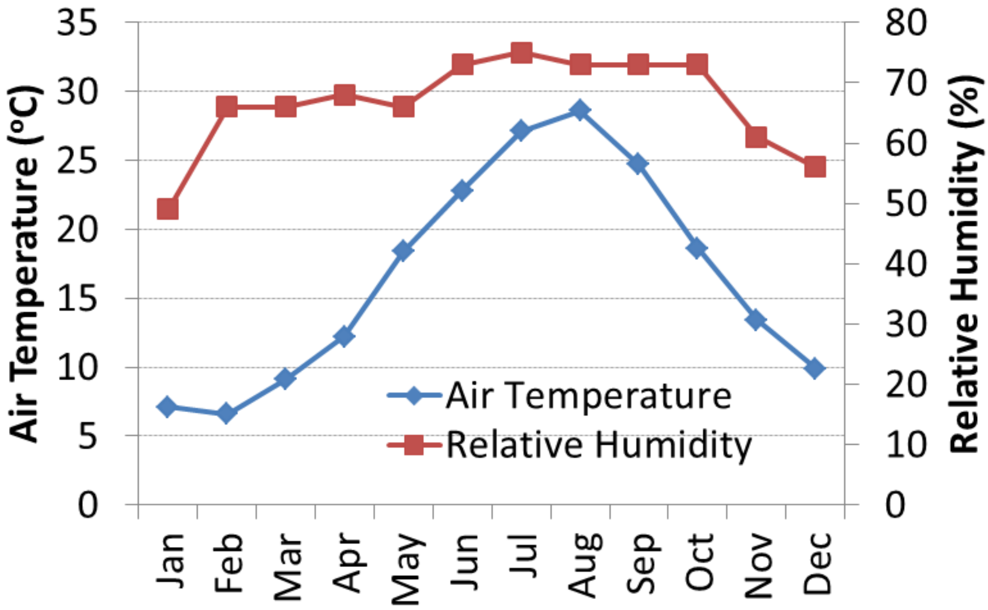


Fig. 23 Air temperature and relative humidity in Yokohama, 2010

[Source: JMA]

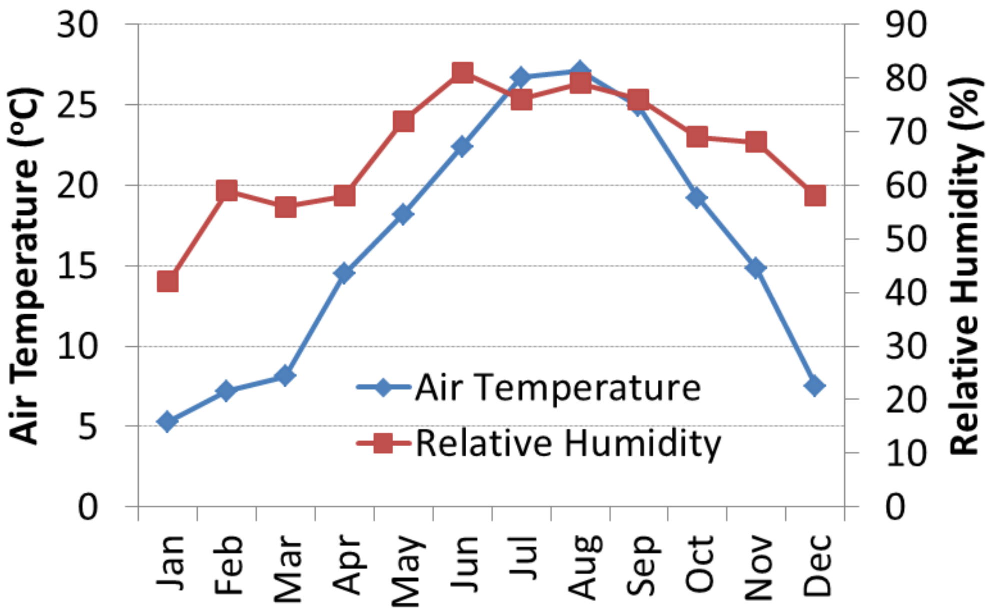


Fig. 24 Air temperature and relative humidity in Yokohama, 2011

[Source: JMA]

II.1. Baseline system

An existing commercial air-conditioning system using R-410A as refrigerant, nominal cooling capacity of 2.5 kW and catalogue COP of 5.68 is used as a baseline system, referred to Fig. 25. This conventional system has an air-cooled condenser, which is copper-tubing of 22.3 m length and 8 mm outside diameter. The copper tube of the condenser is attached with aluminum fin. Short specification of the baseline system is described in Table 1.

Table 1. ***Short-specification of the baseline air conditioning system***

Cooling capacity (kW)		2.5
Cooling COP		5.68
Refrigerant		R-410A
Outdoor unit	Height (mm)	550
	Width (mm)	780
	Length (mm)	290
	Weight (kg)	32

Fig. 25 Baseline air conditioning system: indoor unit (upper) and outdoor unit (lower).

II.2. Air-conditioning system with water-cooled condenser

An air-conditioning system with water-cooled condenser, which is called “water” air-conditioning system or “water” cooling system in this document from now on, is selected to perform experiment. Modified outdoor unit is pictured in Fig. 26. The condenser is cooled by connecting with cooling tower, as shown in Fig. 27 and Fig. 28. Water-cooled condenser is a double copper-tubing adjacent to each other, with total length of 21 m, refrigerant outside diameter of 6.35 mm and water outside diameter of 8 mm. The cooling tower used in the “water” air-conditioning system has nominal cooling capacity of 13.6 kW with a 0.25 kW pump and a 0.05 kW fan being commercially available as the minimum capacity, which is shown in Table 3.

Fig. 26 Outdoor unit of “water” air-conditioning system

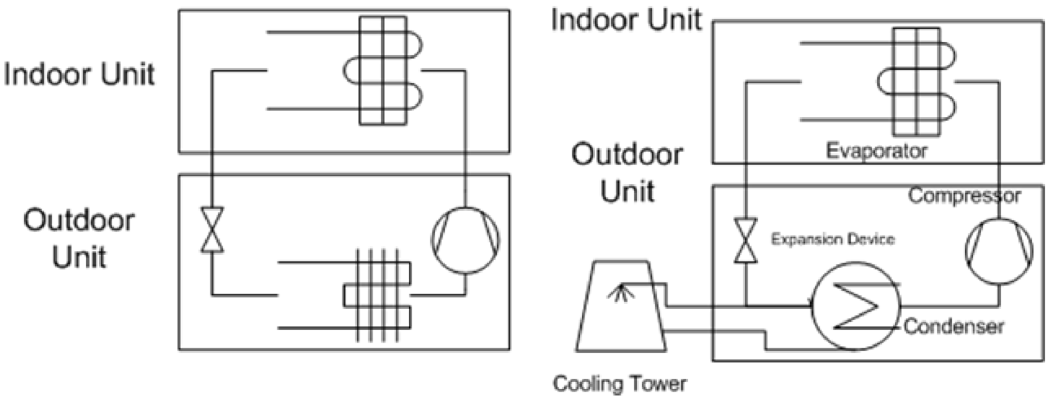


Fig. 27 Sketch of conventional AC (left) and “water” AC systems (right)

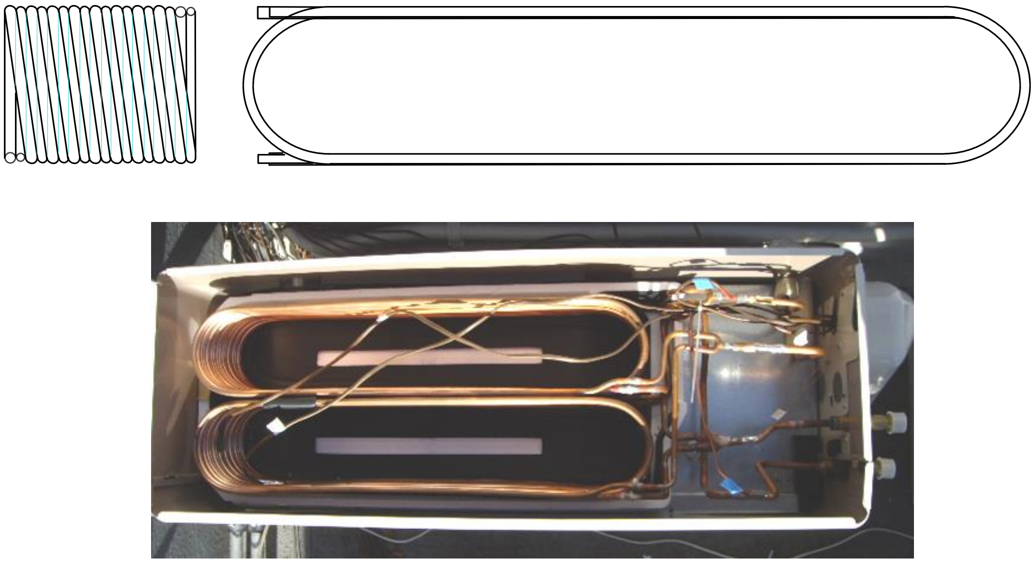


Fig. 28 Water-cooled heat exchanger

Table 2 Specification of the modified water-cooled heat exchanger

Refrigerant side	Outer diameter (mm)	6.35
	Thickness (mm)	0.6
	Length (m)	10.5
	Number of pass	2
Water side	Outer diameter (mm)	8
	Thickness (mm)	0.6
	Length (m)	10.5
	Number of pass	2

Table 3 Cooling tower specification

Cooling tower	Power	3 phase, 200V, 50 Hz
	Height (mm)	1199
	Diameter (mm)	600
	Cooling capacity (kW)	13.61 (37 ^{o}C → 32^{o}C, @ W.B 27^{o}C)
	Water flowrate (L·min^{-1})	39
	Fan power consumption (kW)	0.05

Fig. 29 Cooling tower

"Water" cooling system has been performed to test the condensing ability under various conditions compared to conventional system in actual summer weather. Experiment of the conventional system was operated 4 days in summer 2009 (daytime) and 5 days in summer 2010 (24 hours); while "water" system was operated 5 days long (24 hours) in summer 2010.

II.3. Proposed air-conditioning system

In this experiment, a proposed air-conditioning system is modified from the existing system by using new evapo-transpiration condenser. An actual view from the back side of outdoor unit is taken as in Fig. 30.

Fig. 30 Actual proposed outdoor-unit (from the back side)

Sketch of new condenser is shown in Fig. 31, which includes top water tank, copper tube covered by porous ceramics heat-exchanger and a bottom water tank. Water is spread automatically to horizontal direction by hydrophilic property of porous ceramics and flows downward vertically by gravity. As a result, ceramics surface is always kept wet. Air-flow by fan crosses this wet surface, enhancing evaporation. Remaining water that does not evaporate is collected in the bottom tank. The flow-rate of water is controlled manually by a valve to keep it comparable with rate of evaporation on ceramics surface. For this reason, collecting water in the bottom tank is negligible.

In the experiment with heat exchanger above, a small 13W-pump has been used to circulate water from bottom tank to the first row of heat-exchanger. Figuring out that the height of outdoor unit is not high and high flow-rate of water is not necessary, we decide to use direct tap water to provide water to ceramics. Besides, water condensed from indoor is re-used to save water. Hence, no additional pump or any other additional power is used.

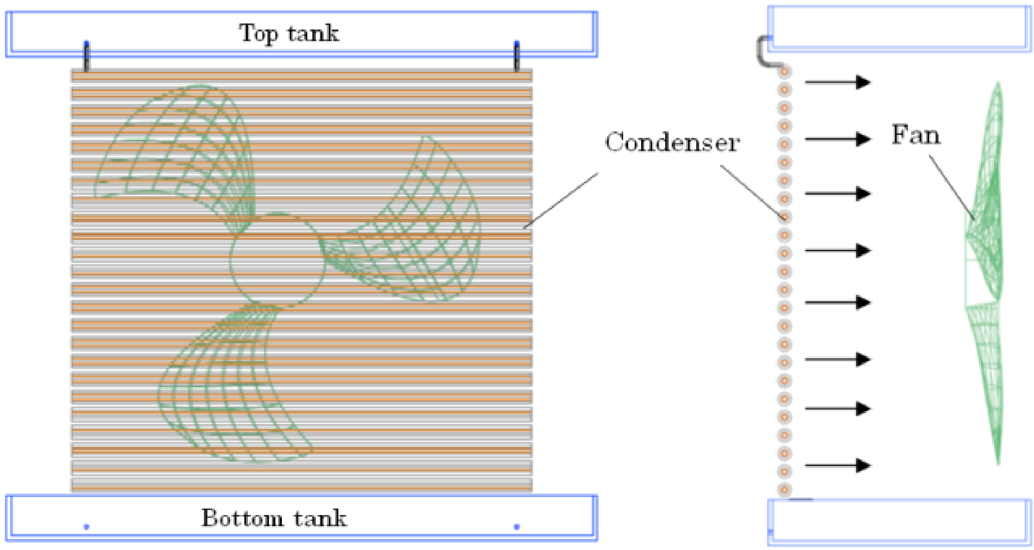


Fig. 31 Sketch of new evapo-transpiration condenser

Type-T-thermocouples are used to measure temperatures; data is connected to computer and recorded by data logger CADAC with the uncertainty of 0.02% of reading +0.03˚C. Temperatures were measured on the middle surface of condensers, right before and after compressors, before expansion valves; temperatures of the water in the top and bottom tanks. Besides, temperature and relative humidity of ambient air and outlet air from outdoor-unit are recorded by humidity loggers, Fig. 32. Watt-Hour Meter is used for measuring power consumption, Fig. 33.

Fig. 32 Humidity logger

Fig. 33 Watt-hour Meter and its specification

Chapter 4 Calculation

I. Heat-transfer coefficient of proposed heat-exchanger

Rate of heat transfer inside copper tube is calculated by:

$$q_{hw} = \dot{m}_{hw} c_{pw} \left(T_{ih} - T_{oh} \right) \quad (5)$$

where q_{hw}: rate of heat transfer (W);

$\dot{m}_{hw}$: mass flow-rate of hot-water (kg·s^{-1});

c_{pw}: specific heat capacity at constant pressure of water (J·kg^{-1}·K^{-1});

T_{ih}: hot-water inlet temperature (K);

T_{oh}: hot-water outlet temperature (K).

Heat-transfer rate through the whole process is also calculated by:

$$q_e = U_o A_o \Delta T_{lm} \quad (6)$$

where $\Delta T_{lm} = \dfrac{\Delta T_o - \Delta T_i}{\ln\left(\Delta T_o / \Delta T_i \right)}$: logarithmic mean temperature difference with $\Delta T_o = T_{hwo} - T_{fi}$ and $\Delta T_i = T_{hwi} - T_{fo}$, where T_{hwo} and T_{hwi} are hot water temperatures at the outlet and inlet, respectively; T_{fo} and T_{fi} are temperatures at the outlet and inlet of the fan, respectively; U is overall heat transfer coefficient based on outside area and A_o is surface area outside of the copper tube.

Since amount of heat transfer is conserved, hence: $q_e = q_{hw}$

Then, overall heat transfer coefficient is evaluated:

$$U_o = \frac{q_e}{A_o \Delta T_{lm}} \tag{7}$$

II. Air-conditioning performance

II.1. Coefficient of Performance (COP)

A typical standard vapour compression cycle is sketched in Fig. 34.

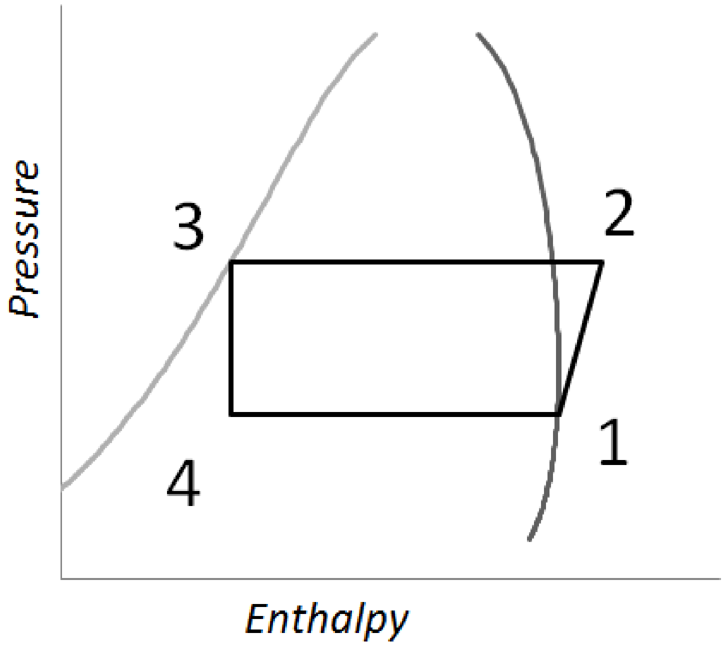


Fig. 34 Standard vapor compression cycle on P-h diagram

In order to estimate COP of the air-conditioning system, properties of refrigerant are evaluated using REFPROP Version 8, Fig. 35.

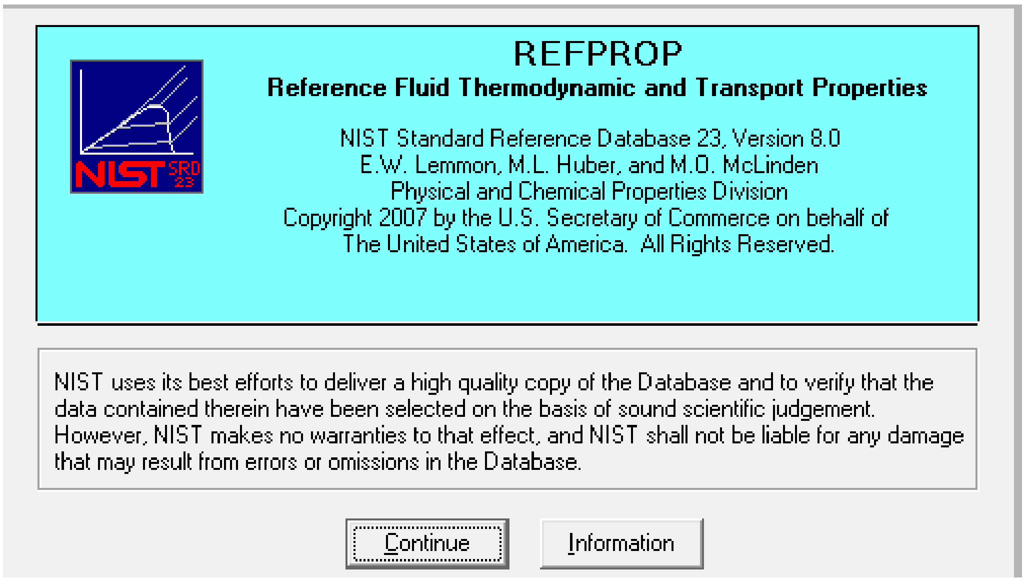


Fig. 35 REFPROP software

The process of evaluation the results follows the flowchart in Fig. 36. In the process, data input is from experiments of the three air conditioning systems.

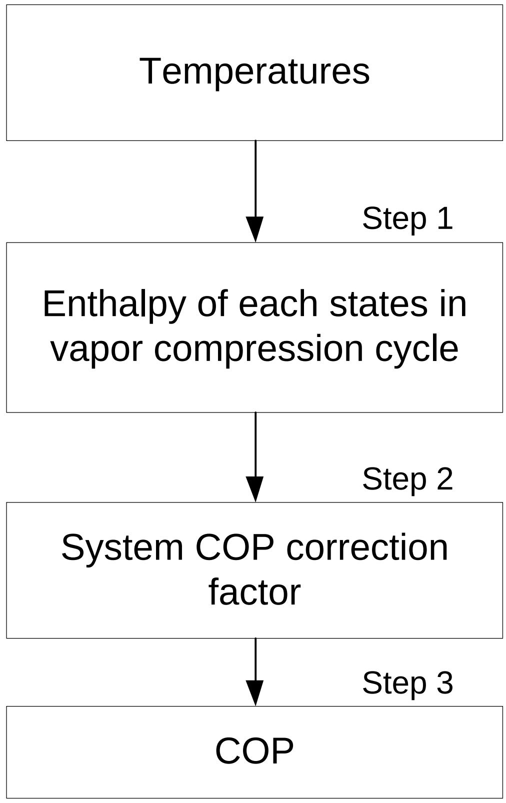


Fig. 36 COP evaluation flowchart

- ***Step 1:***

Temperatures at each point in the vapor compression cycle of each air conditioning system are recorded. Those points' enthalpies are evaluated by using PEFPROP.

- ***Step 2:***

Theoretical COP of a vapour compression cycle system is calculated by:

$$\mathrm{COP}_{theo} = \frac{h_1 - h_4}{h_2 - h_1} \tag{8}$$

where h_1, h_2, h_3 and h_4 are enthalpy at each stage in the vapour compression cycle following Fig. 34. They are evaluated by using REFPROP at respectively states of R-410a, which is the composition of R-32 (50 %) and R-125 (50 %):

- h_1 at the state of saturation liquid at evaporation temperature
- h_2 at the state of superheated vapor at condensation temperature.
- h_3 at the state of saturation liquid at condensation temperature
- h_4 at the state at evaporation temperature, which

From the value of evaporation and condensation temperature of refrigerant, h_1, s_1, p_1 and h_3, s_3 and p_3 are evaluated. Then, other parameters of each state are evaluated from the relationships as follows:

- Entropy $s_1 = s_2$;
- Pressure $p_2 = p_3$;
- Enthalpy $h_{4=}h_3$

In the manufacturer's catalogue, nominal condition for testing is set that: outdoor temperature of 35 ˚C, indoor setting temperature of 27 ˚C, indoor relative humidity is about 43 %, dew point temperature is 13.5 ˚C and evaporation temperature is 7 ˚C lower than dew point, which is 6.5 ˚C. Hence, the value of COP provided in the catalogue is also evaluated at this nominal condition. However, in reality, the system doesn't work exactly the same as testing condition. Hence, we assumed that actual COP will be shifted from theoretical COP by a factor:

$$\mathrm{COP} = \eta_{sys} \cdot \mathrm{COP}_{theo} \tag{9}$$

in which, η_{sys} is system correction factor.

In order to evaluate system correction factor, theoretical COP at similar condition, COP_{theo_n}, of outdoor and indoor temperatures, 35˚C and 27˚C respectively, in the case of actual current system is calculated:

$$COP_{theo\_n} = \frac{h_{1n} - h_{4n}}{h_{2n} - h_{1n}} \tag{10}$$

in which, COP_{cata} is COP in manufacturer's catalogue.

Then, system correction factor is evaluated:

$$\eta_{sys} = \frac{COP_{cata}}{COP_{theo\_n}} \tag{11}$$

- Step 3:

At each outdoor condition, estimated COP is also evaluated by using theoretical COP and system correction factor.

Theoretical COP at each case of outdoor temperature is calculated by:

$$\mathrm{COP}_{theo} = \frac{h_1 - h_4}{h_2 - h_1} \tag{12}$$

Hence, COP is estimated by:

$$\mathrm{COP} = \eta_{sys} \cdot \mathrm{COP}_{theo} \tag{13}$$

II.2. Performance of the proposed evapo-transpiration condenser

To judge the performance of proposed heat exchanger, evaporation rate of water at the condenser is calculated:

$$\dot{m}_e = \frac{q_e}{h_{fg}} \tag{14}$$

in which: h_{fg} is latent heat of water vaporization and q_e is the rate of cooling ability that from the process of water vaporization outside the condenser.

- ❖ This cooling capacity is equal to the condensing heat transfer of refrigerant inside condenser:

$$q_{cond} = q_e \tag{15}$$

On the other hand, condensing heat transfer is calculated from refrigerant side by:

$$q_{cond} = \dot{m}_{ref}\left(h_2 - h_3\right) \tag{16}$$

where $\dot{m}_{ref}$ is mass flow rate of refrigerant and h_2, h_3 are enthalpy after and before condenser, respectively.

Mass flow rate of refrigerant $\dot{m}_{ref}$ is evaluated from:

$$\dot{m}_{ref} = \frac{W}{h_2 - h_1} \tag{17}$$

where W is work from compressor.

We assume that:

- Compressor efficiency is 0.7 as common compressor efficiency
- Since mainly power consumed of air conditioning system is from compressor, other power consumption components, such as: fan, controlling system… are neglected.

Hence, work of compressor is calculated from power consumption, *P*, as:

$$W = 0.7P \tag{18}$$

Therefore, mass flow rate of refrigerant is able to be calculated:

$$\dot{m}_{ref} = \frac{0.7P}{h_2 - h_1} \tag{19}$$

❖ Latent heat of vaporization is evaluated by:

$$h_{fg} = h_f - h_g \tag{20}$$

where:

- h_f [J/kg] is enthalpy of saturated water at temperature of air-vapor mixture;
- h_g [J/kg] is enthalpy of saturated steam at temperature of air-vapor mixture.

These above enthalpy are evaluated at wet-bulb temperature, whose value is calculated from humidity ratio and water-vapor pressure p_s.

- Humidity ratio, ω , is calculated by the following formula (Stoecker 1982):

$$\omega = 0.622\, p_s / \left(p_t - p_s \right) \tag{21}$$

where p_t is atmospheric pressure, which is assumed to take the value of 101.325 kPa; while p_s is evaluated by:

$$p_s = \mathrm{RH}\, p_{s,sat} \tag{22}$$

in which $p_{s,sat}$ is water-vapor pressure at saturation condition at the same temperature.

Chapter 5 Results and Discussions

I. Proposed heat-exchanger

In this section, proposed heat-exchanger performance is presented.

Transpiration and evaporation are main principles in developing the new heat-exchanger. Transpiration helps to minimize energy used for pump using porous ceramics, while heat-exchange rate between inside and outside is enhanced by evaporation of water. Even ceramics' performance should be carefully investigated, its property to spread water automatically and continuously along copper-tube surface is satisfied. This experiment was carried out in summer weather to confirm the actual performance. Besides, experiment has done in 4 cases of different ceramic arrangement (vertical and horizontal) and types (90µm and 20µm porosity).

a. Overall heat-transfer coefficient

Overall heat-transfer coefficient of the new heat-exchanger is evaluated in order to compare with that of the air-cooled condenser. The overall heat-transfer coefficient of the proposed heat-exchanger is demonstrated in Fig. 37. The overall heat-transfer coefficient is about 340 to 1000 $W \cdot m^{-2} \cdot K^{-1}$ with the air velocity about 1.63 m/s. For air-cooled condenser, range of heat-transfer coefficient of copper tube and fin is from 50 to 90 $W \cdot m^{-2} \cdot K^{-1}$ with air velocity of 0.5 to 3 $m \cdot s^{-1}$ (Seshimo 1992). Therefore, the proposed heat exchanger has higher heat-transfer coefficient than that of the conventional one from 3.5 to more than 10 times. For that reason, it is expected that the size of condenser of the proposed air conditioning system will be smaller than the existing one. In addition, among the 4 cases of experiment, as explained in Fig. 17, the result is not clear that which one gives the best solution.

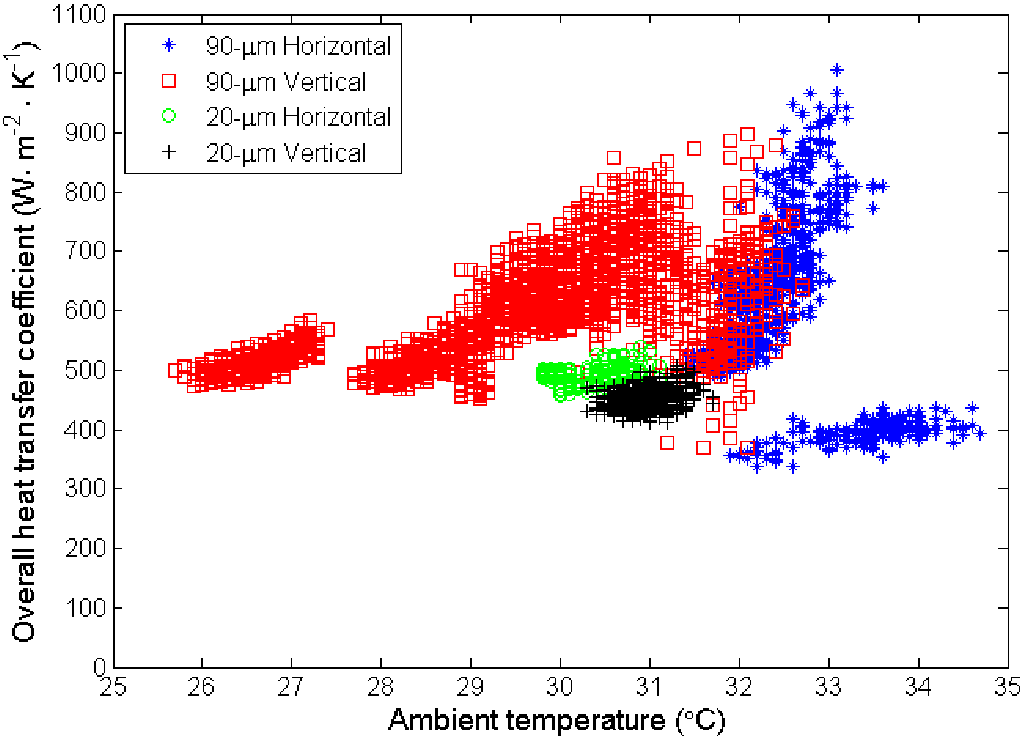


Fig. 37 Overall heat-transfer coefficient of 4 cases

b. Cooling water temperature

Cooling water circulation is important to this kind of heat exchanger. Cooling water in the tank has lower temperature than ambient temperature in most of the cases of experiments. As in Fig. 38, cooling water temperature is lower than ambient air temperature. The temperature difference between ambient air and cooling water in the tank, increases by time. This system has worked like a cooling tower, which is used to cool the water.

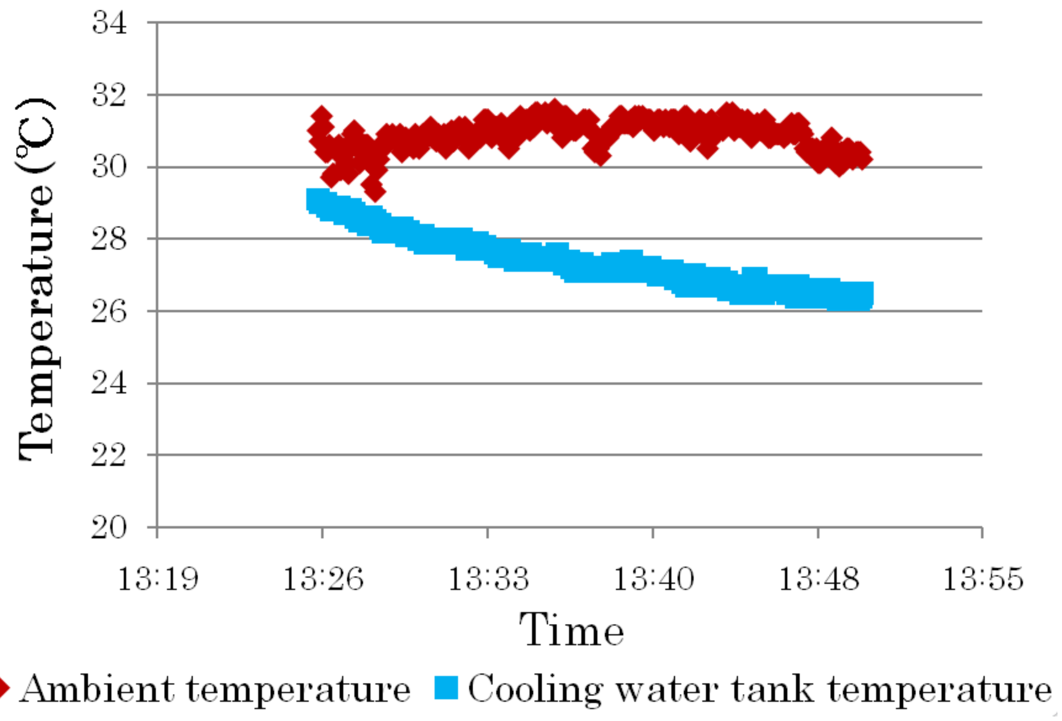


Fig. 38 Temperatures of ambient air and cooling water with 90-μm ceramics in vertical arrangement.

c. Temperature of outlet air flow

Because of water evaporation at wet surface of ceramics, the air, which flows out from the fan, has higher relative humidity than ambient air. Hence, temperature of this air is nearer to the wet-bulb temperature on psychrometric chart at ambient condition. From graph in Fig. 39, fan outlet temperature is lower than ambient air about 2 K.

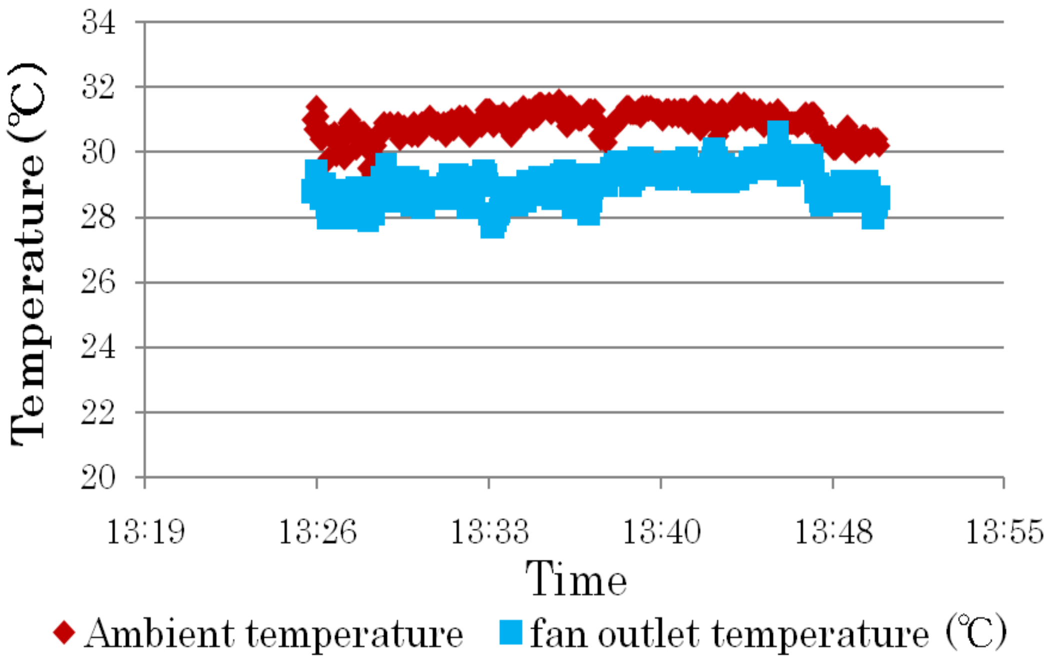


Fig. 39 Temperatures of ambient air and air outlet from fan of heat exchanger

Discussions

In Fig. 37, the horizontal axis is ambient temperature. For the experiment was performed in the actual weather, it is effected by both ambient temperature and humidity.

Since the heat exchanger can operate like a cooling tower, it is effective that a lot of energy is saved. Property of porous ceramics that can spread water automatically helps to save pumping energy consumption of a cooling tower.

Moreover, temperature of the air flows out from the system has lower temperature and higher humidity, it makes the space surrounding more comfortable. As study of Omer (Omer 2008), for air temperature lower than

36 ˚C, he did mentioned that, in warm humid conditions, like in summer, airflow can conduct heat from human skin. In steady airflow, human cooling sensation (CS), of airflow can be estimated in degrees Celsius using equation:

$$\mathrm{CS} = 3.67(V - 0.2) - (V - 0.2)^2 \tag{23}$$

when average airflow, V, is in m·s^{-1}.

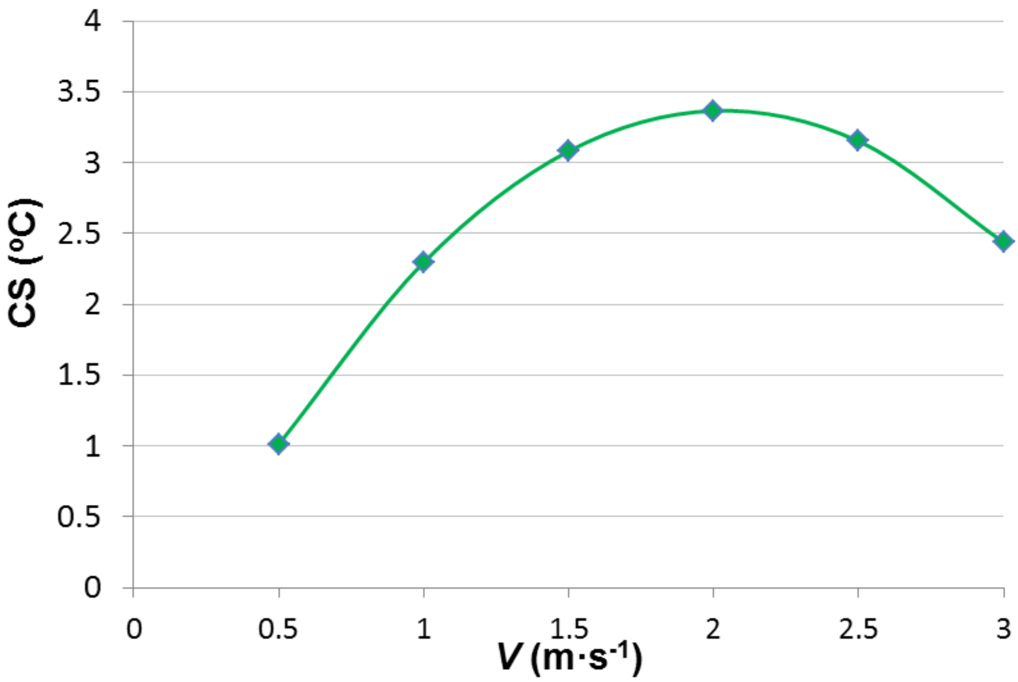


Fig. 40 Cooling sensation vs. air velocity

Also, from Fig. 40, with the air velocity at the outlet of about 1.5 m/s, people can feel cooler about 3 ˚C than the actual temperature if they stay in front of the heat exchanger.

In short, by using the evapo-transpiration heat exchanger with a small 13-W bath-pump, the hot water inside the tube is cooled, but the surrounding air is not heated. It is confirmed that if evapo-transpiration heat exchanger is applied to the condenser in the outdoor unit of the air conditioning system, waste heat to the environment, which contributes to the heat island problem, can be avoided.

II. Air conditioning systems

Besides the purpose of reducing or avoiding the waste heat to the surrounding air, performance of the air conditioning system with evapo-transpiration condenser, is necessary to be considered. In order to confirm the performance, three air conditioning systems need to be performed along with the simulation of Coefficient Of Performance, COP.

Three experiments with 3 air conditioning systems with the purposes as follow:

- Conventional system (baseline system) with *air-cooled condenser* of copper tube and aluminum fin: This experiment is to investigate the actual performance of the existing system.
- Air conditioning system with *water-cooled condenser*, in which cooling water is from cooling tower: The experiment is to confirm the possibility of reducing condenser temperature, which can lead to the reducing in power consumption of the compressor.
- Air conditioning system with the proposed *evapo-transpiration condenser*: The experiment is to test the performance of the prototype system and its possibility to reduce the bad environmental effects.

Along with these above experiment, evaluating COP is necessary. Based on the theory, to evaluate COP, cooling capacity and net work must be evaluated. However, in reality, except outdoor condition is controlled like in the manufacturer testing facility, actual cooling capacity is difficult to control because of the fluctuation of the weather. Therefore, in order to evaluate the real COP, a simulation program is developed.

II.1. Performance of the conventional air conditioning system for cooling mode

For the present purposes, condenser temperature is concerned. Thermocouple at the middle of the condenser copper-tubing, which is supposed to be very close to condenser temperature, is used as condenser temperature.

In the experiment, data is recorded every minute. In order to compare the performance among air-conditioning systems, integral power consumption for an hour is considered. From now on, hourly average temperature of outdoor and middle-condenser are briefly called as outdoor temperature and condenser temperature.

Temperature at condenser is shown in Fig. 41 for experiment with conventional air conditioning systems in the years of 2009 and 2010. In addition, the results shows that, in conventional system, temperature difference between condenser and outdoor, Δt, is getting higher as outdoor temperature increases, as shown in Fig. 42.

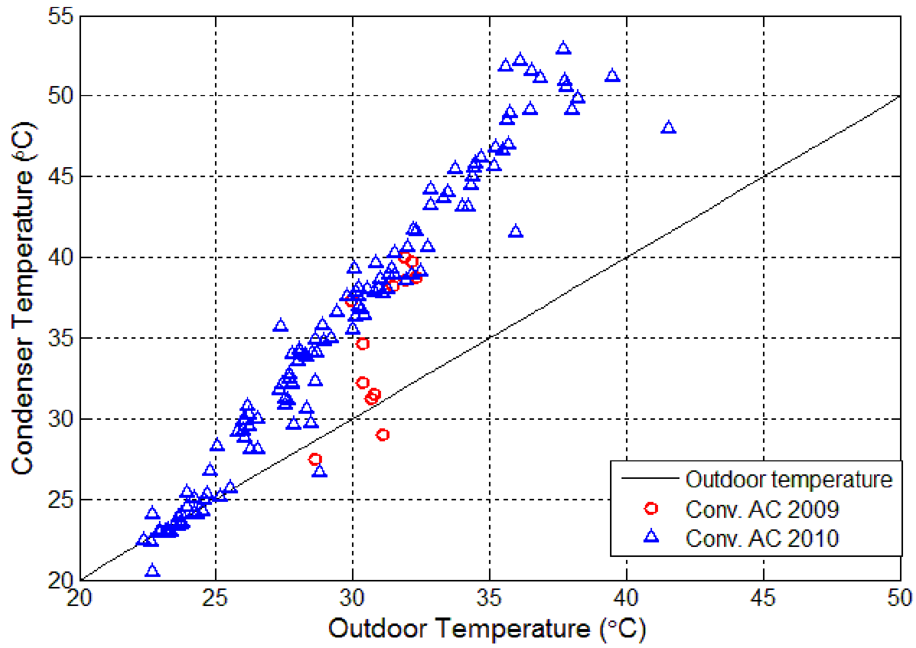


Fig. 41 Temperature at condenser of conventional air conditioning system

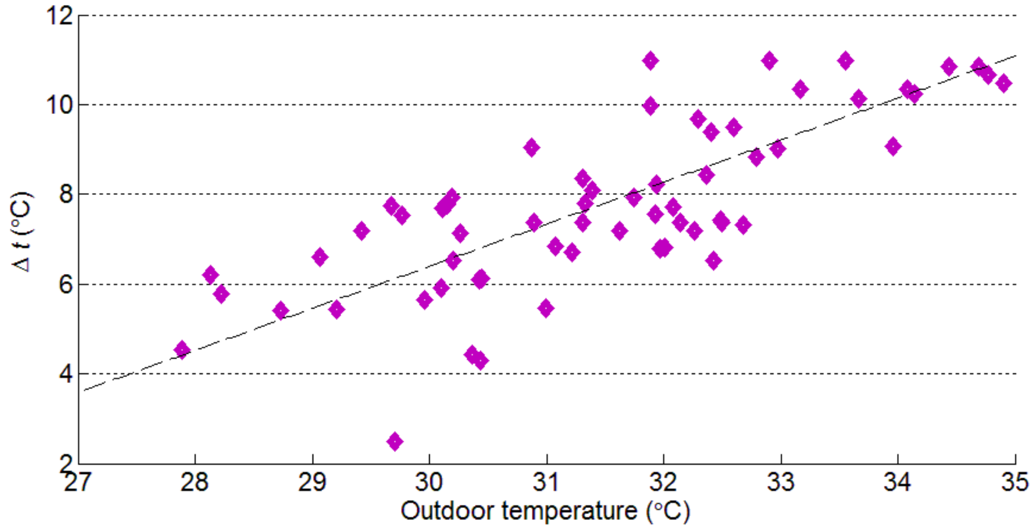


Fig. 42 Temperature difference between condenser and outdoor temperature

With the same indoor setting temperature, when outdoor temperature is higher, integral power consumption increases. The increase due to two reasons:

(1) Higher temperature difference between outdoor and indoor, which increases the cooling load required indoor

(2) Higher pressure difference between condenser and evaporator dues to higher temperature difference between them. From Fig. 42, as outdoor temperature increases, temperature difference between condenser and the outdoor air, *Δt,* increases. For that reason, the higher *Δt* is, higher energy requires.

The relationship of the change of integral power consumption with respect to *Δt* is shown in Fig. 43. It is clear that *Δt* has slightly affected to the integral power consumption of the air conditioning system, beside the cooling load required indoor.

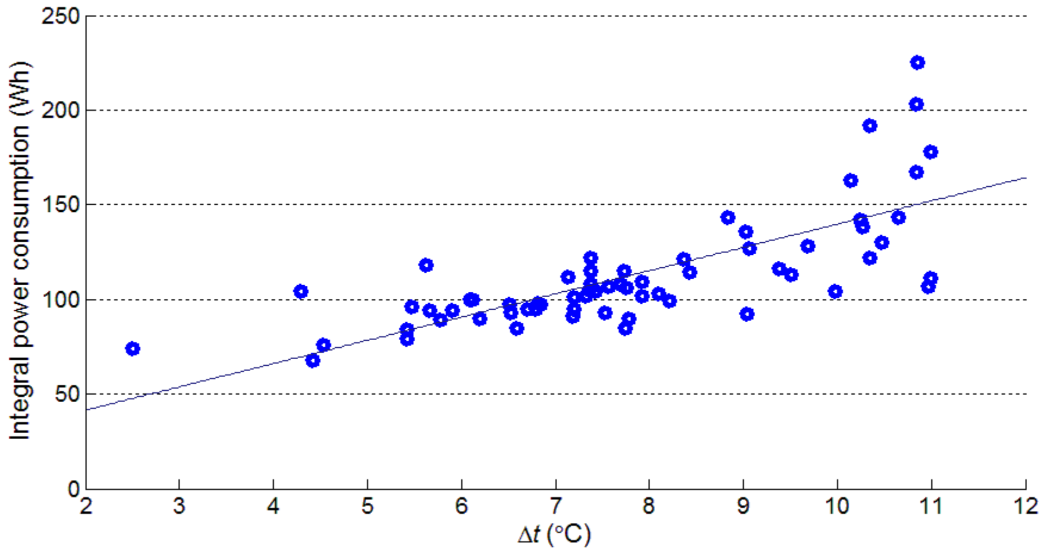


Fig. 43 Integral power consumption vs. temperature difference between condenser and outdoor

For the above reasons, higher performance is expected to achieve at lower condenser temperature. In addition, Fig. 44 shows common operating temperature range of the condenser is from 35 to 45 ^{o}C. When condensing temperature is more than 45 ^{o}C, power consumption starts to increase rapidly.

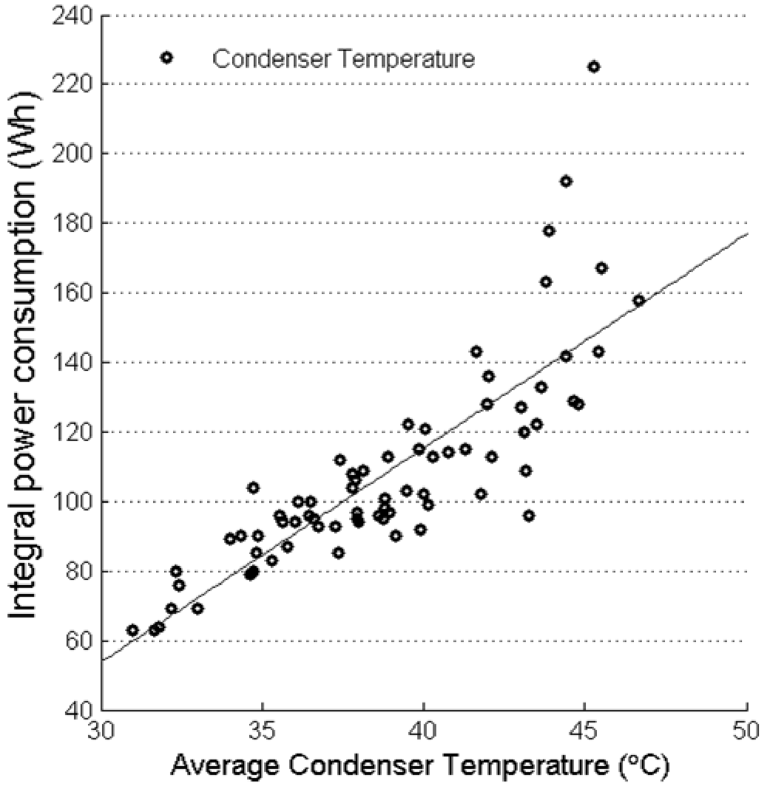


Fig. 44 Integral power consumption vs. condenser temperature

As simulated from using the methodology in Chapter 4, the graph is described in Fig. 45. The result shows that COP reduces with high condenser temperature.

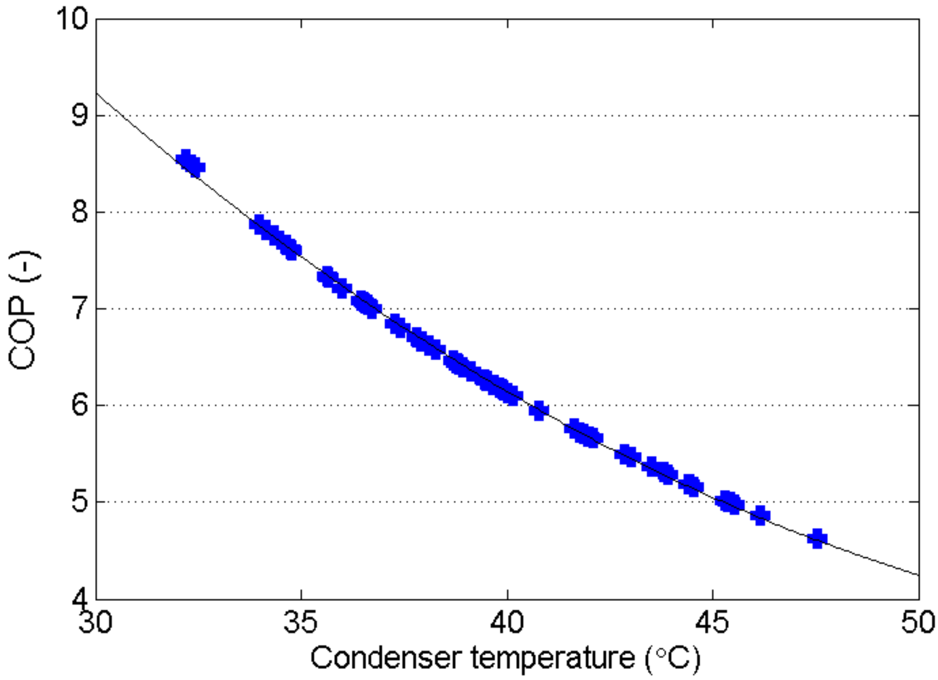


Fig. 45 Estimated COP vs. condenser temperature

Exergy of waste-heat from air-cooled outdoor unit

Beside the bad effect to increase power consumption, hot-air flow of waste-heat to the environment also contains exergy, and it is not utilized for other purposes. In consequence, it transfers directly to the ambient air and performs some change to the surrounding air. Amount of this exergy is calculated by:

$$\dot{E} = \dot{Q}_{cond}\left\{\left(T_{ao} - T_o\right) - T_o \ln\frac{T_o}{T_{ao}}\right\} \tag{24}$$

Waste-heat and its exergy of hot-air flow from air-cooled condenser are described in Fig. 46. Because condenser-temperature increases drastically as outdoor-temperature increases, exergy also increases considerably. For example, in this experiment, at 35 ˚C outdoor-temperature, condenser temperature is nearly 45 ˚C, waste-heat is about 0.9 kW and its exergy is estimated to be approximately 0.15 kW.

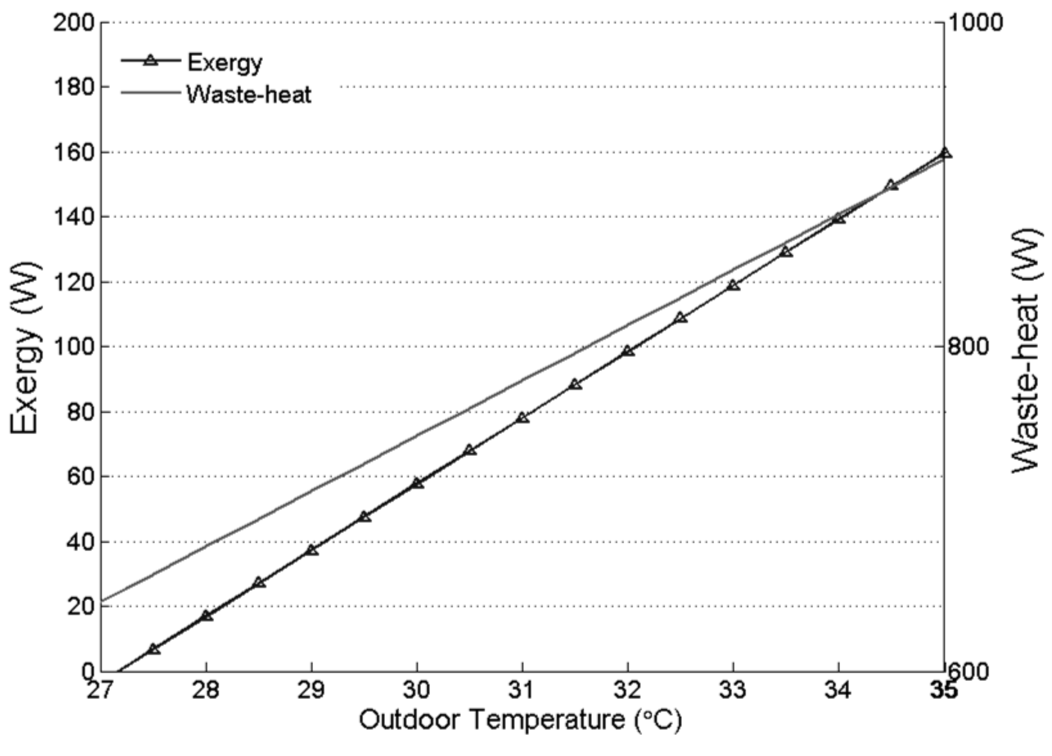


Fig. 46 Waste-heat and its exergy of air-cooled outdoor-unit

Therefore, the purpose of the study is to reduce the difference between condensing temperature and outdoor temperature as much as possible, to avoid the increase of power consumption due to the performance of condenser itself, and to reduce the bad effect to the environment by the waste-heat release from the air conditioning system being operating.

II.2. Performance of the air conditioning system with water-cooled condenser

Even power consumption of cooling tower is high compared to the power consumption of small residential air conditioning system, the water-cooled condenser is used to test the possibility of reducing the condenser temperature in the case of using high cooling capacity from cooling tower.

Temperatures of condenser is sketched in Fig. 47. They are compared with those of conventional ones at outdoor temperatures from 27 ˚C to 35 ˚C. Condenser temperature increases sharply with increasing of outdoor temperature. Temperature at the conventional condenser surface is approximately from 5 to 10 ˚C higher than that of the water-cooled condenser, which is nearly the same as outdoor temperature.

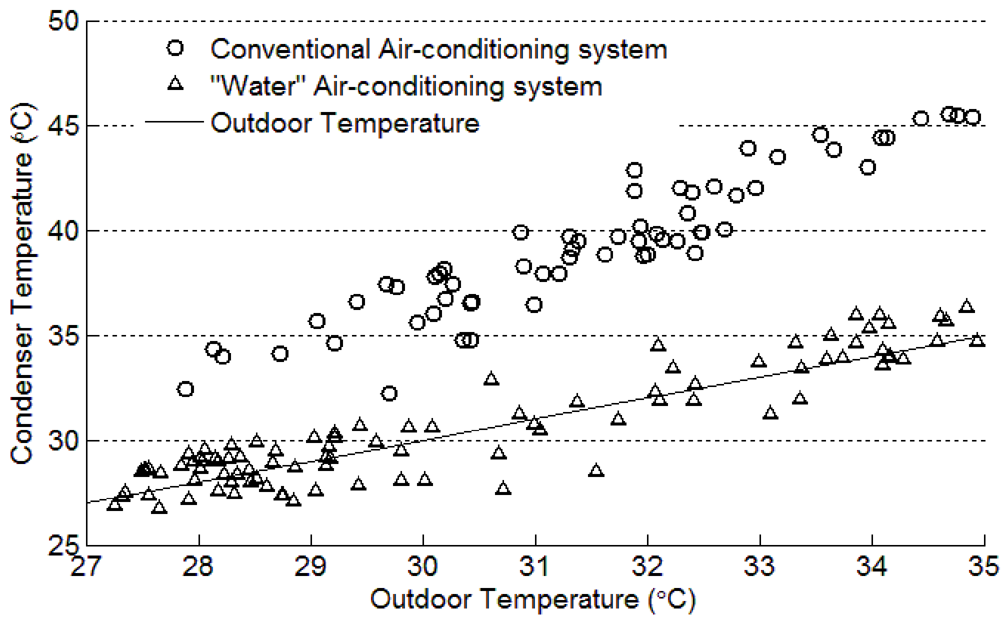


Fig. 47 Condenser temperature vs. outdoor temperature

Discussions

As mentioned, the integral power consumption of the “water” air-conditioning system is not included since cooling tower has high power consumption. Based on temperatures from experimental results, COP of the “water” system is evaluated by simulation results with the input data recorded from experiment. It is higher than that of the conventional system up to more than 30 %, as shown in Fig. 48.

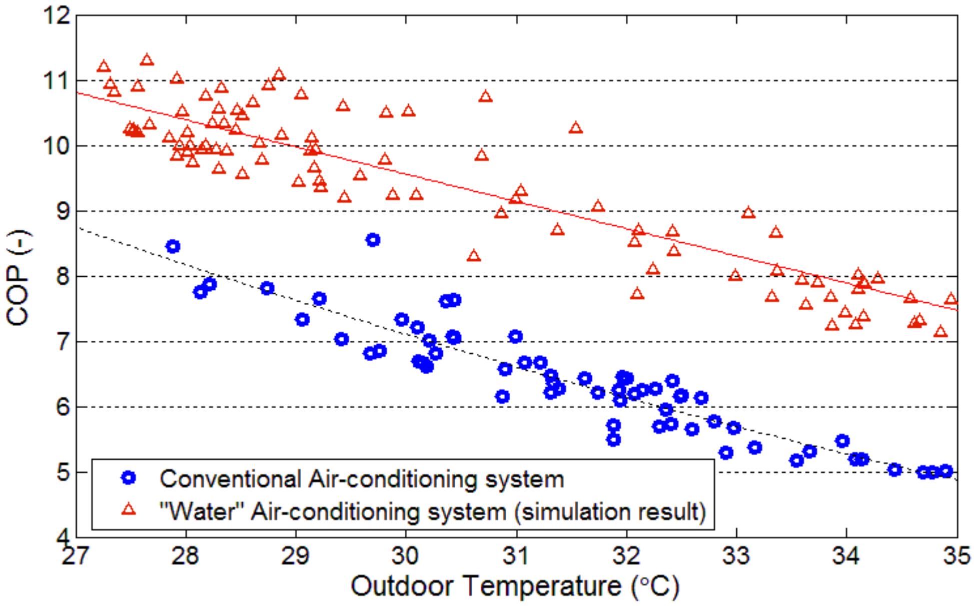


Fig. 48 Estimated COP vs. outdoor temperature

Therefore, with the same cooling space, if any air-conditioning system can have low temperature at the condenser as in the “water” air-conditioning system without using cooling tower, its integral power consumption is expected to save up to 30 %, as shown in Fig. 49.

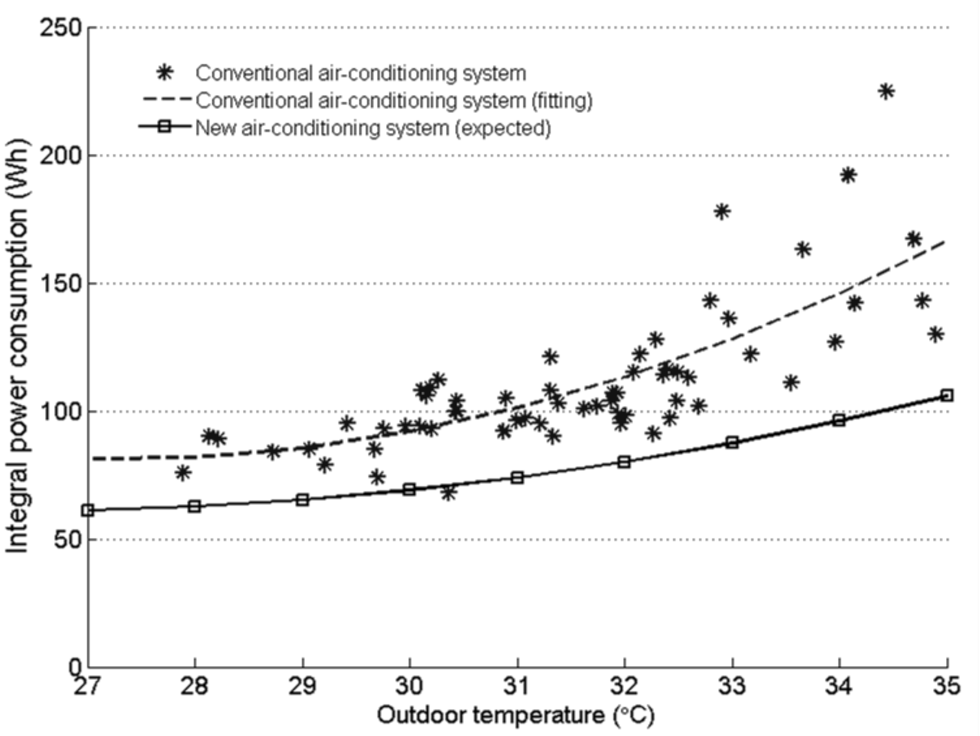


Fig. 49 Integral power consumption vs. outdoor temperature

III. Performance of the proposed cooling system

III.1. Results

a. Outlet air from outdoor-unit

Typical examples of relationship between outlet-air temperature of conventional and proposed outdoor-units are shown in Fig. 50 and Fig. 51, respectively, with every minute data recorded. From the experiment with conventional system, temperature of outlet-air is higher than outdoor-temperature up to 5 °C, as shown in Fig. 50. On the other hand, in case of using the proposed system with the same cooling space, outlet-air temperature is as nearly as the ambient temperature, Fig. 51.

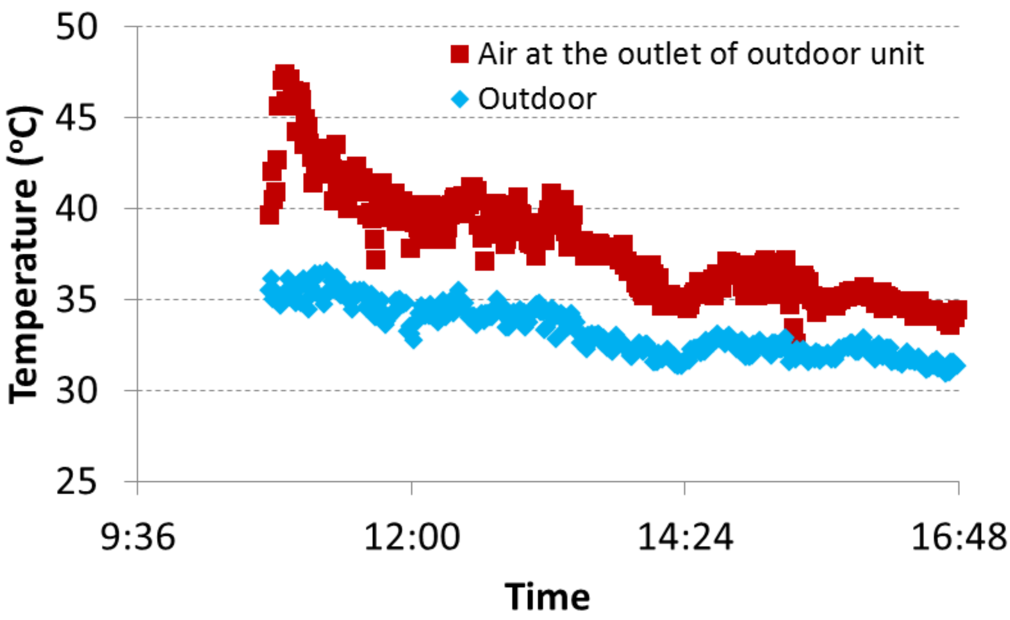


Fig. 50 Temperatures of outlet air of conventional system on Aug 7th 2009

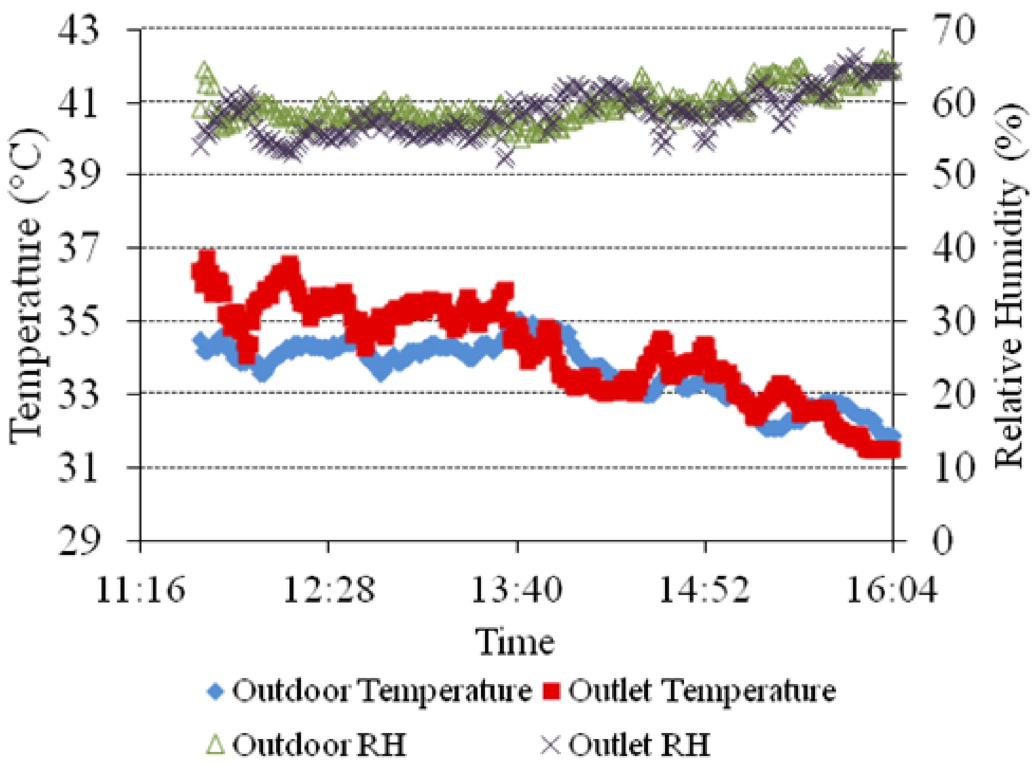


Fig. 51 Temperatures of outlet air of the proposed system on Aug 6th 2011

The hourly average outlet-air temperature from outdoor unit compared to that of the outdoor is shown in Fig. 52. As principle is to enhance cooling effect from water evaporation by flowing ambient air through wet surface, relative humidity (RH) should be considered. In summer weather, relative humidity is commonly about 40 % to 80 % and more than 90 % on rainy days, wet-bulb temperature of 16 to 25 °C. As shown in Fig. 53, outlet-air temperature is as low as that of the outdoor, while relative humidity of air that gets out from the fan is slightly higher than that of the ambient, at an average of 5 %.

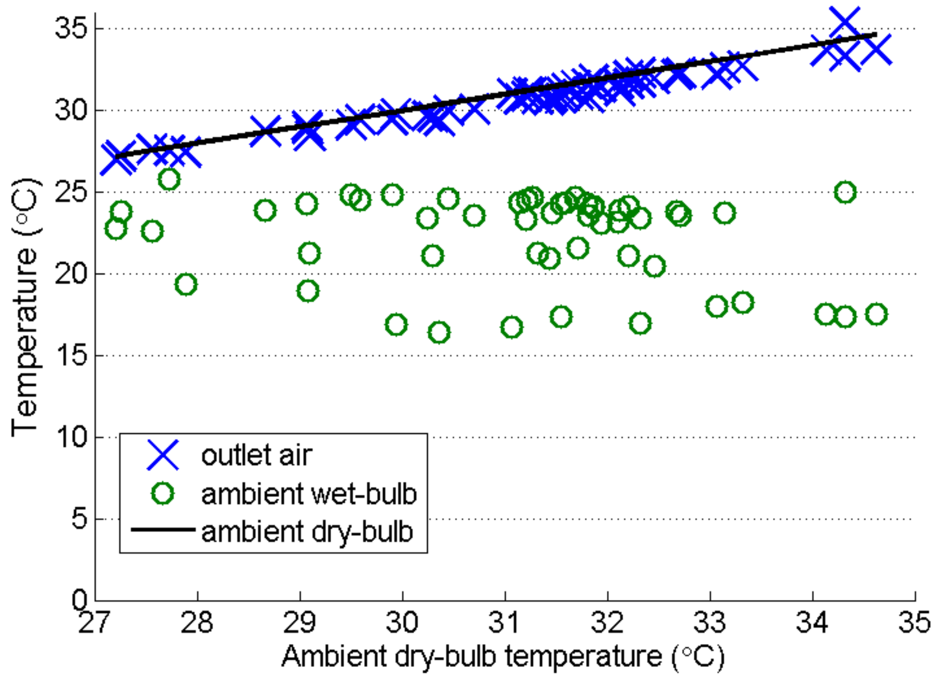


Fig. 52 Temperatures of outlet air, dry-bulb and wet-bulb temperature

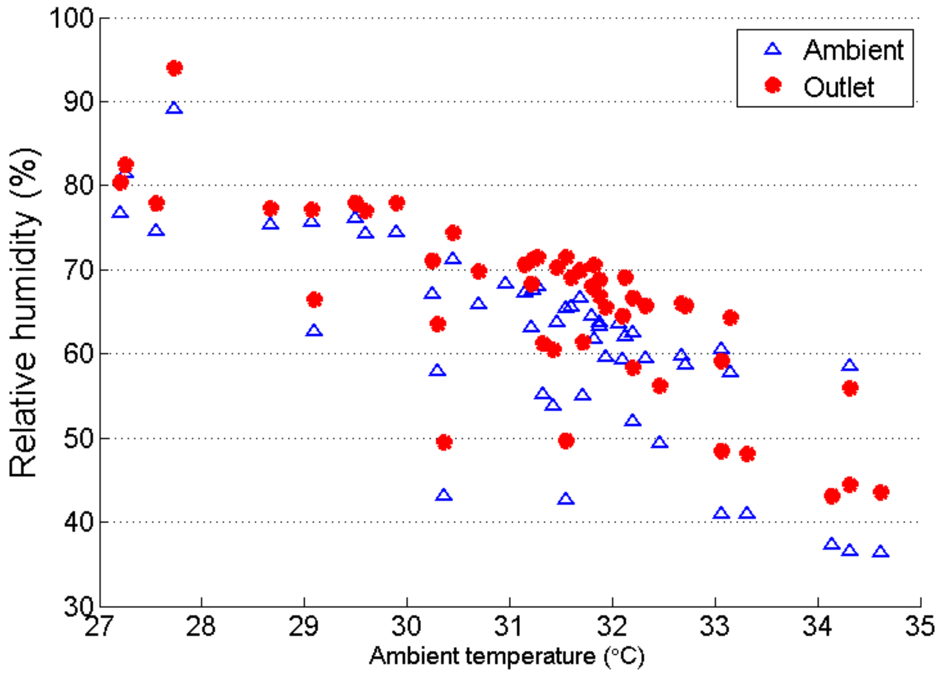


Fig. 53 Relative humidity of inlet and outlet air

In this experiment, the condenser is modified based on the conventional air-cooled copper tube with the same dimension without fin, which is external force convection of air passing bank of tubes. The cooling capacity of the proposed condenser depends on the evaporation rate at ceramics surface. This amount of water evaporation is affected by outdoor temperature and relative humidity. From Figs. 54 and 55, rate of evaporation seems to be affected stronger by outdoor temperature. It may be explained that outdoor temperature affects to the total cooling capacity required for indoor space; and the cooling capacity leads to the operation of the fan in the outdoor unit through the controlling system, which is optimized for conventional air-cooled heat exchanger. Therefore, the cooling capacity per unit area also increases when outdoor temperature increases, as in Fig. 56.

With the outside ceramics surface of about $0.5\,m^2$, range of evaporation rate is about from 0.1 g/s to 0.35 g/s of water and cooling capacity of the condenser per unit area can get up to 1680 W/m^2 using the existing fan.

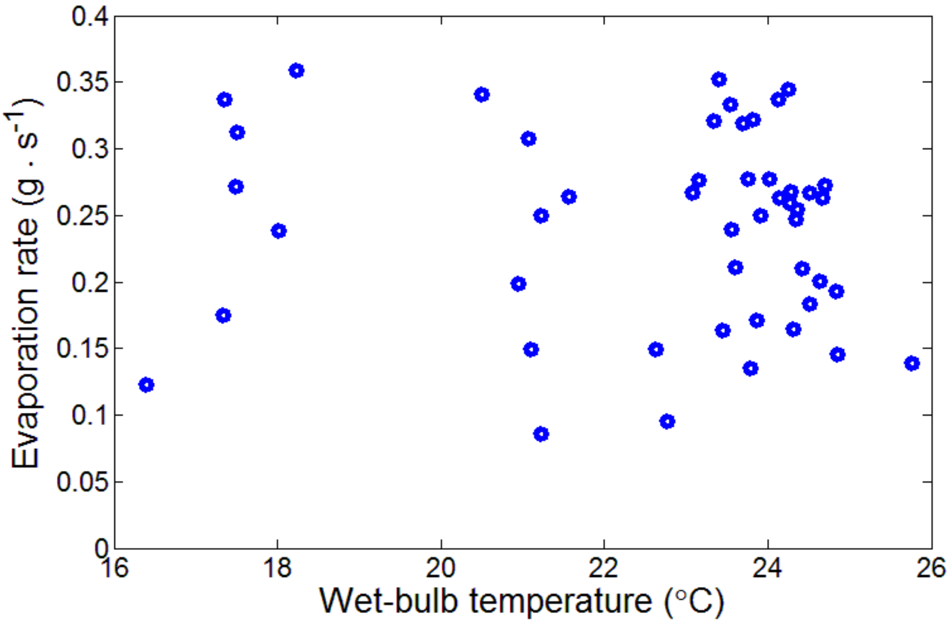


Fig. 54 Evaporation rate vs. outdoor wet-bulb temperature

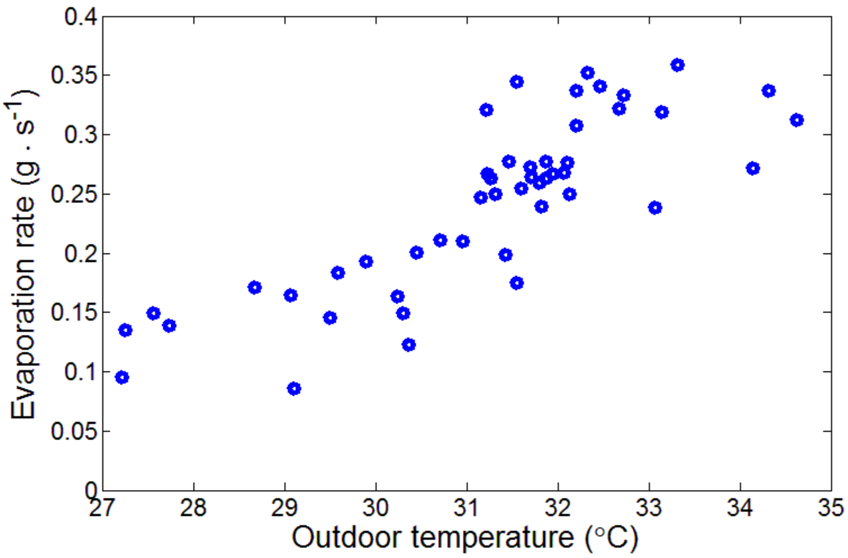


Fig. 55 Evaporation rate vs. outdoor temperature

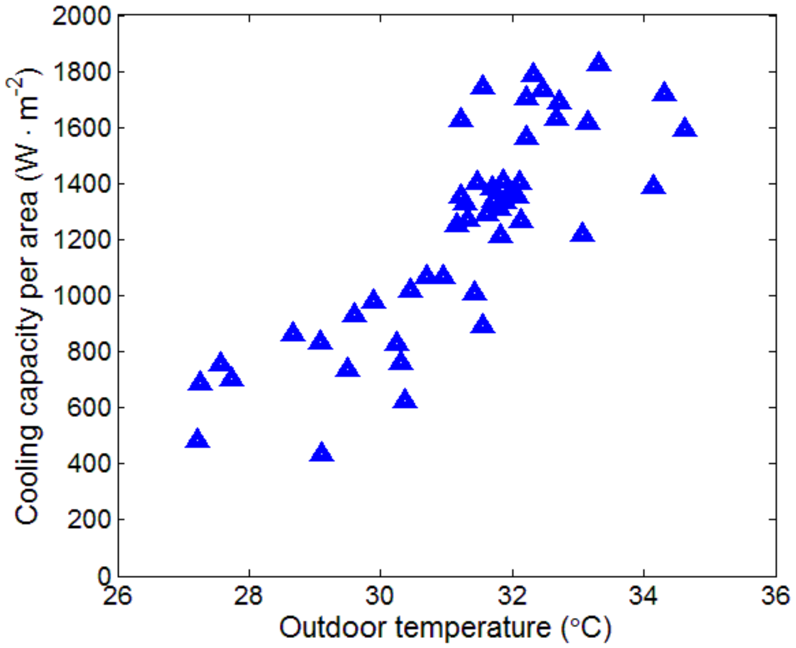


Fig. 56 Cooling capacity of the condenser per unit area

This experiment was conducted is mostly in warm humid zone. Also, as existing fan in outdoor-unit has velocity of 0 to 3 $m \cdot s^{-1}$, it is expected that there is some degree of cooling sensation, which is shown in Fig. 40. It means that people who stay in front of the outdoor-unit possibly feel some degrees cooler than the surrounding.

b. Power consumption

Different from the above water-cooled condenser in the "water" air-conditioning system, the condenser in the proposed air conditioning system is cooled by using evapo-transpiration instead of cooling water from cooling tower. In Fig. 57, result of condenser temperature is in the range of -5 ˚C and +5 ˚C compared to outdoor temperature. This result is the same as the result with the water-cooled condenser that we use in the "water" air-conditioning system in the previous study.

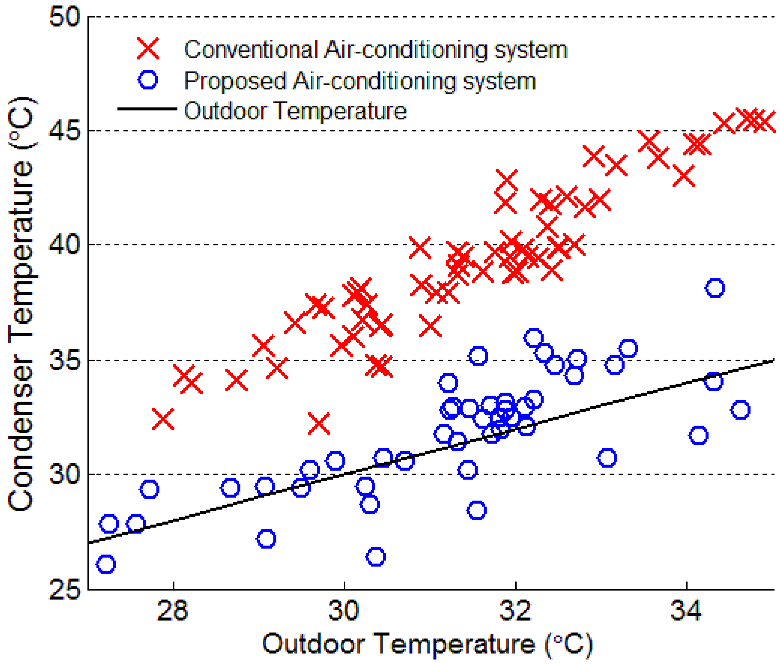


Fig. 57 Temperature at condenser

With the same setting indoor-temperature of 27 ^{o}C, even integral power consumption may also depend on the trend of outdoor temperature of the day; hourly integral power consumption at each outdoor temperature should be considered.

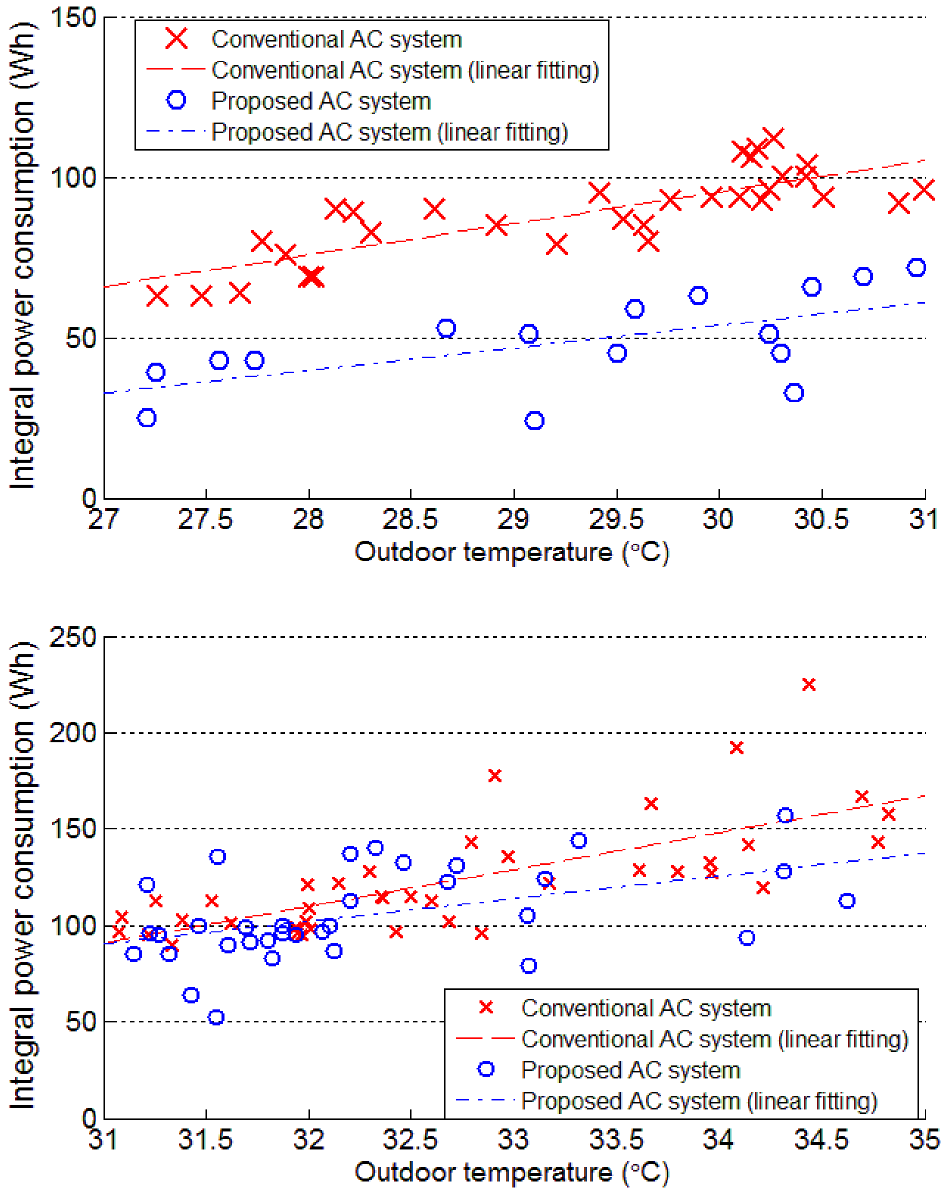


Fig. 58 Integral power consumption vs. outdoor temperature

For this prototype system, integral power consumption of the proposed system is lower than the conventional system and can be saved up to more than 30%, as clearly see in the upper figure of Fig. 58. For outdoor temperature beyond 31 ^{o}C, the difference is not clear, as shown in the bottom figure. The reason is considered from different effects: (1) the more frequently of the on-off operation since it takes shorter time to get to the setting temperature indoor for small space, in other words, the controller has forced the compressor to work with high ability for larger space; (2) the fan has worked harder than it should be with high outdoor temperature.

III.2. Discussions

a. New outdoor-unit heat exchanger

From the result in Fig. 57, condenser temperature of the proposed system is about outdoor temperature, which is lower than that in the conventional air-cooled condenser from 5 to 10 ˚C. Since outlet-air temperature is also as near as outdoor-temperature, almost no waste-heat releases from proposed outdoor unit. Therefore, we can confirm that new condenser has qualified the objectives of reducing condenser-temperature, no waste-heat and smaller condenser size.

b. Air-conditioning system performance

For the specific system used in this study, the power consumption measured with the prototype one is lower than that of the baseline conventional system, up to about 30 %. It is almost confirmed with the simulation result of the air-conditioning system with water-cooled condenser as mentioned in Fig. 49.

As condenser temperature has been reduced in whole range of outdoor temperature, it can be expected that if the system is optimized in the purpose to use new evapo-transpiration heat exchanger as condenser, energy can be saved more without release waste heat to the environment.

As shown in Fig. 59, temperature difference between condenser and the outdoor, Δt, has been shifted from the range of 5 to 15 ˚C in the air-cooled condenser, to the range of -5 to +5 ˚C in the new condenser. As a result, higher COP is expected to achieve with the proposed system.

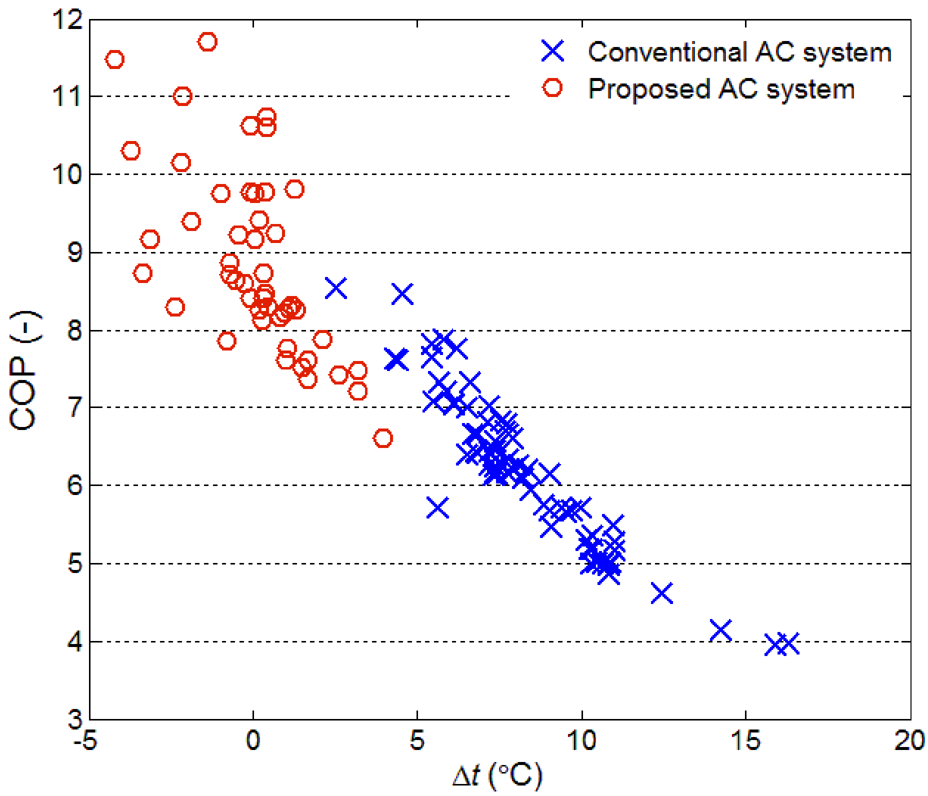


Fig. 59 Estimated COP vs. temperature difference of condenser and the ambient air

c. Outdoor-unit location and size

Based on the principle of evapo-transpiration, besides using fan to enhance evaporation rate, solar radiation also should be utilized. It is preferable that the condenser faces directly to the sun. In addition, the proposed outdoor unit can be located in the semi-open space, i.e., interface space to create surrounding space around the house, which is somehow reducing cooling load indoor. Hence, the proposed air-conditioning system has double cooling effect both inside and outside of the room.

Last but not least, as the new condenser has higher heat transfer coefficient compared to the existing air-cooled condenser, by optimizing the air flow for the proposed condenser, heat transfer surface area can be optimized; thus, a smaller-size outdoor-unit can be expected.

Chapter 6 Conclusions

The cooling system proposed is one of the devices having a natural way of cooling, i.e., evapo-transpiration like plants, which doesn't heat the air in summer time. The realization of the proposed system was experimentally confirmed as follows:

- Evapo-transpiration is applied in the proposed condenser of copper tubing covered with porous ceramics, which can spread water automatically by the capillary phenomenon. The porous ceramics can be used for three years without degradation of the performance even using the tap water. Hence, the durability of the ceramics would be reasonable.
- Experiment of heat exchanger was first performed to test the performance of porous ceramics. The result shows that overall heat transfer coefficient can increase 3.5 to 10 times compared to the air-cooled condenser. In addition, cooling water temperature is getting lower than that of the ambient air in the experiment, which can prove that the proposed heat-exchanger can work like a cooling tower even with very small pump. Moreover, temperature of the air flown out from the fan is also slightly lower than that of the ambient air. This air flow is possible to create comfortable space.
- Experiment of the "water" air-conditioning system, which has water-cooled condenser connected with cooling tower, has been performed to test the possibility of reducing condenser temperature. The results has shown that condenser temperature

can reach to the outdoor temperature. From simulation result, it is expected that the system with temperature at condenser nearly as temperature of outdoor is possible to save up to 30 % energy consumption.

- The experiment for the proposed cooling system using evapo-transpiration condenser was the retrofit system from the commercial system, in which aluminium fin in the condenser is replaced by porous ceramics. The results are the following:

    - The system does not release higher temperature heat to the environment since the air coming out from outdoor unit has temperature nearly the same as outdoor temperature and relative humidity is slightly higher than that of the ambient air with the average of 5 %.

    - According to the sensation cooling definition, while the system is operating, air flow from the fan of outdoor unit can provide some cooling effect to people who stand in front of it. It is said to reduce people's feeling temperature up to 3 degrees compared to the actual outdoor temperature at velocity of air from the fan is about 1.5 m/s.

    - In this specific experiment, energy consumption has confirmed to be possible to save up to 30 %, which is almost the same as expected from simulation result. Therefore, it is expected to save energy using the evapo-transpiration condenser in cooling system.

- Besides tap water, cooling water can be utilized from the extract water from the humidity of the indoor air to conserve water.

As written in the background, temperature of thermal energy is free from the energy conservation. The proposed air-conditioning system has a higher performance by a smaller temperature difference between evaporator and condenser. Moreover, the possibility of creating an environmental-friendly equipment from engineering side by keeping the temperature of the outside air stable at each weather condition, has been confirmed in this study by using evapo-transpiration.

In conclusion, this study proposes an air-conditioning system, which is satisfied not only eliminating CO_2 emission but also not heating the outside air as a measure for the global warming and urban heat island issues.

Further Work

In order to apply the system in countries with four seasons, like Japan, it should be combined heating mode in the system. Solar thermal panel would be applied to the system.

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