APPENDIX A

Publication

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Effect of Thermosyphon Evaporative Length on Temperature Reduction of Cool Water

Nammont Chotivisarut* and Tanongkiat Kiatsiriroat

Department of Mechanical Engineering, Faculty of Engineering, Chiang Mai University,
Chiang Mai, Thailand 50200
*E-mail: nammont4426409@hotmail.com

Abstract
The objective of this study is to evaluate thermal characteristics of vertical thermosyphon heat pipe, generating cool water at its evaporative section. The effect of the evaporative length on the heat transfer of thermosyphon heat pipe has studied experimentally and numerically by CFD. In the experimental study, three copper thermosyphon heat pipes with R-134a inside of 19.05 mm. diameter with 1.00, 0.85, 0.70 m. length are used to extract heat from 3.5 L. water, containing in a well-insulated vessel. At the condenser part, there is cooling water, maintained at 10 °C. The heat transfer coefficients of water, having higher evaporative length increase monotonously with the ratio of evaporative length to the water level. The value of the heat transfer coefficients are used to estimate the history of water temperature in the vessel. The CFD simulated temperature agrees very well with the experimental data within 8.0 percent errors.
Keywords: thermosyphon, temperature reduction, cool water, heat pipe

1. Introduction

Nowadays, changing of building style and function has led to a greater dependence on artificial forms of lighting, heating, cooling and ventilation within modern buildings. Recent environmental concerns have however, led to a greater focus on traditional passive methods of solar control, natural ventilation and other passive cooling methods. Designers, architects and engineers have adapted many traditional basic principles to fit in with the modern office environment, both in terms of building practices and materials and in the way in which we work today, often resulting in innovative design solutions.

One of the passive cooling methods is night sky cooling technique, which has been studied by many researchers. Vimolrat [1] has designed the new alternative night sky cooling systems by using the thermosyphon attached with the radiator, with no water circulation as shown in Figure 1.

This work is to extend the study of Vimolrat by considering the effect of evaporative length on the heat extraction rate of the cool water storage. The CFD simulation is also carried out to evaluate the water temperature.

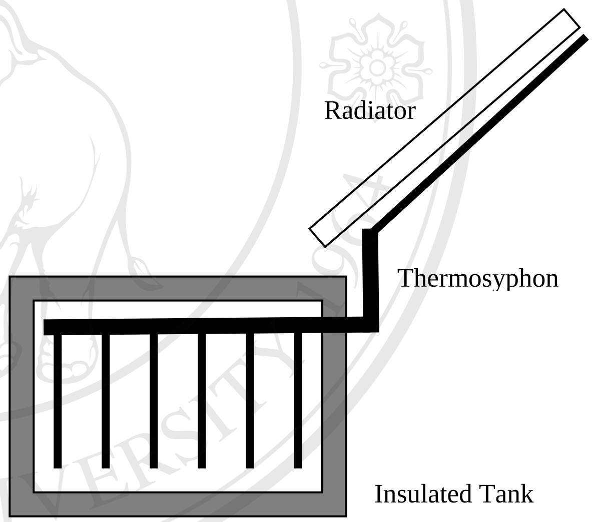

Figure1. Schematic drawing of night sky cooling system by using a thermosyphon radiator.

2. Two-Phase Closed Thermosyphon Heat Pipe

The closed two-phase thermosyphons is an effective heat transfer device that obtains its heat from the evaporator section by means of the evaporating mechanism and, then releasing the heat out at the condenser section by means of the condensing phenomenon. Since the latent heat of vaporization of the working fluid is relatively high, a large amount of heat can be transported through the thermosyphon. The thermosyphon can mainly operate under the assistance of the gravity; as shown in Figure 2. The unit can be separated into 3 parts as evaporator, adiabatic and condenser sections.

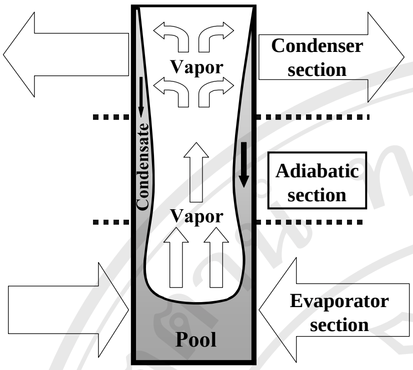

Figure2. Thermosyphon [2].

The actual overall rate of heat transfer, Q is then related by:

$$Q = \frac{\Delta T}{Z_{total}}, \qquad (1)$$

where Q is the rejected heat (W),

ΔT is the effective temperature difference between the heat source and the heat sink (°C):

$$\Delta T = T_{so} - T_{si} - \Delta T_h \quad , \qquad (2)$$

where Tso is the heat source temperature (°C),

Tsi is the heat sink temperature (°C),

ΔT_h is the mean temperature difference due to hydrostatic head (°C).

where Z_{total} is the overall thermal resistance of the thermosyphons can be represented by the idealized network of thermal resistances Z_1 to Z_{10} as shown in Figure 3.

where Z_1 and Z_9 are the thermal resistance between the heat source and the evaporator and the condenser external surface and the heat sink respectively

Z_2 and Z_8 are the thermal resistances across the thickness of the contains wall in the evaporative and the condenser respectively

Z_3 and Z_7 are the internal resistances due to pool and film boiling of the working fluid

Z_{3p} is the resistance from pool boiling

Z_{3f} is the resistance from film boiling at the evaporator section

Z_4 and Z_6 are the thermal resistance those occur at the vapor liquid interface in the evaporator and the condenser respectively

Z_5 is the effective thermal resistance due to the pressure drop of the vapor as it flows from the evaporative to the condenser

Z_{10} is the axial thermal resistance of the wall of the container.

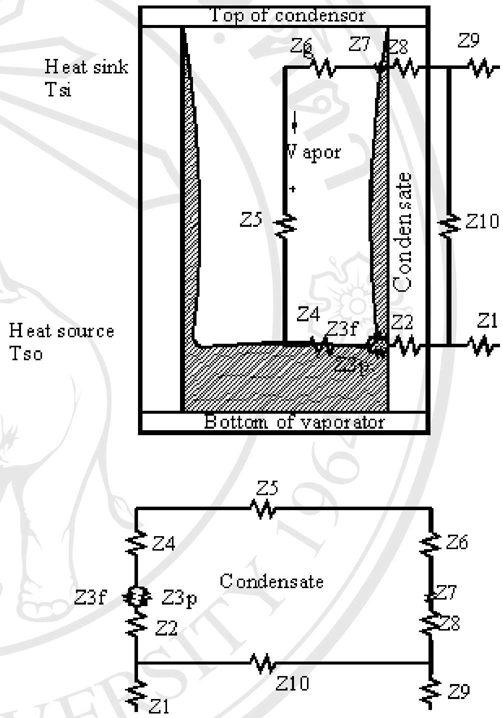

Figure3. Thermal resistance and their locations.

The details and calculated methods of all Z are shown in all available designed text books [3].

3. Experimental Setup

The experimental sets were based on three vessel of 3.5 L. of water, containing in a well-insulated vessel. In this experimental study, three copper thermosyphon heat pipes having R-134a inside with 19.05 mm. diameter and 1.00, 0.85, 0.70 m. length were used to extract heat from water in a water storage. The condenser length (L_c) and the adiabatic length (L_a) of each thermosyphon heat pipe were at 0.45 and 0.10 m. respectively, when the evaporative length (L_e) is varied. The L_e/L_c ratios of three thermosyphon heat pipe were 1.00, 0.67 and 0.33 respectively. The initial water temperature in each vessel was at 45 °C. At the condenser part, there was cooling water, maintained at 10 °C by a cold bath, as shown in Figure 4.

Fluid and surface temperature were measured by IC sensors, calibrated individually so that the differential error of measurement was less than 0.5 °C. All data were transferred to a 16 channel data-logger and collected every 10 minutes. The fluid and the surface temperatures were multi-pointed measurement and used the average value for calculation.

The water temperature data from an experiment were used to calculate the rejected heat by the thermosyphon heat pipe by:

$$Q = \frac{mC_p(T_{w,t} - T_{s,t+\Delta t})}{\Delta t}, \quad (3)$$

where Q is the rejected heat (W),

m is the mass of water in study time step (kg),

Cp is the heat capacity of water (4179 kJ/kg K),

Tw is the water temperature (°C),

Δt is the time interval (s).

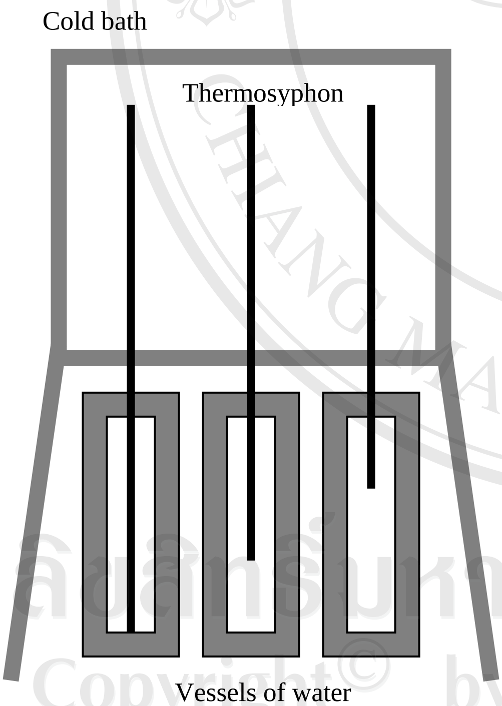

Figure4. Schematic drawing of the experimental setup.

The rejected heat was used to calculate the heat transfer coefficients around each thermosyphon heat pipe as:

$$h = \frac{Q}{A(T_w - T_s)}, \quad (4)$$

where h is the heat transfer coefficients (W/ m^2 K),

A is thermosyphon heat pipe surface area (m^2),

T_s is the thermosyphon surface temperature (°C).

4. Computation Fluid Dynamics (CFD)

A cell-centered control volume solution approach was deployed. This approach implies that the discrete equations are formulated by evaluating and integrating the fluxes across the faces that surround each control volume. In addition, this calculated system uses a pressure-based methodology in which the pressure becomes one of the dependent equations evaluated at each cell. This method allows the solutions for incompressible problems where pressure is loosely coupled to density.

The computation domains were the water in each 0.10 m. width x 0.45 m. height vessels, which were divide into 16 x 45 grids. The adiabatic boundary conditions are set around the computation domains, when the convective boundary conditions are set at the thermosyphon surfaces, using the average heat transfer coefficients from each experiment as the input data in the CFD simulation. The calculated outputs were vector flow fields and also fluid temperature at each time steps.

5. Result and Discussion

The experimental results of water temperature and rejected heat between thermosyphon heat pipe, having L_e/L_c = 1.00, 0.67 and 0.33 are shown in Figures 5 and 6, respectively. Due to the larger evaporative part surface area, the L_e/L_c = 1.00 thermosyphon heat pipe unit can reject heat from the system more than the L_e/L_c = 0.67 and L_e/L_c = 0.33 uint by 14% and 45%, respectively. This condition also gives the lowest water temperature.

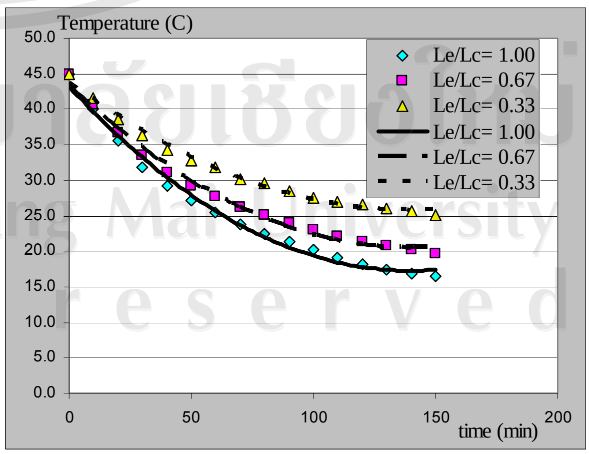

Figure5. The experimental result of water temperature between L_e/L_c = 1.00, 0.67 and 0.33.

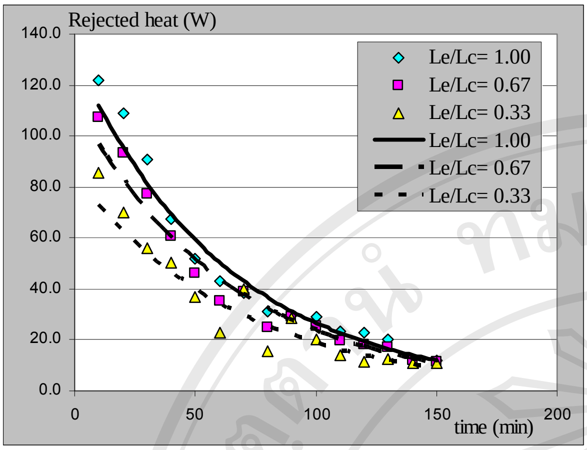

Figure6. The experimental result of rejected heat between L_e/L_c = 1.00, 0.67 and 0.33.

Figures 7, 8 and 9 show the water temperatures from the CFD that agree very well with the experimental results within 8% errors, with the 3.1% average error. But when comparing the experimental and the theoretical study, the average error was 12.8%. Both maximum error occurred when L_e/L_c = 0.33.

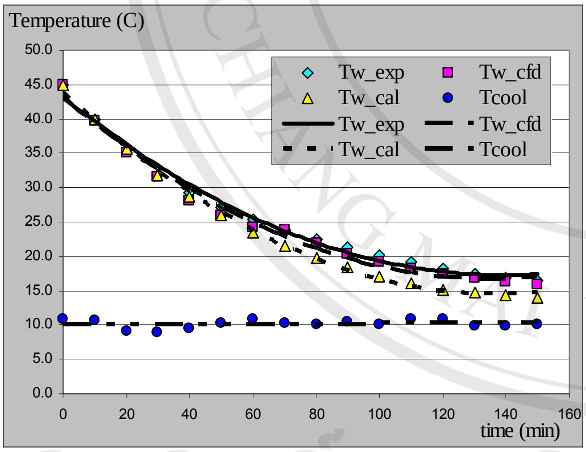

Figure7. Water temperature of the experiment, CFD and theoretical study when L_e/L_c = 1.00.

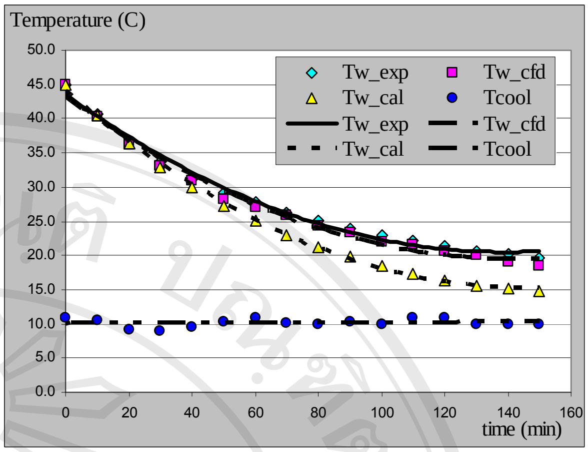

Figure8. Water temperature of the experiment, CFD and theoretical study when L_e/L_c = 0.67.

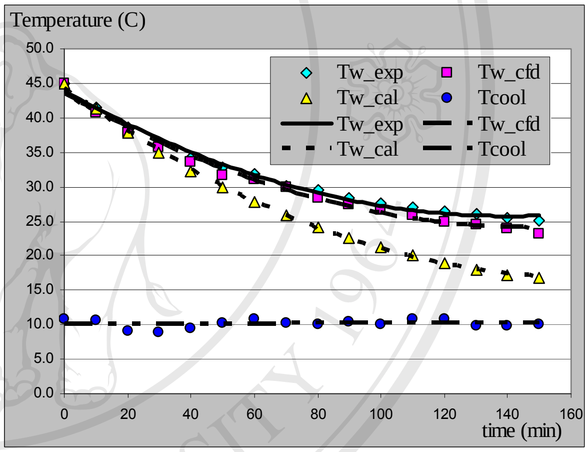

Figure9. Water temperature of the experiment, CFD and theoretical study when L_e/L_c = 0.33.

The rejected heat in the experiment and CFD simulation can be found after knowing the water temperature data at each time steps by using equation (3). As shown in Figures 10, 11 and 12, the experimental and CFD simulated result have 10.5% average error. But when comparing the experimental and theoretical study, the average error was 30.9%. Both maximum error occurred when L_e/L_c = 0.33.

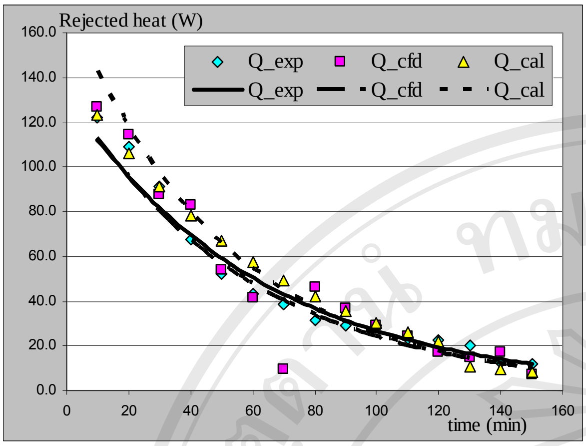

Figure10. Rejected heat of the experiment, CFD simulation and theoretical study when $L_e/L_c = 1.00$.

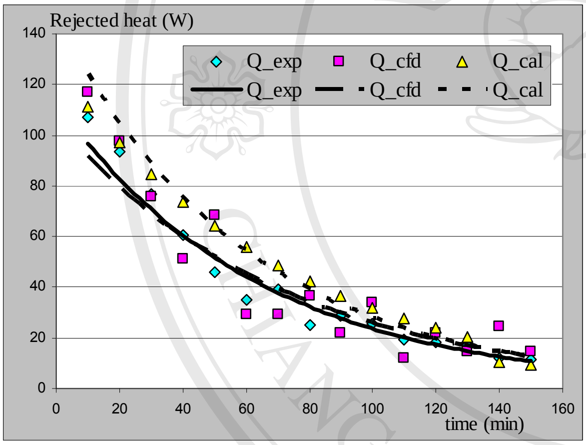

Figure11. Rejected heat of the experiment, CFD simulation and theoretical study when $L_e/L_c = 0.67$.

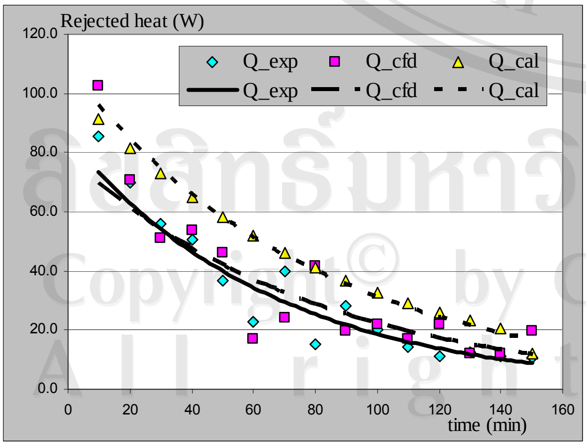

Figure12. Rejected heat of the experiment, CFD simulation and theoretical study when $L_e/L_c = 0.33$.

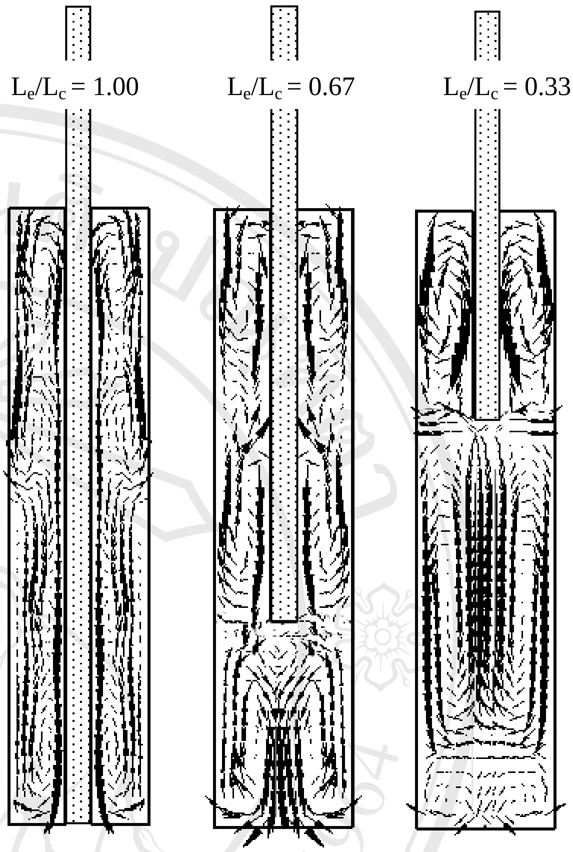

Figure13. The vector flow fields at time = 60 minutes.

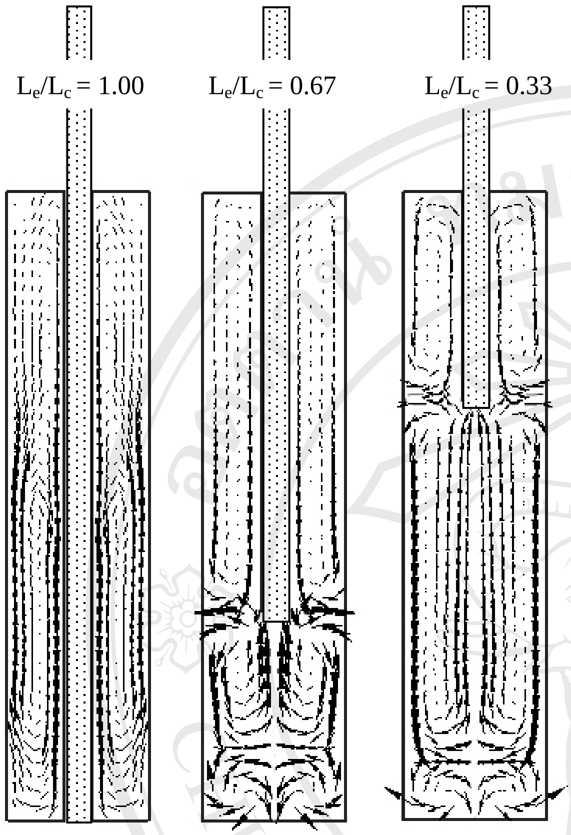

Figure14. The vector flow fields at time = 300 minutes.

The resulted error in each cased study can be explained by the CFD simulation. Figures 13 and 14 show the vector flow fields of all case studies at time 60 and 300 minutes, respectively. When L_e/L_c = 1.00, the vector flow fields around the thermosyphon heat pipe was one loop circulation, that the water in the vessel could flow throughout the vessel. This factor leads to high accuracy when using measured water temperature at each water level to calculate the average heat transfer coefficients to be the input data in the CFD simulation and theoretical study. When L_e/L_c = 0.67 and 0.33, the vector flow fields around the thermosyphon heat pipe were clearly divided into three regions. This unsatisfied circulation leading to an error when using measured water temperature in the studies, cause of the temperature different in each loop. As mention above, using the average heat transfer coefficients from an experiment as the input in both CFD simulation and theoretical study give the highest accuracy in the case L_e/L_c = 1.00, when be lower L_e/L_c will be lower in an accuracy.

6. Conclusion

This study has shown the effect of the evaporative length on the heat transfer of thermosyphon heat pipe. Due to the larger evaporative part surface area, the L_e/L_c = 1.00 thermosyphon heat pipe unit can reject heat from the system more than the L_e/L_c = 0.67 and L_e/L_c = 0.33 uint by 14% and 45%, respectively. This condition also gives the lowest water temperature.

Moreover, this study has also shown the significant of using average heat transfer coefficients from the experiment as the input in the CFD simulation and theoretical study to find water temperature and rejected heat in the un-uniformed heat source temperature situation. When comparing with the experiment, the resulted error from the theoretical study seems to be much higher than that from the CFD simulation. The maximum error always occurs in the case study when L_e/L_c = 0.33. The vector flow fields and water temperature from CFD simulation were used to explain the phenomenon.

Acknowledgements

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Experimental Study on Nocturnal Cooling System Using Thermosyphon Heat Pipe Radiator

Chotivisarut N. and Kiatsiriroat T.

Department of Mechanical Engineering, Faculty of Engineering, Chiang Mai University,
Chiang Mai, 50200 Thailand, Tel. +66 53 944144 ext. 431
(nammont4426409@hotmail.com tanong@dome.eng.cmu.ac.th)

ABSTRACT

The objective of this paper is to study a nocturnal cooling potential in a well-insulated room having dimensions of 3 m x 3 m x 2.5 m by using thermosyphon heat pipe as a thermal radiator. The radiator consists of 48 thermosyphon heat pipe tubes each of 19.05 mm in diameter and it rejects heat to the sky for producing cool water in a 1.0 m^3 insulated rectangular tank during the nighttime. The cooled water is filled in a set of six heat exchangers each of 0.87 m^2 in surface area installed at the room ceiling then absorbs heat and reduces the temperature inside the tested room during the daytime. The experimental site is located in Chiang Mai, Thailand. The thermal load inside the room is performed by an electrical heater. The initial water temperature in the storage tank at 27 ºC could be gradually cooled down to12.1 ºC within 4 nights. The cooled water is fed into the room when the electrical heater supplied heat at 1000 W. From the experiment, it could be found that the cooled water absorbed heat and the indoor temperature was decreased around 12.8 ºC or 21.8 % compared with that without water cooling. Energy balance has shown only 3.34 % error between calculated value from thermosyphon theory and that from experimental result. This system could save energy consumption and should be expanded to commercial scale.

Keywords nocturnal, cooling, thermosyphon, radiator, heat pipe, cooled water.

1. INTRODUCTION

One of the passive cooling methods is night sky cooling technique which has been studied by many researchers. Vimolrat and Kiatsiriroat [1] proposed a new concept of nocturnal cooling by using a thermosyphon-radiator for producing cool water. The system is shown in Fig. 1. The evaporation part of the thermosyphon is dipped in a water tank and the condenser part is exposed to the ambient air. During the nighttime, the condenser is cooled by convection with the surrounding air and by radiation to the sky. Then the water temperature in the tank could be reduced by transferring heat to the evaporator of the thermosyphon. The cool water could be fed into building zone during the daytime when the cooling is needed. Therefore, the air conditioning load by a conventional unit could be reduced.

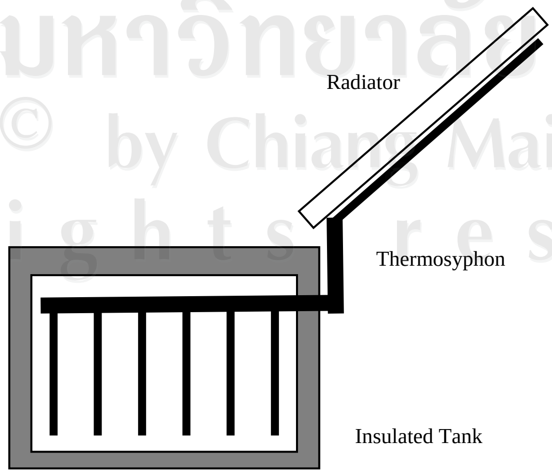

Fig. 1 Schematic drawing of a night sky water cooling system having a thermosyphon radiator [1].

The net radiative heat transfer at the surface of the radiator to the night sky can be expressed as

$$R_{net} = 4\varepsilon\sigma T_{air}^{3}\left(T_{rad} - T_{sky}\right), \tag{1}$$

R_{net} is the net radiative heat loss (W/m^2), ε is the emissivity of the radiator surface, σ is Stefan-Boltzmann constant where σ =5.67 x 10^{-8} W/m^2 K^4, T_{air} is the ambient air temperature (K), T_{rad} is the radiator temperature (K) and T_{sky} is the equivalent sky temperature (K).

This work is to extend the study of Vimolrat and Kiatsiriroat by investigated the potential of using the cool water obtained in the nighttime to reduce temperature inside a living room in the daytime. Thus this concept could be applied in any air-conditioned building for energy saving in air conditioning system.

2. CHARACTERISTICS OF CLOSED END TWO-PHASE THERMOSYPHON HEAT PIPE

Closed end two-phase thermosyphon heat pipe is an effective heat transfer device that obtains heat from the evaporator section by means of the evaporating mechanism and, then releasing the heat out at the condenser section by means of the condensing phenomenon. Since the latent heat of vaporization of the working fluid is relatively high, a large amount of heat can be transported through the heat pipe. The unit can mainly operate under the assistance of the gravity as shown in Fig. 2.

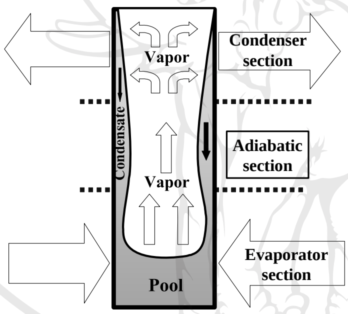

Fig. 2 Thermosyphon [2].

The actual overall rate of heat transfer, Q is calculated by

$$Q = \frac{\Delta T}{Z_{total}}, \tag{2}$$

Q is the rejected heat rate (W) and ΔT is the effective temperature difference between the heat source and the heat sink (°C) which could be calculated by

$$\Delta T = T_{so} - T_{si} - \Delta T_{h}, \tag{3}$$

T_{so} is the heat source temperature (°C), T_{si} is the heat sink temperature (°C), ΔT_h is the mean temperature difference due to hydrostatic head (°C) and Z_{total} is the overall thermal resistance of the heat pipe.

An idealized network of thermal resistances Z_1 to Z_{10} is shown in Fig. 3. Z_1 and Z_9 are thermal resistance between the heat source and the evaporator and the condenser external surface and the heat sink respectively. Z_2 and Z_8 are thermal resistances across the thickness of the wall in the evaporator and the condenser respectively. Z_3 and Z_7 are internal resistances due to pool and film boiling of the working fluid. Z_{3p} is the resistance from pool boiling. Z_{3f} is the resistance from film boiling at the evaporator section. Z_4 and Z_6 are thermal resistances those occur at the vapor liquid interface in the evaporator and the condenser, respectively. Z_5 is effective thermal resistance due to the pressure drop of the vapor as it flows from the evaporator to the condenser. Z_{10} is axial thermal resistance of the container wall.

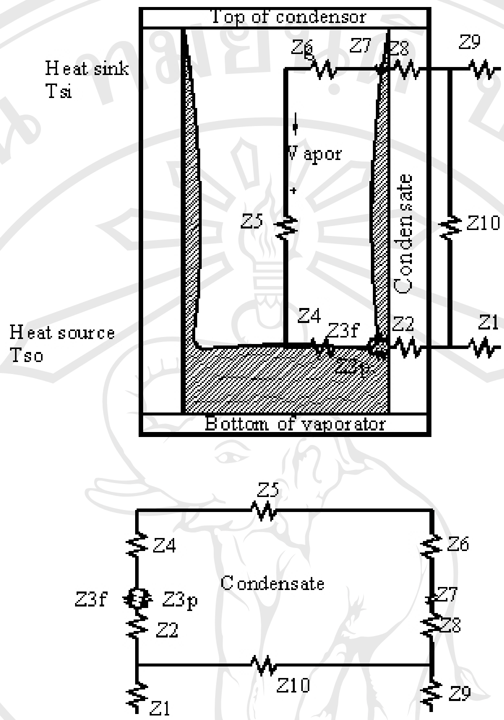

Fig. 3 Thermal resistances.

The details of the calculating methods of all resistances are shown in [3].

3. THE TESTED ROOM

The schematic drawing of the tested room is shown in Fig 4.

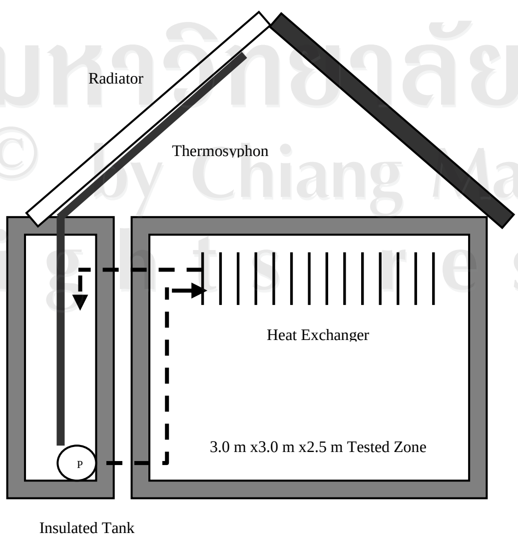

Fig. 4 Schematic drawing of the tested room.

The room was well-insulated with dimensions of 3 m x 3 m x 2.5 m. Its walls and ceiling are10.0 mm. gypsum boards and 25.4 mm polyethylene sheets were taken inside as the room insulation. The 1 m^3 insulated rectangular water storage tank was made of 1 mm galvanized steel with 304.8 mm polyethylene insulation. The two-sided 45 degree tilting roof was made of galvanized metal sheets each of 9 m^2. The thermosyphon heat pipes were made of 48 copper tubes each of 19.05 mm in diameter. The evaporative and condenser parts had the same length of 1.5 m whereas its 0.2 m adiabatic part was well-insulated. The evaporative length was design for maximum rate of heat transfer [4]. Each condenser part of thermosyphon heat pipes was well-attached with the metal roof. The tested room is shown in Fig. 5. In the room there are six heat exchanger units each of 0.87 m^2 in area installed at the ceiling as shown in Fig. 6.

Fig. 5 The tested room.

Fig. 6 Heat exchangers installed at the ceiling.

4. EXPERIMENTAL PROCEDURE

As mention above, production of cold water in the nighttime and use it to reduce room temperature in the daytime is the main purpose of this study. The experiment was done in a winter during December 2005-February 2006. The night-time ambient temperature was always below 15 ºC but the daytime ambient temperature was still high. The maximum temperature difference during a day might be over 20 ºC. The sky was quite clear thus high radiative heat transfer form the radiator surface to the sky could be obtained.

In the tested room, a 1000 W heater with a current dimmer was installed as an artificial load.

The data of dry-bulb and wet-bulb temperatures were recorded for evaluating dew point temperature and relative humidity. Wind speed data were collected by a three cup anemometer. The solar insolation was measured by a pyranometer.

Three points of indoor temperature, four points of night-sky radiator surface, five points of water temperature in tank and the ambient wet bulb and dry bulb temperatures were collected, including the inlet and outlet water temperatures by a set of calibrated IC sensors of which the differential error of measurement was always less than 0.5 °C. All data were transferred to a 16 channel data-logger and collected every 10 minutes.

The water temperature data from each time interval were used to calculate the rejected heat by the thermosyphon heat pipe by

$$Q = \frac{mC_p(T_{w,t} - T_{w,t+\Delta t})}{\Delta t} \tag{4}$$

Q is the rejected heat (W), m is the mass of water (kg), Cp is the heat capacity of water where Cp=4179 kJ/kg K, T_w is the water temperature (°C) at time t and Δt is the time interval (s).

In the daytime a 75 W submersible pump was used to circulate cool water from the tank into the tested room. The mass flow rate of the cool water in the experiment was at 0.0167 kg/s.

5. EXPERIMENTAL RESULTS

The experiment results of the cooled water production were shown in Fig. 7. Under the clear sky condition, when the initial water temperature started at 27 ºC, the water temperature in the storage tank was gradually decreased to 12.1 ºC which was 1 ºC below the lowest ambient temperature within 4 nights. The insulation around tank seems to work well that could protect heat gain from the surrounding between the daytime. After 4th night, the water temperature seemed to be constant and might not decrease more.

The comparison of the tested room temperature during the cooling operation and no cooling under the 1000 W thermal load of the electrical heater was shown in Fig. 8. The indoor temperature of the tested zone was decreased around 12.8 ºC or 21.8 % of that without cooling water condition. This result shows the potential of the cooling system in cooling load reduction and energy saving in an active cooling operation of an air-conditioned building.

The comparison of rejected heat between theoretical study of thermosyphon heat pipe and calculated value from experimental result by using Eq. (4) has shown in Fig. 9. The error between calculated value form thermosyphon theory and that from experimental result is only about 3.34 %. So, using thermosyphon heat pipe theory could be predicted the heat transfer and this may lead to water temperature prediction.

6. CONCLUSIONS

Experimental study of the nocturnal cooling system of a tested room by using thermosyphon heat pipe radiator, at Chiang Mai, Thailand, was carried out. With the initial water temperature of 27 ºC, the system could produce the cool water to 12.1 ºC which was 1.0 ºC below the lowest ambient temperature within 4 nights. For cooling operation during a daytime, under 1000 W thermal load, the indoor temperature of the tested room, with water cooling, could be decreased around 12.8 ºC or 21.8 % from that without the water cooling. Energy balance has shown the 3.34 % error between thermosyphon theoretical study and calculated value from experimental result. As mentioned above, this system could save energy consumption and should be expanded to commercial scale. Underground seasonal storage may be applied to work couple with this system for advanced operation.

ACKNOWLEDGEMENTS

This research is financially supported by the Royal Golden Jubilee, Thailand Research Fund, contract no. PHD/ 0273/ 2545.

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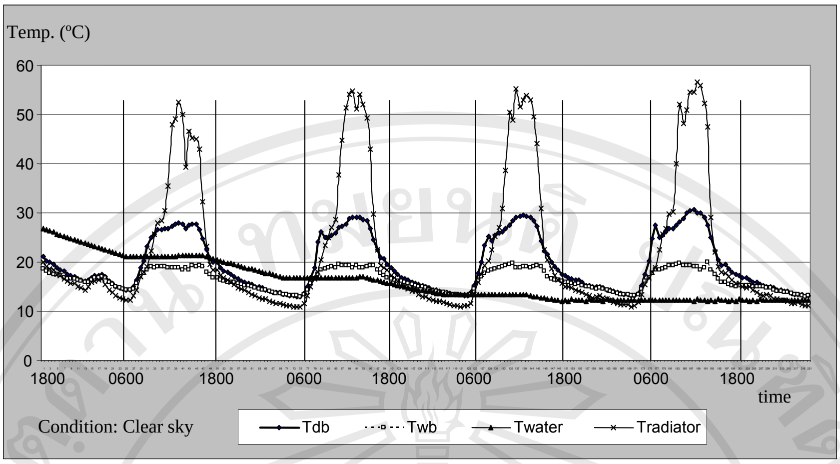

Fig. 7 Cooled water production, 26 – 31 January, 2006.

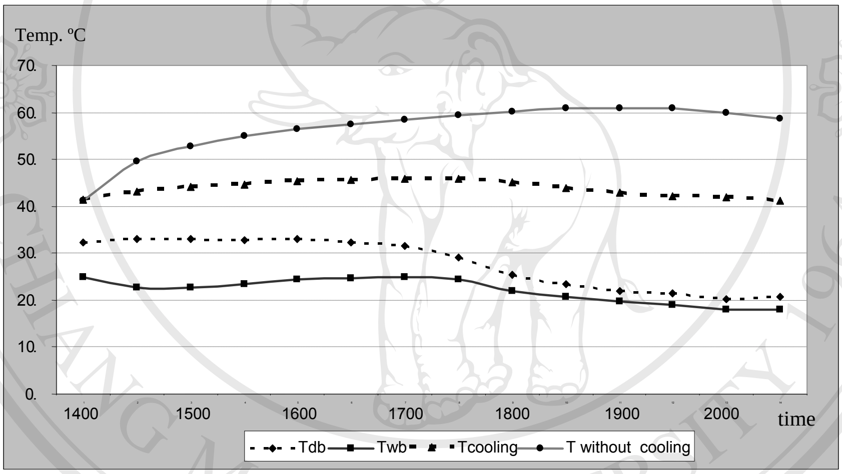

Fig. 8 Comparison of the tested room temperatures with and without water cooling. A 1000 W heat source is used as an artificial thermal load.

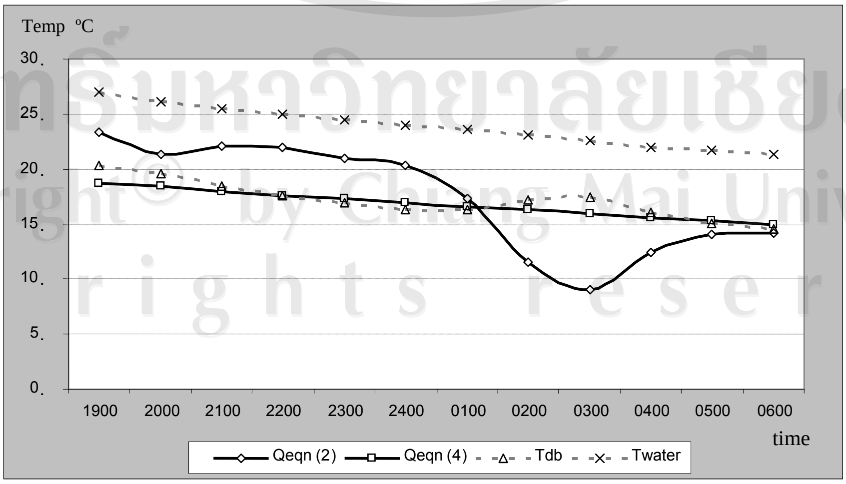

Fig. 9 Rejected heat rate calculated from thermosyphon heat pipe theory, Eq. (2), compared with that calculated from Eq. (4).

Chotivisarut N. and Kiatsiriroat T. (2007), *Design of Central Solar Heating with Underground Seasonal Storage in Australia*, The 6th Conference on Energy, Heat and Mass Transfer in Thermal Equipments, 15-16 March 2007, Chiang Mai, Thailand.

DESIGN OF CENTRAL SOLAR HEATING WITH UNDERGROUND SEASONAL STORAGE IN AUSTRALIA

N. Chotivisarut
T. Kiatsiriroat
Department of Mechanical Engineering, Chiang Mai University, Thailand

L. Aye
International Technologies Centre, Department of Civil and Environmental Engineering, The University of Melbourne, Australia

Abstract

The simulation of the central solar heating system with seasonal storage has done by using TRNSYS to predict thermal performances and economic aspects. The location of the simulated site is at Glenroy, Melbourne, a southern part of Australia (latitude: 37º 42´ S, longitude: 144º 5´ E, altitude: 97 m.). The meteorological conditions data are prepared by Meteonorm, based on 10 years of measurements. The central solar heating system consist of solar collector, a well-insulated underground seasonal thermal storage tank with heater, fan coil unit that installed in the heating zone and two single speed pump. In the daytime, water from the underground storage tank will feed pass through the solar collector, collecting energy and return to underground storage tank, that will keep the water temperature gradually goes up. In the winter time, and when the indoor house temperature is below 21 ºC, the warm water from the underground storage tank will feed into the house. The heating coil, install in the house, will work as heat exchanger so that energy from water could transfer to an air space in the house to keep the indoor temperature above 21 ºC and the residents may feel comfortable. The sizing of the underground storage tank and also the auxiliary heater capacity are limited by its initial cost and availability, so it will be set as selected parameters. After that, the area of solar collector, initial water temperature would selected by the simulation. The simulation result has shown that, the underground storage tank suitable for this application is around 50 m^3 with 5m^2 solar collector area. The initial water temperature in the staring of January should be at 50 ºC. The solar factor for this application is 0.76. The economical analysis has shown that the Internal Rate of Return (IRR) of this investment when compare with widely used natural gas heater is around 8.5% that seems to be very economically and favorably.

Keywords: solar heating, underground storage, seasonal storage

1. INTRODUCTION

The main energy consumption in Australia in the winter time is the residential heating. When we take a look at Australia we can detect certain trends that relate to the consumption of energy. According to data from the World Resources Institute, the total commercial energy consumed in 1993 was 3.92 EJ. This was a major increase of 99% from 1973. We can also look at the total energy consumed in 1993 and the projected energy consumed in 2014. If the projected data are correct there will be a 55% (5.698 EJ) in the energy consumed from 1993 to 2014. Recent energy consumption has been on the order of 1 EJ per year. It is expected to reach nearly 10 EJ per year within the next 15 years. Reduction of energy consumption was promoted by Australian Government for a long time. One of the most effective solutions is to use the alternative energy. Solar energy, an abundant, clean and safe source, is an attractive substitute for conventional fuels for passive and active heating applications. During the day, excess solar heat is collected for short or long term storage, and it is recovered at night in order to satisfy the heating needs of greenhouses. Efficient and economical heat storage is the main factor in utilization of solar energy for daily used purposes. Solar thermal energy seasonal storage system is the appropriate alternative way for Australia cause of the meteorological conditions.

Thermal storage systems in buildings act as energy buffers which balance the periodic thermal demands and supplies. Systems can be designed to cope with the diurnal (24 hours), intermediate (several days) or even seasonal thermal energy fluctuations. Thermal energy storage is made practical by the large heat capacity of water. One metric ton of water, just one cubic meter, can store 334 MJ. Appropriate applications of thermal storage systems can significantly reduce the thermal energy consumption as well as improve the comfort conditions of the indoor environment.

In Germany, since 1995, eight central solar heating plants with seasonal heat storage have been built within the governmental Research and Development program “Solarthermie-2000” (Schmidt et al., 2004). In Turkey, annual periodic performance of a solar assisted ground-coupled heat pump space heating system with seasonal energy storage in a hemispherical surface tank is investigated using analytical and computational methods (Ramyutus and Unsal, 2000). Moreover, thermal performance and economic feasibility of central solar heating system with seasonal storage under four climatically different Turkey locations are studied based on a finite element analysis and finite element code ANSYS™ (Ucar and Inalli, 2005). As in Korea, a central solar heating plant with seasonal storage is simulated using TRNSYS to predict thermal performances and economic aspects (Chung el al.,1998). As mentioned above, central solar heating system with seasonal heat storage has studied by many researchers in many countries. This work shows the feasibility and suitableness of this system in Australia.

2. SYSTEM LAYOUT AND CONTROL

The central solar heating system that studied was shown in Fig 1. The system consist a set of solar collector, a well-insulated underground seasonal thermal storage tank with heater, fan coil unit that installed in the heating zone and two single speed pump.

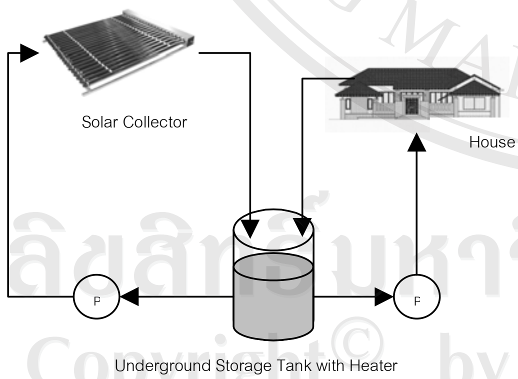

Fig 1. Layout of the central solar heating system with cylindrical seasonal thermal storage.

In the daytime, 8.00 am – 4.00 pm daily, the controller set would operate a single speed pump, and then the water from the underground storage tank will feed pass through the solar collector, collecting energy and then return to underground storage tank. This recharge process not only maintains the energy of the system but also keeps the water temperature gradually goes up. In the winter time, around middle of May to the middle of August, and when the house temperature is below 21 ºC, the controller will operate another single speed pump, and then the warm water from the underground storage tank will feed into the house. The heating coil, install in the house, will work as heat exchanger so that energy from water could transfer to an air space in the house to keep the indoor temperature above 21 ºC and the residents may feel comfortable (recommended by ASHRAE standard).

3. METHODOLOGY & SIMULATION TOOL

The central solar system with seasonal storage tank were setup as a model in simulation program TRNSYS, a well-known and widely used system simulation program of transient thermal process. This program has high ability to develop the central solar heating system. It provides libraries of subroutines such as solar collectors, water tanks, pumps, pipes, valves and controller. Users can write their own subroutine components to respond their particular need. The system layout (as shown in Fig 1.) is reproduced by assembling the corresponding the TRNSYS components. The simulation time step should be short enough to reproduce the variation of the thermal process, 1 hour for each time step was found to be suitable for this simulation. The system design was defined by all the parameters for each component. Initial and boundary conditions are determined respectively by the initial values of the component variables and by the dynamics of meteorological conditions. The components, selected in this study, are listed as follows:

- The TYPE1b quadratic efficiency solar thermal collectors: This component models the thermal performance of a flat-plate solar collector. The solar collector array may consist of collectors connected in series and in parallel.
- The TYPE 38 plug-flow storage tank: This component models the behavior of a temperature stratified storage tank using variable size segments of fluid. The size of segments is governed by the simulation time step, the magnitude of collector and load flow rates, heat losses and auxiliary input. The main advantage over fixed node simulation techniques is that temperature stratification can be modeled with small segments in the temperature gradient zone without the need to use small simulation time steps to obtain a good solution.
- The TYPE 52b cooling and heating coil rectangular fin: This component models the performance of a cooling and heating coil using the effectiveness model outlined by Braun. The user must specify the geometry of the cooling and heating coil and air duct. The continuous flat plate fins are specified. Either a simple or detailed level of analysis may be chosen by the user. The level of detail determines the method used in modeling a coil operating under partially wet and dry conditions.

- The TYPE 3b single speed pump: This pump model computes a mass flow rate using a variable control function and a fixed maximum flow capacity. Pump power may also be calculated, either as a linear function of mass flow rate or by a user-defined relationship between mass flow rate and power consumption. A user-specified portion of the pump power is converted to fluid thermal energy.
- The TYPE 14h forcing functions: In a transient simulation, it is sometimes convenient to employ a time dependent forcing function which has a behavior characterized by a repeated pattern. The pattern of the forcing function is established by a set of discrete data points indicating the value of the function at various times throughout one cycle.

The reliability of all selected components was well-tested by TRNSYS and global user.

Normally, the sizing of the underground storage tank will be limited by its initial cost that should not be over than 50 m^3 for 4 people family. The auxiliary heater capacity, easy available, should not be over than 5000 W. After that, the area of solar collector, initial water temperature will be selected by the simulation.

The main thermal characteristics of the studied system were shown in Table 1.

Table 1. Main thermal characteristics of main components of the system.

Solar Thermal Collector		
area (series)	5	$m.^2$
orientation	North	
incident angle	45	degree
specific flow rate	100	kg./hr.
Plug-flow underground storage tank		
tank volume	50	$m.^3$
thermal conductivity	1.5	kJ./(hr.m.K.)
overall heat loss coef.	5	kJ./(hr. K.)
max. heating rate	5	kW.
auxiliary thermostat temp.	50	ºC
Heating coil rectangular fin		
specific flow rate	50	kg./hr.
no. of HX row	7	
no. of tube in each row	4	
outside tube diameter	0.025	m.
inside tube diameter	0.020	m.
tube spacing	0.350	m.
tube thermal conductivity	500	kJ./(hr.m.K.)
fin thickness	0.010	m.
fin spacing	0.100	m.
number of fin in coil	25	
fin thermal conductivity	700	kJ./(hr.m.K.)

Energy in and out from each components of the system was calculated both instantaneous and summary values to check the energy balance.

4. WEATHER CONDITION AND HEAT LOAD

The location of the simulated site is at Glenroy, Melbourne, Australia (latitude: 37º 42´ S, longitude: 144º 5´ E, altitude: 97 m.). The meteorological conditions data are prepared by Meteonorm, based on 10 years of measurements. Hourly values of the meteorological and radiation data are used in the simulation.

The annual heat load is the annual energy used of the 148 m^2 single floor house. The number of people lived in this house was set to 4 people with normally Australian activities. Some heat generated equipments such as lights and computers were studied. Infiltration of both living and attic zones were also set as inputs too.

5. SIMULATION RESULTS

From the simulation, the initial temperature from the starting of January that suitable to this study is 50.0 ºC, with solar collector area 5.0 m^2. Reheating the water via solar collector in the daytime every day can rise up its temperature to 56.5 ºC in the middle of May. In the winter time, whenever the indoor house temperature is below than 21.0 ºC, the heating system will be operated to keep the indoor temperature above 21.0 ºC. This will make the water temperature goes down to 40.6 ºC. The auxiliary heater in storage tank, controlled by its thermostat, will start automatically, couple with reheating the water via solar collector every daytime. After this period, the water will reheat without heating the house, so that its temperature can raise up to 50.0 ºC in the end of year. Fig 2. has shown the simulation result of water temperature in underground storage tank, indoor house temperature and auxiliary energy consumption in the winter time for the selected parameters. The simulation values of the energy for each component have shown in Table 2.

Table 2. The simulation values of the energy for each component (units in GJ).

Whole year house heating load required	14.04
Energy supplied to the tank from the collector	15.46
Auxiliary energy from heater	1.81
Overall tank losses	0.90
Energy store in the tank	0.78
Energy supplied to the house from the tank	10.64

The energy balanced of the system should be express as:

Energy supplied to the tank from the collector + Auxiliary energy from heater = Tank losses + Energy store in the tank + Heating load ------------------- (1)

By substitute the simulation value from Table 2. in Equation (1)., the energy balanced has shown the whole year 8.9% error. This error should be an effect of accumulation error in each interval time step from each calculation.

The solar fraction of the system should be express as:

Solar fraction = Energy supplied to the house from the tank / Heating load --------------------------------------- (2)

Varying the solar collector area was done in this study, which the effect of changing solar collector area on solar fraction has shown in Fig 3. The solar fraction of the selected collector area, 5 m^2, is around 0.76.

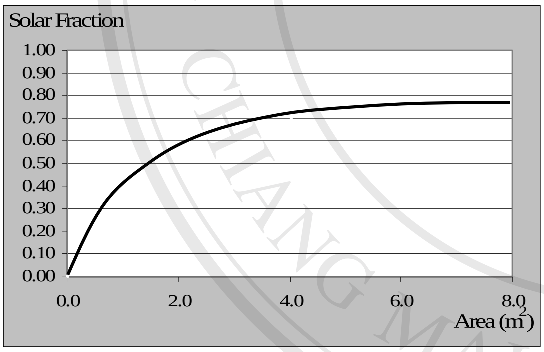

Fig 3. The optimization of solar fraction and solar collector area.

The energy that can be saved by this central solar heating system should be express as:

Saved energy = Energy supplied to the house from the tank - Auxiliary energy from heater ------------------- (3)

By substitute the simulation value from Table 2., the energy that can be saved by this system is around 8.83 GJ per year. This can lead to be the saving for natural gas fuel cost that normally used as energy source of Australian housed heater around 80 AUS$ a year.

6. ECONOMICAL ANALYSYS

The Internal Rate of Return (IRR) is the index in this economical analysis. It is the discount rate that results in a net present value of zero for a series of future cash flows. It is one of the Discount Cash Flow (DCF) approach to valuation and investing just as Net Present Value (NPV). Both IRR and NPV are widely used to decide which investments to undertake and which investments not to make. NPV show the stream of future cash flow discounted back to the present by some percentage that represents the minimum desired rate of return. The formulae to find IRR should be express as:

$$NPV = \sum_{n=1}^{N}\left(\frac{NCF_n}{(1+i)^n}\right) - TIC = 0 \quad \text{---------} (4)$$

or

$$TIC = \sum_{n=1}^{N}\left(\frac{NCF_n}{(1+i)^n}\right) \quad \text{-----------} (5)$$

where TIC is the initial investment at the present value (AUS$),

i is the internal rate of return,

N is the number of years for operating life of the system (years),

NCF_n is the net cash flow at the year nth that be energy cost saving compare with the natural gas heating system (AUS$).

The estimation this investment when compared with natural gas heater system could be found by substitute the economical value in the Equation (4) or (5). The discount rate of the progress was set to be 9% a year (7% for MLR interest and 2% for inflation in Australia) when the cash flow reduce 1% a year for maintenance cost. With trial and error the values of internal rate of return (IRR) until getting the net present value (NPV) of zero. An equation will give us that the internal rate of return (IRR) of 8.5%.

7. CONCLUSIONS

The simulation result has shown that, from the selected parameters: underground storage tank volume 50 m^3, auxiliary heating energy 5000 W., the solar collector area suitable for this application is 5 m^2. The initial water temperature in the staring of January should be at 50.0 ºC which will increase up to 56.5 ºC when recharge daily via solar collector (without house heating load). In the winter time, warm water from underground storage tank will be used in house heating application that its temperature will drop to around 40.6 ºC. The water temperature will be warm up back to around 50.0 ºC at the end of the year. The solar factor for this application will be 0.76. The economical analysis has shown that the Internal Rate of Return (IRR) of this investment when compare with widely used natural gas heater is around 8.5% that seems to be very economical and favorable to use in the single house family in Australia.

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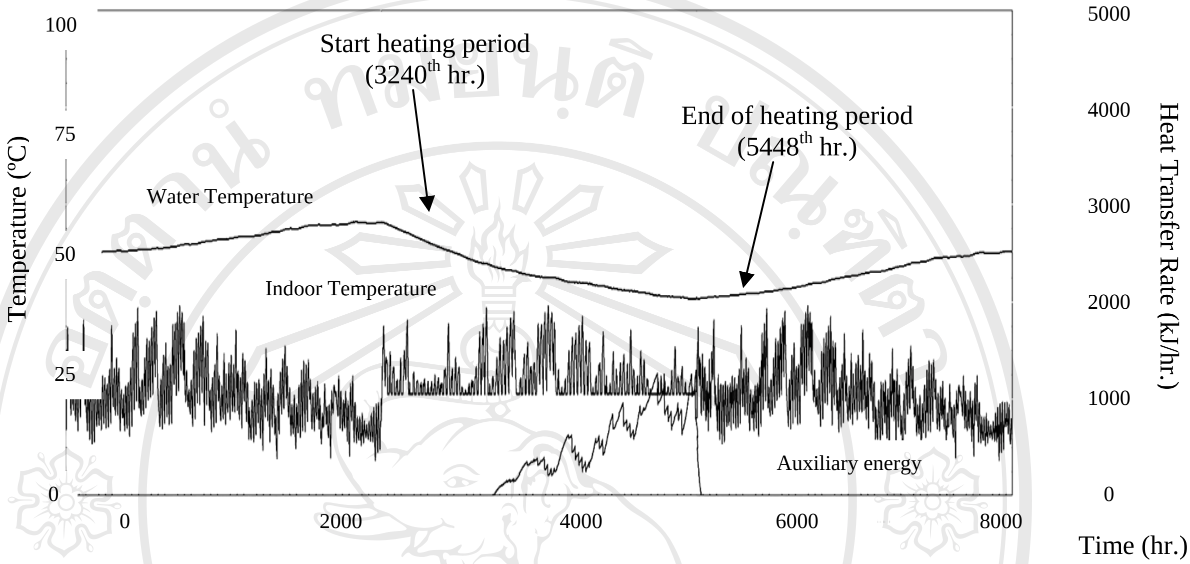

Fig 2. The simulation result by TRNSYS

Chotivisarut N. and Kiatsiriroat T. (2007), *Modeling of Cool Water Production by Thermal Convective and Radiative Nocturnal Cooling*, The 1st Energy Environment and Materials Conference, 31 August 2007, Bangkok, Thailand.

Modeling of Cool Water Production by Thermal Convective and Radiative Nocturnal Cooling

แบบจำลองกระบวนการผลิตน้ำเย็นภาคกลางคืนแบบการพาและการแผ่รังสีความร้อน

Nammont Chotivisarut[1] and Tanongkiat Kiatsiriroat[2]

[1,2]Department of Mechanical Engineering, Faculty of Engineering, Chiang Mai University, Thailand

239, Huay Kaew Rd., 50202, Chiang Mai, Thailand

[1](E-mail: *nammont4426409@hotmail.com*)

[2](E-mail: *tanong@dome.eng.cmu.ac.th*)

Abstract

The objective of this research is to develop a model of cool water production by a thermal convective and radiative nocturnal cooling unit. The heat transfer unit is an inclined thermosyphon heat pipe of which the condenser acts as a thermal radiator that transfers heat from its evaporator dipped in a water tank to the surrounding ambient. Cool water could be produced during the nighttime and used to serve the cooling load in a room during the daytime. The input data are the dry-bulb and the wet-bulb temperatures, the building cooling load, the area of the radiator including its heat transfer data and the volume of cool water. An experimental room has been built to verify the model. The radiator consists of a set of thermosyphon heat pipes made of 48 copper tubes each of 19.05 mm in diameter. Its condenser section of 6.36 m^2 acts as the radiator in the nighttime and its evaporator section is dipped in a well-insulated rectangular water storage tank of 1 m^3. The radiator is installed on a 45 degree tilting roof of the tested room. Cool water in the storage tank is fed through six cooling coils each of 0.87 m^2 installed in a well-insulated room at the ceiling to extract artificial heat load obtained during the daytime from a set of electrical heaters. The room has dimensions of 3.0 m x 3.0 m x 2.5 m. With a period of time, the temperature of the cool water in the storage tank could be lower than that of the ambient air in the nighttime. The simulation results agree well with those of the experimental data.

Keywords

air-conditioning/ nocturnal cooling/ system modeling/ water cooling

บทคัดย่อ

งานวิจัยนี้มีจุดประสงค์เพื่อสร้างแบบจำลองทางคณิตศาสตร์ของกระบวนการผลิตน้ำเย็นภาคกลางคืนแบบการพาและการแผ่รังสีความร้อน โดยใช้ข้อมูลจากการทดลองมาสร้างเป็นแบบจำลอง ระบบประกอบด้วยท่อความร้อนแบบเทอร์โมไซฟอน โดยท่อความร้อนที่ใช้เป็นท่อทองแดงขนาดเส้นผ่านศูนย์กลาง 19.05 มม. จำนวน

48 ท่อ ซึ่งส่วนทำระเหยนั้นจุ่มอยู่ในถังเก็บน้ำขนาด 1 ม.^{3} โดยส่วนควบแน่นนั้นวางตัวในแนวเอียง 45 องศา และยึดติดกับแผงแผ่รังสีความร้อน ในเวลากลางคืนเมื่ออากาศแวดล้อมมีอุณหภูมิต่ำกว่าอุณหภูมิของน้ำ ท่อความร้อนจะดึงความร้อนออกจากถังเก็บน้ำโดยการพาความร้อน อุณหภูมิของน้ำในถังเก็บน้ำจะมีค่าลดลงเรื่อยๆ นอกจากนั้นในเวลาที่อุณหภูมิของท้องฟ้าต่ำกว่าอุณหภูมิของแผงแผ่รังสี ระบบจะแผ่รังสีความร้อนสู่ท้องฟ้าเป็นการช่วยให้อุณหภูมิของน้ำในถังเก็บน้ำมีค่าลดลงอีกทางหนึ่ง โดยน้ำเย็นที่ผลิตได้ในเวลากลางคืนจะถูกนำไปใช้เพื่อช่วยการระบายความร้อนในเวลากลางวันออกจากห้องทดสอบขนาด 3.0 ม. X 3.0 ม. X 2.5 ม. โดยผ่านแผงแลกเปลี่ยนความร้อนที่ติดตั้ง ณ เพดานห้องจำนวน 6 แผง ซึ่งแต่ละแผงมีขนาดพื้นที่ 0.87 ม.^{2} ในห้องทดสอบมีแหล่งกำเนิดความร้อนแบบปรับค่าได้เพื่อใช้เป็นตัวแทนภาระความร้อนในชีวิตประจำวัน ผลการทดลองและการทำนายอุณหภูมิน้ำแสดงให้เห็นว่า อุณหภูมิของน้ำเย็นที่ผลิตได้สามารถต่ำว่าอุณหภูมิบรรยากาศแวดล้อมในเวลากลางคืนได้ และแบบจำลองที่สร้างขึ้นสามารถทำนายอุณหภูมิน้ำในถังเก็บได้เป็นอย่างดี

คำสำคัญ

การปรับอากาศ/ การทำความเย็นภาคกลางคืน/ แบบจำลองของระบบ/ การทำน้ำเย็น

Introduction

One of the passive cooling methods that can reduce energy consumption in cooling building application is night-sky cooling technique. Former study (Vimolrat and Kiatsiriroat 2004) proposed a new concept of nocturnal cooling by using a thermosyphon-radiator for producing cool water in a Trout fish farm. The evaporation part of the thermosyphon heat pipe is dipped in a water tank and the condenser part is exposed to the ambient air. During the nighttime, the condenser is cooled by convection with the surrounding air and by radiation to the sky. Then the water temperature in the tank could be reduced by transferring heat to the evaporator of the thermosyphon. Cool water obtained in the nighttime was used to reduce the nursery room temperature in the daytime. This concept could be applied in any air-conditioned building for reduction of energy used in its air-conditioning system.

In this study, a model of cool water production by thermal convective and radiative nocturnal cooling is developed. The heat transfer unit is an inclined thermosyphon heat pipe of which the condenser acts as a thermal radiator that transfers heat from its evaporator dipped in a water tank to the surrounding ambient. Cool water could be produced during the nighttime and used to serve the cooling load in a room during the daytime.

Experimental Setup

The tested room as shown in Figure 1 was a well-insulated with dimensions of 3.0 m x 3.0 m x 2.5 m. Its walls and ceiling were 10 mm gypsum boards with 25.4 mm polyethylene sheets attached inside as the room insulation. There was a 1 m^3 insulated rectangular water storage tank made of galvanized steel with 304.8 mm polyethylene insulation. The room had two-sided 45 degree tilting roof made of galvanized metal sheets. A thermosyphon heat pipe consisted of a set of 48 copper tubes each of 19.05 mm in diameter. The evaporator and the condenser parts had the same length of 1.5 m whereas its 0.2 m adiabatic part was well-insulated. The evaporator length was designed for a maximum rate heat transfer (Chotivisarut and Kiatsiriroat, 2005). The condenser tubes of the thermosyphon heat pipe were well attached to the metal roof.

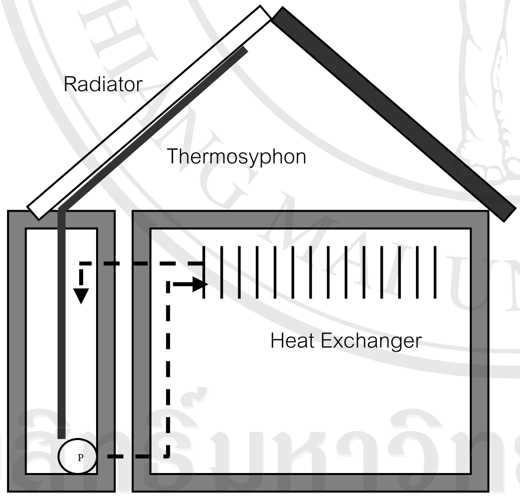

Figure 1 The tested room.

In the tested room, there were six cooling coils each of 0.87 m^2 installed at the ceiling. There was a set of electrical heaters inside as an artificial load. During the daytime a 45 W submersible pump was used to circulate cool water from the tank into the tested room. The mass flow rate of the cool water in the experiment was at 0.0167 kg s^{-1}. The IC sensors temperature data logger was used to collect multi-point temperature data every 1800 seconds time interval which the change in all data can be covered. The experiment was done in a winter during December 2005-February 2006. The nighttime ambient temperature was always below 15 ºC but the daytime ambient temperature was still high. The maximum temperature difference during a day might be over 20 ºC. The sky was quite clear thus high radiative heat transfer form the radiator surface to the sky could be obtained.

Models

By setting a control volume around the water storage tank and the thermosyphon heat pipe radiator as shown in Figure 2(a), we get

$$Q_{gain} + Q_{conv} + Q_{rad} = Q_{store}, \tag{1}$$

or

$$\frac{k_{ins} A_{\tan k}}{L}\left(T_a^i - T_w^i\right) + h_{rad} A_{rad}\left(T_a^i - T_{rad}^i\right) + \varepsilon\sigma A_{rad}\left(\left(T_{sky}^i\right)^4 - \left(T_{rad}^i\right)^4\right) = \frac{mC_p}{\Delta t}\left(T_w^{i+1} - T_w^i\right), \tag{2}$$

then

$$T_w^{i+1} = \frac{\Delta t}{mC_p}\left[\frac{k_{ins} A_{\tan k}}{L}\left(T_a^i - T_w^i\right) + h_{rad} A_{rad}\left(T_a^i - T_{rad}^i\right) + \varepsilon\sigma A_{rad}\left(\left(T_{sky}^i\right)^4 - \left(T_{rad}^i\right)^4\right)\right] + T_w^i. \tag{3}$$

k_{ins} is thermal conductivity of polyethylene sheet which equals 0.051 W m^{-1} K^{-1}, A_{tank} is the tank outer surface which is 22.65 m^2, T_a is dry-bulb temperature (K), T_w is water temperature (K) in the tank, h_{rad} is convective heat transfer of radiator sheet (W m^{-2} K^{-1}), A_{rad} is radiator sheet surface which is 6.36 m^2, T_{rad} is the radiator temperature (K), ε is the emissivity of the radiator surface which is 0.91, σ is Stefan-Boltzmann constant which is 5.67 x 10^{-8} W m^{-2} K^{-4}, T_{sky} is the sky temperature (K), m is mass of water which is 1000 kg, C_p is water heat capacity which is 4187 J $kg^{-1}K^{-1}$ and Δt is data-collected time interval which is1800 s.

The sky temperature can be expressed as (Bliss 1961):

$$T_{sky}^{i} = T_{a}^{i}\left[0.8 + \frac{T_{dp}^{i} - 273}{250}\right]^{1/4} \quad . \tag{4}$$

T_{dp} is dew point temperature (°C).

To simplify the model, the radiator temperature should be in between the in-tank water temperature and the sky temperature, we now can be express as

$$T_{rad}^{i} = C\left(\frac{T_{w}^{i} + T_{sky}^{i}}{2}\right)^{D} , \tag{5}$$

C and D are constant values and they can be found by regression method, which are C= 1.0484 and D= 0.9943, then

$$T_{rad}^{i} = 1.0484\left(\frac{T_{w}^{i} + T_{sky}^{i}}{2}\right)^{0.9943} . \tag{6}$$

Beside, the term $h_{rad} A_{rad}$ is experimentally found to be 55.1 W K^{-1}. .

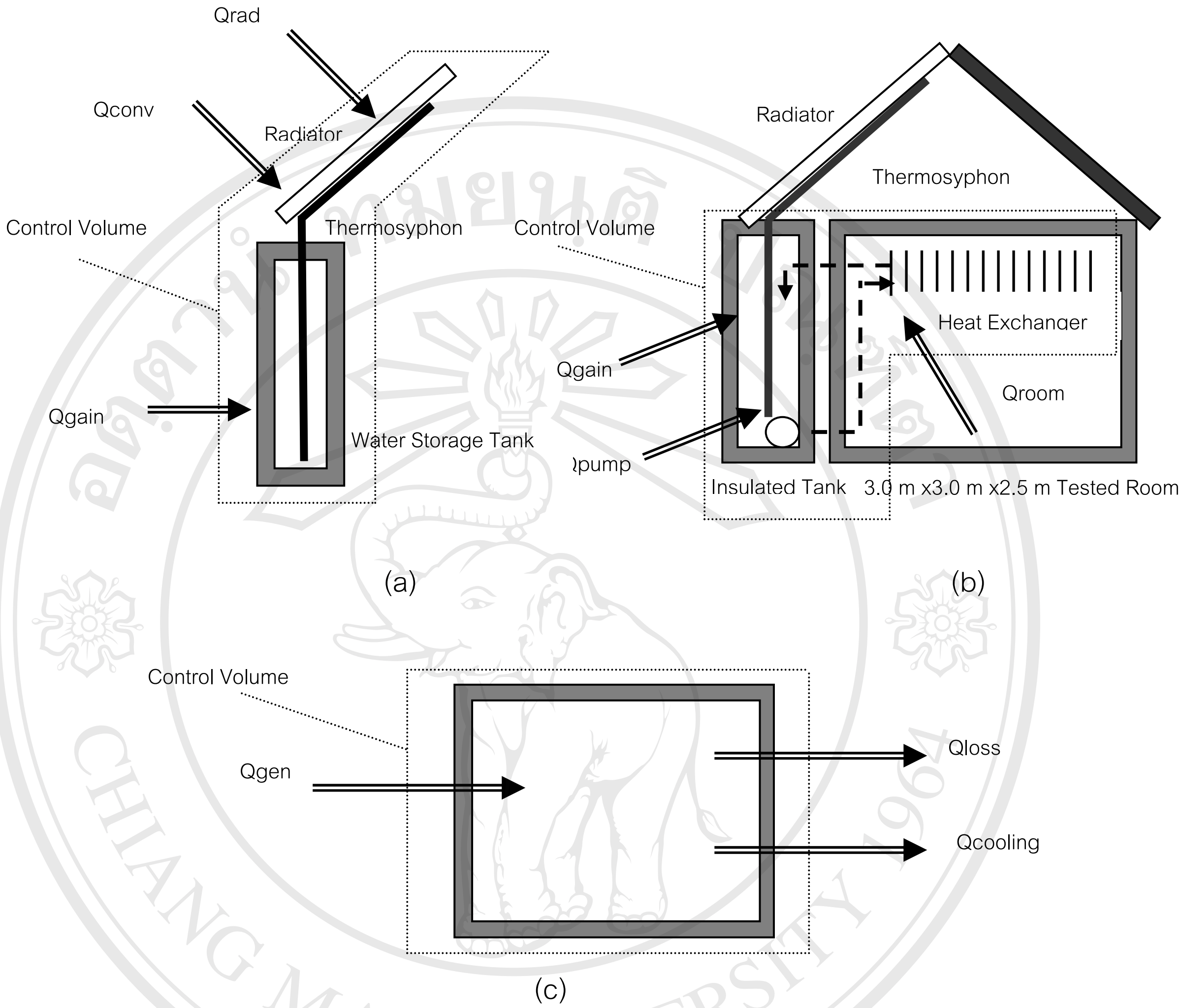

Figure 2 Schematic drawing of the cooling system.

In the daytime, the thermosyphon heat pipe works as a thermal diode, we can neglect convection and radiation, therefore, Equation (3) can be expressed as

$$T_w^{i+1} = \frac{\Delta t}{mC_p}\left[\frac{k_{ins}A_{\tan k}}{L}\left(T_a^i - T_w^i\right)\right] + T_w^i . \qquad (7)$$

The energy balance at the water storage tank and the cooling coil as shown in Figure 2(b) during the daytime when the room and the tank are well-insulated could be

$$Q_{gain} + Q_{pump} + Q_{room} = Q_{store} . \qquad (8)$$

The power from the submersible pump is 45 W, then

$$\frac{k_{ins}A_{i\tan k}}{L}\left(T_a^i - T_w^i\right) + 45 + (UA)_{coil}\left(T_{room}^i - T_{wc}^i\right) = \frac{mC_p}{\Delta t}\left(T_w^{i+1} - T_w^i\right). \qquad (9)$$

T_{room} is the room temperature (K), T_{wc} is the water temperature in the cooling coil (K). $(UA)_{coil}$ could be found by experimental testing which is 103.7 W K^{-1}. To simplify the calculation, the The water temperature T_{wc} is assumed to be

$$T_{wc}^i = \frac{T_{wc(in)}^i + T_{wc(out)}^i}{2} \quad , \qquad (10)$$

where $T_{wc(in)}$, $T_{wc(out)}$ are inlet and outlet water temperatures (K) of the cooling coil. The inlet water temperature is also assumed to be the same as the water temperature in the storage tank, then

$$T_{wc}^i = \frac{T_w^i + T_{wc(out)}^i}{2} \,. \qquad (11)$$

The outlet water temperature could be found from the heat exchanger design equation as

$$T_{wc(out)}^i = T_w^i + \left(T_{room}^i - T_w^i\right)\left(1 - e^{\left(-(UA)_{coil}/\dot{m}C_p\right)}\right), \qquad (12)$$

$\dot{m}$ is the mass flow rate of the cool water and in the experiment, it is 0.0167 kg s^{-1}.

Finally, the energy balance at the tested room could be

$$Q_{gen} = Q_{cooling} + Q_{loss} \,, \qquad (13)$$

or

$$Q_{gen} = (UA)_{coil}\left(T_{room}^i - T_{wc}^i\right) + \frac{k_{ins}A_{room}}{L}\left(T_{room}^i - T_a^i\right). \qquad (14)$$

The rates of heat generation in the experiments are 500, 1000, 1500 and 2000 W.

Experimental and Simulation Results

Figure 3 shows the experimental data of water temperature in the storage tank with different artificial thermal loads in the daytime. It could be seen that the water temperature drops down in the nighttime and increases in the daytime due to the nocturnal cooling and daytime heat absorbing, respectively. The system can produce cool water that its temperature is lower than the lowest ambient temperature in that night around 1.8 ºC. The simulation results agree quite well with the experimental data which means that the developed model could predict the phenomena of the system quite well.

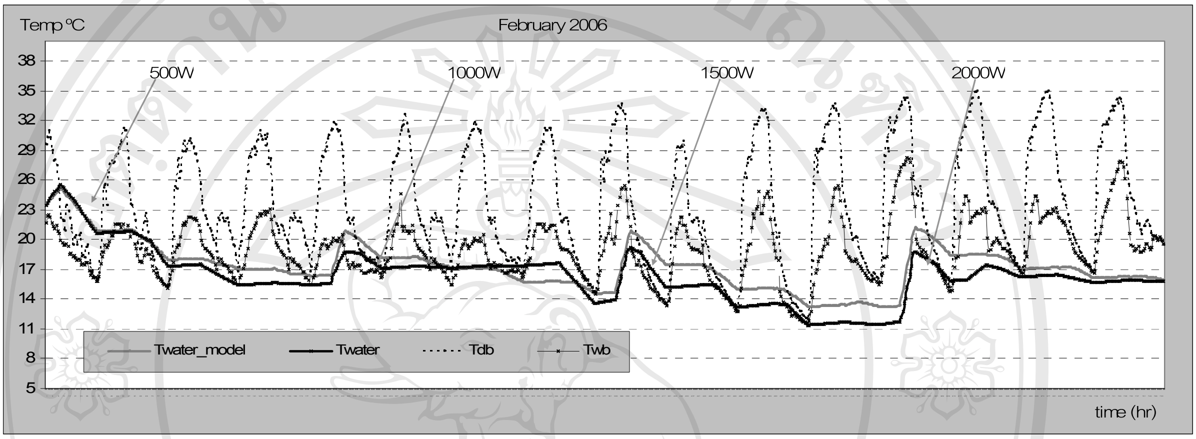

Figure 3 The experimental and simulation results in February 2006.

Conclusion

The design of nocturnal cooling by using thermosyphon radiator gives a high potential to generate cool water for air-conditioning in building during daytime. In our experiment, the temperature of water in the storage tank could be lower than that of the ambient temperature in the nighttime. The developed model seems to be a good tool on this passive nocturnal cooling to study the heat transfer phenomena inside the room. The simulated results such as the water temperature agree quite well with the experimental data.

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Authors

Nammont Chotivisarut and Tanongkiat Kiatsiriroat

First Author (Corresponding Author): Nammont Chotivisarut
Email: nammont4426409@hotmail.com
Tel: +66 85 866 9848
Fax: +66 53 810 258
Postal Address: Department of Mechanical Engineering, Chiang Mai University, Chiang Mai, Thailand, 50200

Second Author: Prof. Dr. Tanongkiat Kiatsiriroat
Email: tanong@dome.eng.cmu.ac.th
Tel: +66 81 950 8630
Postal Address: Department of Mechanical Engineering, Chiang Mai University, Chiang Mai, Thailand, 50200

Affiliations

Department of Mechanical Engineering, Chiang Mai University, Thailand

Abstract

In this study, a concept of using thermosyphon heat pipe to extract heat from water in a storage tank to generate cooling water was proposed. Heat pipe condenser was attached to an aluminum plate and acted as a thermal radiator while its evaporator was dipped in the water storage tank. Cooling water in the tank could be produced during the nighttime and used to serve the cooling load in a room during the daytime. A heat transfer model to calculate the water temperature and the room temperature during both the nighttime and daytime was developed. The input data were ambient temperature, dew point temperature, area of the radiator, volume of cooling water and room cooling load. The experiment was setup to verify the heat transfer model. A 9.0 m^2 tested room with six cooling coils, each of 0.87 m^2 were installed at the ceiling, was constructed along with the 1.0 m^3 water storage tank. A 500-2000 W adjustable heater was taken as an artificial load inside the room. A 6.36 m^2 radiator is installed on a 45 degree tilting roof of the tested room. The simulated results agreed very well with those of the experimental data. With the developed model, a simulation to find the sizing of the radiator area and the volume of cooling water for cooling water production during winter of Chiang Mai, Thailand was carried out. The cooling water was used for cooling during summer in an air-conditioned room with different cooling loads. The parameters in terms of room temperature, radiator area, volume of cooling water, cooling load and UA of cooling coil were considered to carry out the percent of cooling load reduction.

Keywords: nocturnal cooling: cooling load reduction: seasonal cool storage: thermosyphon heat pipe: nocturnal radiator

1. Introduction

The energy consumption is now the main topic to discuss everywhere. In the summer, cooling building application consumes the large amount of energy as much as that heating technique needs in the winter. Many passive cooling and heating techniques were studied and designed by many researchers [1-8].

Nocturnal cooling is one of the passive cooling methods that generates cooling water by rejecting heat to the surrounding ambient in the nighttime and the produced cooling water is kept in a storage tank and served the room cooling load in the daytime. The common technique is to feed water by a water pump through a metal radiator in the night [9-12]. There is heat dissipating to the ambient air by convection and to the sky by radiation then the water temperature leaving the radiator is reduced and the cooling water is kept in a storage tank. Recently, Wannaree and Kiatsiriroat [13] proposed a new technique that has no water pump by using thermosyphon heat pipe radiator to extract heat from stored water and the heat is transferred to the surrounding ambient. The concept has shown in Figure 1(a). Evaporation part of a thermosyphon heat-pipe was dipped in a water tank and its condenser part was attached to a metal plate exposed to the surrounding ambient. During nighttime, the condenser was cooled by radiation to the sky then the heat pipe extracted heat from the water at the evaporator. After that, the heat was dissipated through the condenser thus the water temperature in the tank could be dropped down. The stored cooling water could be used to serve cooling load during the daytime in any building with or without electrically operated air-conditioning for energy saving or for better thermal comfort. With the experiment for a non air-conditioned room in Chiang Mai, Thailand, the temperature in the room during the summer daytime could be reduced around 4.0-5.0 °C from the ambient temperature.

In this study, a model of cooling water production by thermal convective and radiative nocturnal cooling from thermosyphon heat pipe radiator was developed and verified with the experimental results of a tested room having artificial load. The model was applied for a seasonal analysis under the climate of Chiang Mai, Thailand. Cooling water was produced during winter and used to serve cooling load in a building in summer. The appropriate sizes of the radiator and the volume of cooling water related to the cooling load are considered.

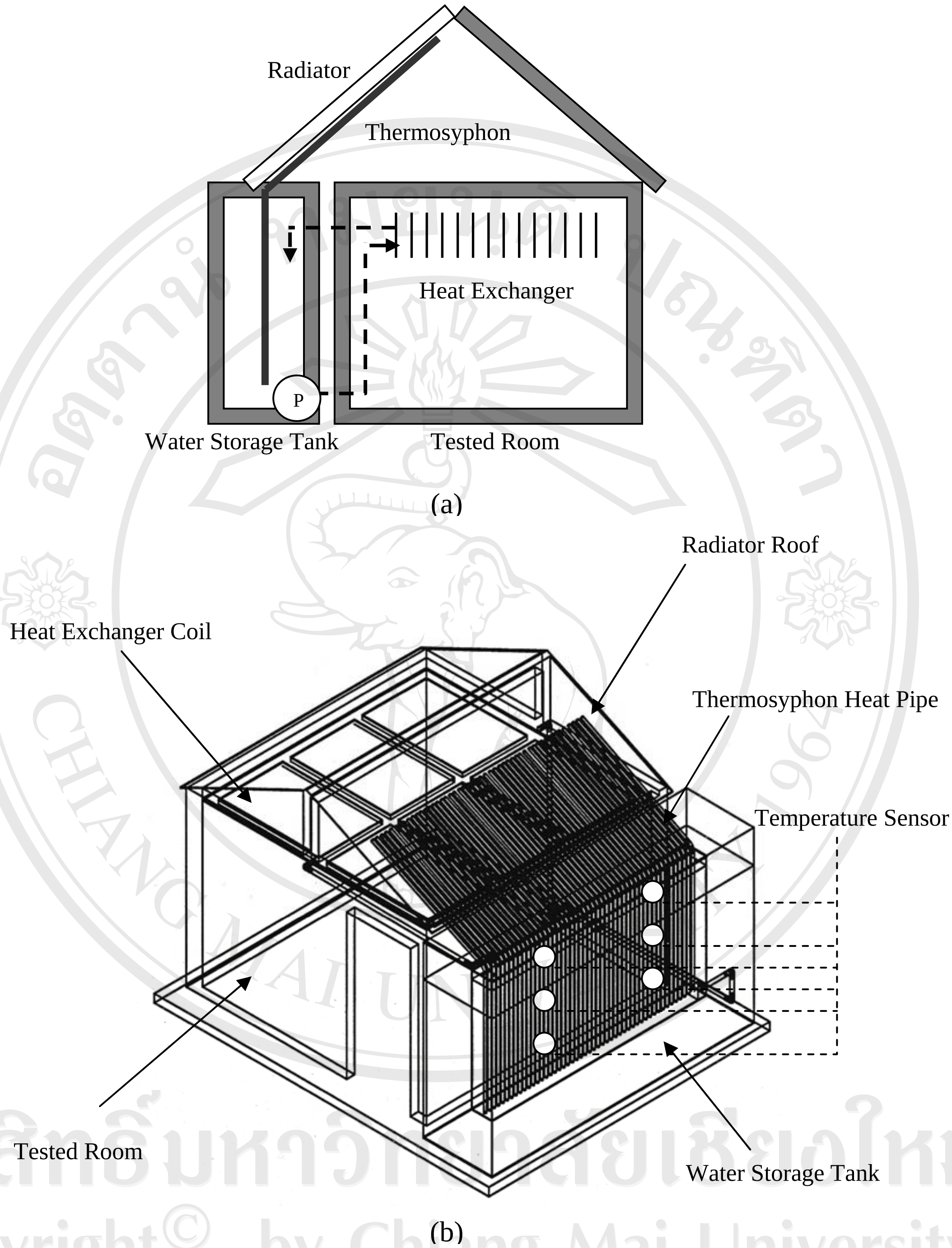


Figure 1. Schematic drawing of the heat input and output of the considered boundaries.

2. The Tested Room

An experimental room with a thermosyphon heat pipe radiator was constructed similar to that in Figure 1(b). The radiator consisted of a set of thermosyphon heat pipes made of 48 copper tubes each of 19.05 mm in outside diameter. The working fluid was R-134a. The other parameters of thermosyphon heat pipe have shown in Table 1. Its condenser section was attached to an aluminum sheet of 6.36 m^2 and acted as the radiator in the nighttime. The evaporator section was dipped in a well-insulated rectangular water storage tank of 1.0 m^3. The radiator was installed on a 45 degree tilting roof of the well-insulated tested room. Cooling water in the storage tank was fed through six cooling coils each of 0.87 m^2 were installed at the ceiling of the tested room. The room cooling load was generated during the daytime by a set of electrical heaters. The tested room had dimensions of 3.0 m x 3.0 m x 2.5 m.

Table 1. The operational parameters of thermosyphon heat pipe.

Description	**Value**	**Unit**
Outside diameter	19.05	mm
Tube thickness	0.75	mm
Evaporator section length	1.5	m
Adiabatic section length	0.2	m
Condenser section length	1.5	m
Thermal conductivity of copper tube	400.0	W m^{-1} K^{-1}
Working Fluid	R-134a	
Filling ratio	0.6	

The rates of heat generation in the tested room were 500, 1000, 1500 and 2000 W. During the daytime, a 45 W submersible pump was used to circulate cooling water from the tank into the tested room. The volume flow rate of the cooling water in the experiment was controlled at 1.00 ± 0.01 L min^{-1}. The ambient dry bulb temperature, wet bulb temperature, the stored water temperature, the room temperature, the radiator temperature, the inlet and outlet temperatures of the room cooling coil were recorded by a set of temperature sensors with ± 0.1 ^{o}C accuracy. The experiments were done during November-February under the climate of Chiang Mai, Thailand. The nighttime ambient temperature was always below 15.0 ºC when the daytime ambient temperature was rather high of which the maximum temperature difference during a day might be over 10.0 ºC [14]. The sky was quite clear thus high radiative heat transfer from the radiator surface to the sky could be obtained.

3. Simulation Model

As shown in Figure 1(a), the radiator extracts heat from the storage tank during nighttime to reduce the water temperature in the tank. The cooling water could be fed into the room through a heat exchanger, installed at the ceiling, in the daytime to cool the air in the room.

At the storage tank/radiator (Nighttime):

The energy balance at the water storage tank and the radiator as shown in Figure 1(a) during the nighttime when the tank was well-insulated could be

$$Q_{gain} + Q_{convection} - Q_{radiation} = \Delta Q_{stored} \ . \tag{1}$$

Q_{gain} is heat gain from surrounding (J), $Q_{convection}$ is convection heat transfer (J), $Q_{radiation}$ is radiation heat transfer (J), ΔQ_{stored} is the energy changed of the storage tank (J). In numerical form, the equation could be

$$\frac{k_{ins}A_{tank}\Delta t}{L_{ins}}\left(T_a^i - T_w^i\right) + h_{rad}A_{rad}\Delta t\left(T_a^i - T_{rad}^i\right) + \varepsilon\sigma A_{rad}\Delta t\left(\left(T_{sky}^i\right)^4 - \left(T_{rad}^i\right)^4\right) = m_w C_{p(w)}\left(T_w^{i+1} - T_w^i\right), \tag{2}$$

or

$$T_w^{i+1} = \frac{\Delta t}{m_w C_{p(w)}}\left[\frac{k_{ins}A_{tank}}{L_{ins}}\left(T_a^i - T_w^i\right) + h_{rad}A_{rad}\left(T_a^i - T_{rad}^i\right) + \varepsilon\sigma A_{rad}\left(\left(T_{sky}^i\right)^4 - \left(T_{rad}^i\right)^4\right)\right] + T_w^i, \tag{3}$$

where k_{ins} is thermal conductivity of storage insulation (W m^{-1} K^{-1}), L_{ins} is tank insulation thickness (m), A_{tank} is outer surface of storage tank (m^2), T_a is dry-bulb temperature (K), T_w is water temperature (K) in storage tank, h_{rad} is radiator convective heat transfer coefficient (Wm^{-2} K^{-1}) which has been experimentally found to be 8.7 W m^{-2} K^{-1}, A_{rad} is radiator area (m^2), T_{rad} is radiator temperature (K), ε is radiator emissivity, σ is Stefan-Boltzmann constant (W m^{-2} K^{-4}), T_{sky} is the sky temperature (K), m_w is mass of water (kg), $C_{p(w)}$ is water heat capacity and Δt is time interval (s) and to avoid numerical instability, this time interval should not be over 3600 seconds.

The sky temperature can be expressed by Bliss model [15] as

$$T_{sky}^i = T_a^i\left[0.8 + \frac{T_{dp}^i - 273.15}{250}\right]^{1/4}, \tag{4}$$

where T_{dp} is dew point temperature (K).

To simplify the model, the radiator temperature should be the value between the stored water temperature and the sky temperature. It could be found by our experiment and it could be expressed as

$$T_{rad}^i = 1.0484\left(\frac{T_w^i + T_{sky}^i}{2}\right)^{0.9943}. \tag{5}$$

At the storage tank/heat exchanger (Daytime):

It could be noted that the thermosyphon heat pipe acts as a thermal diode which causes heat to flow in one direction. During the nighttime, the evaporator of thermosyphon heat pipe at the bottom section is heated and the inside liquid working fluid evaporates and the vapor condenses at the condenser section, at the top, which is cooled by the surrounding ambient. The condensed liquid is then flowing downward back to the evaporator section by the gravity. In the daytime, on the contrary, the thermosyphon heat pipe is heated at the condenser section while the evaporator section, at the bottom, has lower temperature then there is no moving of the inside working fluid thus the heat transfer from the condenser section back to the evaporator stops. So we can neglect the convection and the radiation during the daytime at the radiator which transfer to the storage tank then the energy balance at the water storage tank and the cooling coil as shown in Figure 1(b) during the daytime could be

$$Q_{gain} + Q_{pump} + Q_{removed} = \Delta Q_{stored}\,, \tag{6}$$

where Q_{gain} is heat from the ambient air to the water in the tank (J), Q_{pump} is heat gain from the submersible pump (J), $Q_{removed}$ is part of cooling load absorbed at the cooling coil (J), ΔQ_{stored} is the energy changed of the storage tank (J). The equation could be

$$\frac{k_{ins} A_{\tan k} \Delta t}{L_{ins}}\left(T_a^i - T_w^i\right) + \dot{Q}_{pump}\,\Delta t + (UA)_{coil}\,\Delta t\left(T_{room}^i - T_{wc}^i\right) = m_w C_{p(w)}\left(T_w^{i+1} - T_w^i\right), \tag{7}$$

or

$$T_w^{i+1} = \frac{\Delta t}{m_w C_{p(w)}}\left[\frac{k_{ins} A_{\tan k}}{L_{ins}}\left(T_a^i - T_w^i\right) + \dot{Q}_{pump} + (UA)_{coil}\left(T_{room}^i - T_{wc}^i\right)\right] + T_w^i\,, \tag{8}$$

where T_{room} is room temperature (K) and T_{wc} is water temperature in the cooling coil (K). $\dot{Q}_{pump}$ is the rate of heat transfer from a 45 W submersible pump. *$(UA)_{coil}$* is the overall heat transfer coefficient at the room cooling coil and it could be found experimentally which is 103.7 W K^{-1} for the water volume flow rate of 1.00 ± 0.01 L min^{-1} or $(UA)_{coil}/\dot{m}_w C_{p(w)}$ equals 1.483 when $\dot{m}_w$ and $C_{p(w)}$ are mass flow rate (kg s^{-1}) and specific heat capacity (J kg^{-1}K^{-1}) of cooling water, respectively. The term $\left(T_{room}^i - T_{wc}^i\right)$ is the log-mean temperature difference between the room temperature and the water temperature in cooling coil and it can be expressed as

$$\left(T_{room}^i - T_{wc}^i\right) = \frac{\left(T_{room}^i - T_{wc(in)}^i\right) - \left(T_{room}^i - T_{wc(out)}^i\right)}{\ln\left(\dfrac{T_{room}^i - T_{wc(in)}^i}{T_{room}^i - T_{wc(out)}^i}\right)}, \tag{9}$$

where $T_{wc(in)}$, $T_{wc(out)}$ are inlet and outlet water temperatures at the cooling coil (K), respectively. The inlet water temperature from our experiment is found to be closer to the water temperature in the storage tank, then

$$\left(T_{room}^i - T_{wc}^i\right) = \frac{\left(T_{room}^i - T_w^i\right) - \left(T_{room}^i - T_{wc(out)}^i\right)}{\ln\left(\dfrac{T_{room}^i - T_w^i}{T_{room}^i - T_{wc(out)}^i}\right)}. \tag{10}$$

The outlet water temperature could be estimated by [16]

$$T_{wc(out)}^i = T_w^i + \left(T_{room}^i - T_w^i\right)\left(1 - e^{\left(-(UA)_{coil}/\dot{m}_w C_{p(w)}\right)}\right). \tag{11}$$

At the tested room (Daytime and Nighttime):

The energy balance around the tested room when the cooling load was absorbed by the system could be written as

$$Q_{load} - Q_{removed} + Q_{gain(room)} = \Delta Q_{stored(room)}\,, \tag{12}$$

when Q_{load} is the heat load from artificial heater (J), $Q_{gain(room)}$ is heat load from the ambient air to the air in the tested room (J), $\Delta Q_{stored(room)}$ is the energy changed of the tested room (J).

The room temperature could also be evaluated in numerical form as

$$\dot{Q}_{load}\ \Delta t - (UA)_{coil}\,\Delta t\left(T^{i}_{room} - T^{i}_{wc}\right) + \frac{k_{ins(room)} A_{room}\Delta t}{L_{ins(room)}}\left(T^{i}_{a} - T^{i}_{room}\right) = \rho_{air} V_{air} C_{v(air)}\left(T^{i+1}_{room} - T^{i}_{room}\right), \tag{13}$$

or

$$T^{i+1}_{room} = \frac{\Delta t}{\rho_{air} V_{air} C_{v(air)}}\left[\dot{Q}_{load} - (UA)_{coil}\left(T^{i}_{room} - T^{i}_{wc}\right) + \frac{k_{ins(room)} A_{room}}{L_{ins(room)}}\left(T^{i}_{a} - T^{i}_{room}\right)\right] + T^{i}_{room}, \tag{14}$$

where T_{room} is room temperature (K), ρ_{air} is density of air (kg m^{-3}), V_{air} is inside tested room volume (m^{3}), $C_{v(air)}$ is specific heat capacity (J kg^{-1}K^{-1}) of air, $\dot{Q}_{load}$ is the rate of cooling load from artificial heater (W), $k_{ins(room)}$ is thermal conductivity of tested room insulation (W m^{-1} K^{-1}), $L_{ins(room)}$ is room insulation thickness (m) and A_{room} is outer surface of tested room (m^{2}). To avoid numerical instability, this time interval Δt should not be over 300 seconds.

During the nighttime, the load from artificial heater and cooling load absorbed at the cooling coil were neglected.

During the daytime, the total heat absorbed from the tested room could be indicated by $\Sigma(\Delta Q_{removed})$ from each interval time. From Equation 6, since the storage tank is well-insulated and the heat transfer from pump is neglected, therefore, $\Sigma(\Delta Q_{removed})$ could also still be calculated directly from

$$\Sigma(\Delta Q_{removed}) = \Sigma\left(m_w C_{p(w)}\left(T^{i}_{w} - T^{i-1}_{w}\right)\right). \tag{15}$$

If there is any active cooling such as air-conditioning system, this nocturnal technique can take part in the room cooling load. The percent of cooling load reduction can be

$$\%Load\ Reduction = \frac{\Sigma\left(\Delta Q_{removed}\right)}{\Sigma Q_{load(overall)}} \times 100, \tag{16}$$

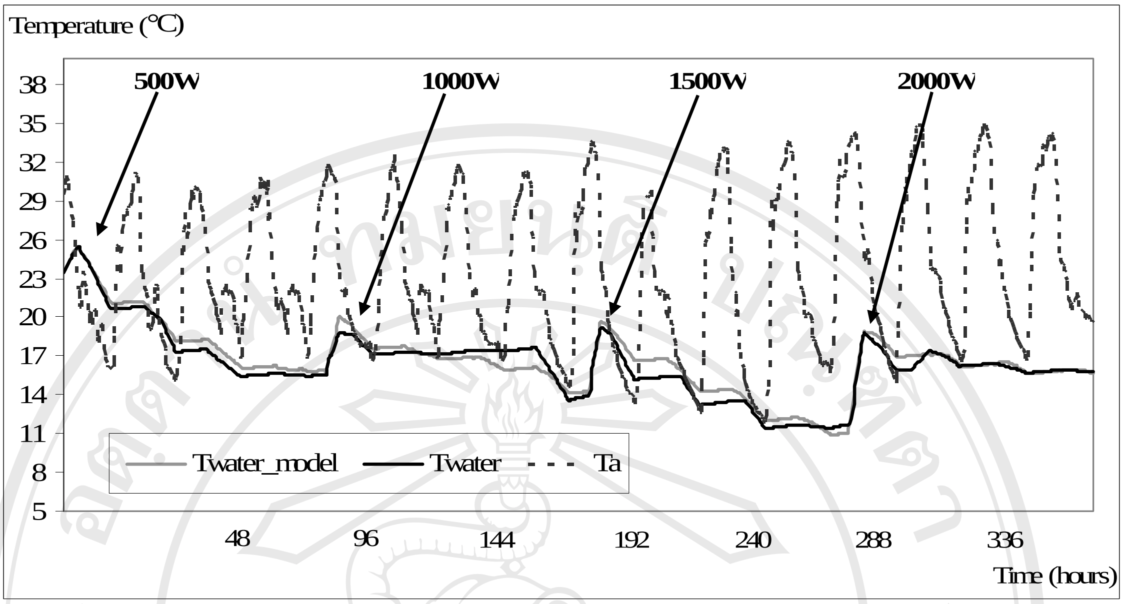
where $\Sigma Q_{load(overall)}$ is the total cooling load (J) during the daytime which include all heat such as heat gains from people, electrical devices and other cooling loads.

4. Model Verification

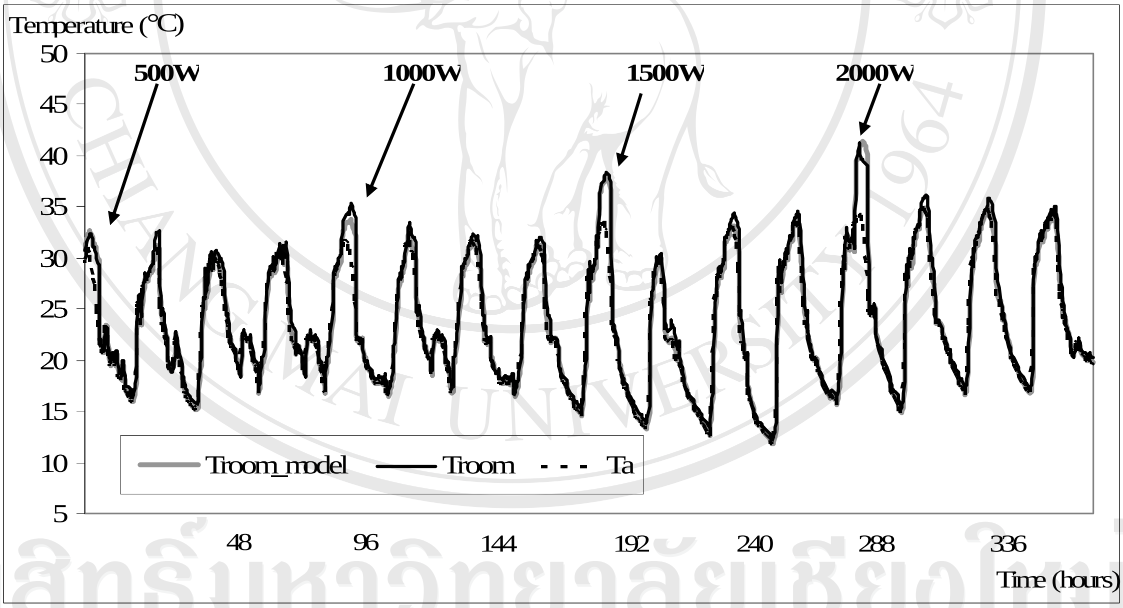
Figure 2 has shown the experimental data of the average water temperature in the storage tank and the room temperature of the tested room. The conditions for the simulation have given in Table 2. The experiment was started around 1.00 p.m. with the average water temperature of 24.0 °C. The artificial load was supplied at 500 W for 5 hours then the load was off. The feed pump of cooling water into the room was also stopped to see the effect of heat reversed back from the radiator during the daytime into the storage tank. It could be seen that the water temperature was up when there was a thermal load in the room and after that when there was no load, the temperature dropped down to a certain value. During the daytime on the next day (without artificial load), even the ambient temperature increased but the water temperature in the tank was nearly constant which meant that very small amount of heat rate from the radiator could return back to the storage tank due to the thermal diode effect of the heat pipe. During the nighttime, the water temperature could continually drop down again. For the room temperature when there was artificial load, since the cooling coil could absorb only some part of the heat load then the room temperature increased to a certain value and when the artificial load was off, the room temperature came closer to the ambient temperature. The system continued without the artificial load on three more consecutive days and it could be found that the water temperature in the tank could be lower than that of the lowest temperature of the ambient temperature on those days. The procedure was repeated by supplying the artificial loads at 1000, 1500 and 2000 W on the first day for 5 hours and let the system to cool down by heat dissipation at the radiator. The profiles of the water and the room temperatures were similar to all thermal loads.

Table 2. The conditions for the simulation of the tested room.

Description	**Value**	**Unit**
Floor area	9.0	m^2
Simulation interval	1800	s
Storage volume	1.0	m^3
Mass of storage water	1000	kg
Thermal conductivity of storage tank insulation	0.040	$W\ m^{-1}\ K^{-1}$
Storage surface area	11.922	m^2
Storage tank insulation thickness	300	mm
Radiator average heat transfer coefficient of convection	8.7	$W\ m^{-2}\ K^{-1}$
Radiator area	6.36	m^2
Radiator emissivity	0.9	
Submersible pump power	45	W
$UA/\dot{m}_w C_{p(w)}$	1.483	
Cooling coil effectiveness	0.773	
Tested room space volume	22.5	m^3
Cooling load	500-2000	W
Overall heat transfer coefficient at the cooling coil	103.7	$W\ K^{-1}$
Thermal conductivity of tested room insulation	0.051	$W\ m^{-1}\ K^{-1}$
Tested room surface area	39.0	m^2
Tested room insulation thickness	25	mm
Daytime operation	5	hours



(a) Stored water temperature.



(b) Tested room temperature.

Figure 2. Comparison of the stored water temperature and tested room temperature from the experiment and those from the simulation results.

It could be seen that the simulated results agreed very well with those of the experimental data within 10.0% and 8.0% errors for the water and the room temperatures, respectively.

It could be noted that during the nighttime, the radiator temperature is always lower than that of ambient, then there is convection heat load from the ambient air to the radiator. The average ratio of the convection heat load to the radiation heat rejection from our experiment is found to be 0.41.

5. Seasonal Analysis

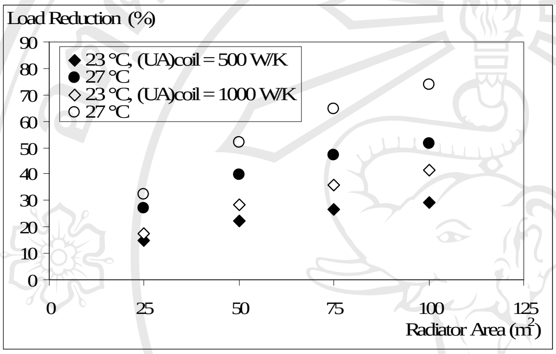
For seasonal analysis, cooling water will be produced and kept in a storage tank during winter and it will be used to serve building cooling load in summer. In this section, the concept of nocturnal cooling with thermosyphon heat pipe radiator to assist thermal load of air-conditioned building was considered. The conditions for the simulation have given in Table 3. The calculations were carried out with the weather data of Chiang Mai, Thailand.

Table 3. The conditions for seasonal simulation.

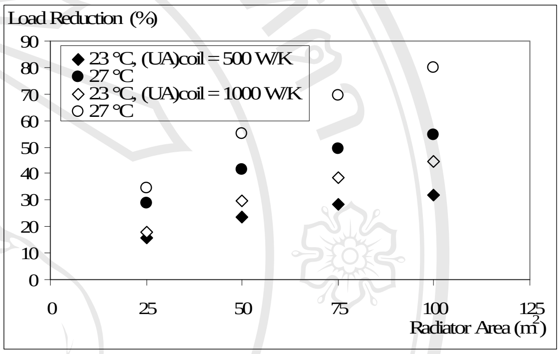
Description	**Value**	**Unit**
Floor area	45.0	m^2
Simulation interval	3600	s
Storage volume	5.0-15.0	m^3
Mass of storage water	5000-15000	kg
Thermal conductivity of storage tank insulation	0.040	$W\ m^{-1}\ K^{-1}$
Storage surface area	57.21-170.43	m^2
Storage tank insulation thickness	300	mm
Radiator average heat transfer coefficient of convection	8.7	$W\ m^{-2}\ K^{-1}$
Radiator area	25-100	m^2
Radiator emissivity	0.9	
$UA/\dot{m}_w C_{p(w)}$	1.483	
Cooling coil effectiveness	0.773	
Cooling load	3517-35170	W
	(1.0-10.0)	Tons of refrigeration
Overall heat transfer coefficient at the cooling coil	500, 1000	$W\ K^{-1}$
Thermal conductivity of tested room insulation	0.051	$W\ m^{-1}\ K^{-1}$
Controlled space temperature	23.0-27.0	°C
Tested room insulation thickness	25	mm
Daytime operation	12	hours

The calculation started from November which was the winter starting time. The stored water temperature would be reduced gradually to the lowest temperature at around 11.7 °C near Christmas. The produced cooling water then was used for cooling in the building during March to June which was summer period. In the last four months, July to October, which was rainy season, the water was not fed into the building, the water temperature was then reduced night by night to a certain value and the new cycle restarted.

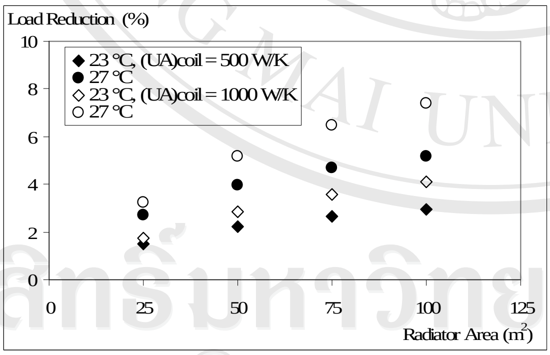
Figure 3 has shown the percent load reduction at different cooling load performed by the nocturnal cooling water with various values of the radiator area, the storage volume and the controlled inside room temperature. Again, it could be seen that the cooling coil with higher value of *UA* could get more heat rate which results in higher percent load reduction. Higher value of radiator area gives more heat extraction rate during the nighttime thus lower cooling water temperature and higher percent load reduction are obtained. From our simulation, it could be noted that the size of the cooling storage tank has slightly effect on the percent cooling load reduction. With a fixed cooling load, when the size of the storage tank was over 5 m^3, the percent load reduction increased only slightly due to the high thermal inertia of the water volume which resulted in low variation of water temperature in the storage tank. But when the size was less than 5 m^3, the water temperature in the storage tank was increased quickly then the rate of rejected heat was also decreased. From the Figure, it could be seen that when the inside room temperature was higher, the potential to reduce the cooling load by the cooling water was higher but when the cooling load is higher, the percent load reduction was less.



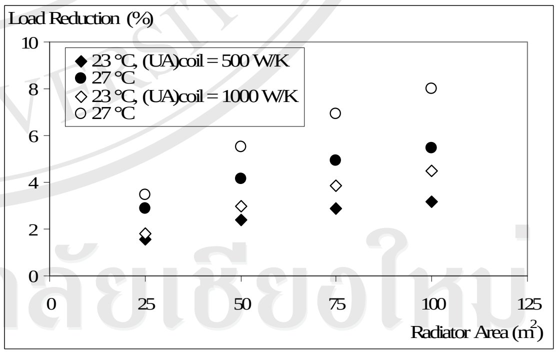
(a) With 5 m^3 storage volume
3517 W cooling load
(1.0 ton of refrigeration).



(b) With 15 m^3 storage volume
3517 W cooling load
(1.0 tons of refrigeration).



(c) With 5 m^3 storage volume
35170 W cooling load
(10.0 tons of refrigeration).



(d) With 15 m^3 storage volume
35170 W cooling load
(10.0 tons of refrigeration).

Figure 3. The percent cooling load reduction in the summer period (1 March -30 June) during daytime.

The simulation program by all equations above could be simplified to an empirical correlation for predicting the percent load reduction, under the specified parameters of thermosyphon heat pipe as shown in Table 1, was also given as

$$\%Load\ Reduction = \frac{0.0063}{\dot{Q}_{load}} A_{rad}^{0.556} V_{tank}^{0.056} \left(T_{room} - 273.15\right)^{3.679} \left(UA\right)_{coil}^{0.399}, \quad (17)$$

where $\dot{Q}_{load}$ is the total rate of cooling load (3517 W< $\dot{Q}_{load}$ < 35170 W), A_{rad} is the radiator area (25.0 m^2 < A_{rad} < 100.0 m^2), V_{tank} is storage tank volume (5 m^3< V_{tank} < 15 m^3), T_{room} is the controlled room temperature (296.15 K < T_{room} < 300.15 K) and $(UA)_{coil}$ is overall heat transfer coefficient of cooling coil (500 W K^{-1} < $(UA)_{coil}$ < 1000 W K^{-1}). The correlation could predict the percentage load reduction within ± 14.0 % error.

6. Conclusion

A concept of using thermosyphon heat pipe to extract heat from water in a storage tank to generate cooling water in the nighttime was presented. Cooling water in the tank was produced during the nighttime could be used to serve some part of the cooling load in a room during the daytime. A heat transfer model for calculating the water and the room temperatures was developed and the results agreed very well with those of the experiments in a tested room. With the developed model, a seasonal analysis to produce cooling water in winter which will be used to serve some part of the cooling load of an air-conditioned building in summer under the climate of Chiang Mai, Thailand, at various room temperatures and cooling loads was described. It could be found that there was a high potential to implement this technique. A correlation for evaluating the percent load reduction with different values of radiator area, cooling load and room temperature was also presented.

Acknowledgements

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NOMENCLATURE AND ABBREVIATIONS

Latin Symbols

Letter	Description	Unit
A	area,	m^2
C_p	constant pressure specific heat,	$J\ kg^{-1}\ K^{-1}$
C_v	constant volume specific heat,	$J\ kg^{-1}\ K^{-1}$
h	convective heat transfer of radiator sheet,	$W\ m^{-2}\ K^{-1}$
k	thermal conductivity,	$W\ m^{-1}\ K^{-1}$
L	thickness,	m
m	mass,	kg
$\dot{m}$	mass flow rate,	$kg\ s^{-1}$
Q	heat transfer,	J
$\dot{Q}$	rate of heat transfer	W
T	temperature,	K
U	overall heat transfer coefficient,	$W\ m^{-2}\ K^{-1}$
V	volume	m^3

Greek Letters

Letter	Description	Unit
Δt	time interval,	s
ε	the emissivity of the radiator surface	--
ρ	the density,	$kg\ m^{-3}$
σ	the Stefan-Boltzmann constant,	$W\ m^{-2}\ K^{-4}$

Subscripts and Superscripts

Letter	Description
a	ambient
air	air
coil	heat exchanger coil
convection	convection
dp	dew point
gain	gain from surrounding to the storage tank
gain(room)	gain from surrounding to tested room
i	the present number of calculation
i+1	the later number of calculation
i-1	the former number of calculation
ins	storage tank insulation
ins(room)	tested room insulation
load	load from artificial heater
load(overall)	total cooling load during the daytime
pump	submersible pump
rad	radiator
radiation	radiation
removed	removed at the cooling coil
room	room

sky	atmosphere and outer space
stored	stored in storage tank
stored(room)	stored in tested room
tank	storage tank
w	water
wc	water in heat exchanger coil
wc(in)	water in heat exchanger coil at inlet point
wc(out)	water in heat exchanger coil at outlet point

APPENDIX B

Source Code of Engineering Equation Solver (EES) for finding Rate of Heat Rejected by Thermosyphon Heat Pipe

```
Procedure Nocturnal(T_so,T_si,D_o,th,k_t,l_e,l_a,l_c,F,Beta,m,time_set,time_step :
A_c,A_e,Q_Exact[0..9],T_so[0..10],time[0..10],Q_MaxSonic[0..9],Q_MaxBoilingLimit[0..9])

g = 9.81 [m/s^2];
P_a = 101300 [Pa];
C_p = 4179 [J/kg*C];
D_i = D_o-(2*th);
A_e = Pi*D_o*l_e;
A_c = Pi*D_o*l_c;
i = 0;
T_so[0]=T_so;
time[0] = 0 [sec];

Repeat

h_e = 7 [W/(m^2*K)];
h_c = 7 [W/(m^2*K)];

z_1 = 1/(h_e*A_e);
z_9 = 1/(h_c*A_c);
z_2 = (ln(D_o/D_i))/(2*pi*l_e*k_t);
z_8 = (ln(D_o/D_i))/(2*pi*l_c*K_t);
z_3 = 0; z_4 = 0; z_5 = 0; z_6 = 0; z_7 = 0;
Z = z_1+z_2+z_3+z_4+z_5+z_6+z_7+z_8+z_9;

T_v = T_si+(((z_7+z_8+z_9)/Z)*(T_so[i]-T_si));
P_v = Pressure(R134a,T=T_v,x=1);
Rho_l = Density(R134a,T=T_v,x=0);
Rho_v = Density(R134a, T=T_v,x=1);
h_f = Enthalpy(R134a,T=T_v,x=0);
```

```
h_g = Enthalpy(R134a,T=T_v,x=1);
L = h_g-h_f;
Mu_l = Viscosity(R134a,T=T_v,x=0);
k_f = Conductivity(R134a,T=T_v,x=0);
Tor = SurfaceTension(R134a,T=T_v);
Cp_l = Specheat(R134a,T=T_v,x=0);
Phi_2 = ((L*k_f^3*Rho_l^2)/Mu_l)^0.25;
P_p = P_v + (Rho_l*g*F*l_e*sin(Beta));
T_p = Temperature(R134a,P=P_p,x=0);
DeltaT_h = (T_p-T_v)*F/2;
DeltaT = T_so[i]-T_si-DeltaT_h;
Q = DeltaT/Z;

Re_f = 4*Q/(L*Mu_l*Pi*D_i);
        If (50<Re_f) and (Re_f<1300) Then
        z_7new = (0.235*Q^(1/3))/(D_i^(4/3)*g^(1/3)*l_c*Phi_2^(4/3));
        Else
        z_7new = (0.235*Q^(1/3))/(D_i^(4/3)*g^(1/3)*l_c*Phi_2^(4/3))*191*Re_f^(-0.733);
        EndIf
z_3f = (0.235*Q^(1/3))/(D_i^(4/3)*g^(1/3)*l_e*Phi_2^(4/3));
Phi_3 = (Rho_l^0.65*k_f^0.3*Cp_l^0.7)*(P_v/P_a)^0.23/(Rho_v^0.25*L^0.4*Mu_l^0.1);
z_3p = 1/(Phi_3*g^0.2*Q^0.4*(Pi*D_i*l_e)^0.6);
        If (z_3p<z_3f) Then
        z_3new = z_3p
        Else
        z_3new = (z_3p*F)+(z_3f*(1-F));
        EndIf
z_3 = z_3new;
z_7 = z_7new;
Z_new = z_1+z_2+z_3+z_4+z_5+z_6+z_7+z_8+z_9;
```

```
Q_new = DeltaT/Z_new;

A_i = Pi*(D_i^2)/4;
S_e = Pi*D_i*l_e;
Q_MaxSonic[i] = 0.5*A_i*L*(P_v*Rho_v)^0.5;
Q_MaxBoilingLimit[i] = 0.12*S_e*L*Rho_v^0.5*(Tor*g*(Rho_l-Rho_v))^0.25;

Q_Exact[i] = Q_new;

T_so[i+1] = T_so[i]-(Q_Exact[i]*time_step/(m*C_p));
time[i+1] = time[i]+time_step;
i = i+1;
Until time[i] >= time_set
End

CALL Nocturnal(T_so,T_si,D_o,th,k_t,l_e,l_a,l_c,F,Beta,m,time_set,time_step :
A_c,A_e,Q_Exact[0..9],T_so[0..10],time[0..10],Q_MaxSonic[0..9],Q_MaxBoilingLimit[0..9])
```

CURRICULUM VITAE

Name and Surname:	Mr. Nammont Chotivisarut
Date of Birth:	14 March 1979
Education:	M. Eng Degree in Mechanical Engineering from Chiang Mai University in 2002
	B. Eng Degree in Mechanical Engineering from Chiang Mai University in 2000
Specialist:	Heat Transfer Analysis, Computational Fluid Dynamics
Research Interest:	Passive Cooling of Building

Publications:

- Chotivisarut N. and Kiatsiriroat T. (2005), *Effect of Thermosyphon Evaporative Length on Temperature Reduction of Cool Water*, The 19th Conference of Mechanical Engineering Network of Thailand, 19-21 October 2005, Phuket, Thailand.

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