[논문] 한국태양에너지학회 논문집
Journal of the Korean Solar Energy Society
Vol. 22, No. 3, 2002

Application of One-Dimensional Heat Transfer Models for Performance Prediction of a Thermodiode

열다이오드의 열전달 성능 예측을 위한 일차원 모델적용 연구

Wongee Chun*, Yong Heack Kang**
천원기*, 강용혁**

Key words : Solar energy(태양열), Thermodiode(열다이오드), Analytical model(해석적 모델), Performance analysis(성능분석)

요 약

그 열흐름의 방향이 인위적으로 조절 가능한 열다이오드 시스템에 관하여 일차원적 열전달 모델을 통하여 시스템의 열성능을 분석하였다. 열다이오드 시스템은 다수의 폐회로 유체 순환 루프로 구성되었으며 루프의 양단은 각각 태양열 흡열판과 방열판에 부착되었다. 한편, 열흐름의 방향 조절을 위하여 루프를 구성하는 튜브재의 연결 부위는 회전 가능한 조인트로 연결하였으며 열매체로는 물을 사용하였다. 본 연구에서는 열다이오드 시스템에 대하여 간단한 1차원 모델을 이용하여 시스템의 열성능을 평가하였으며 아울러 실측 결과와의 비교를 통하여 본 모델의 적용을 통한 시스템의 장기 예측에 대한 가능성을 확인하였다.

1. INTRODUCTION

Solar energy utilization and energy-efficient building designs have received great attentions for the past two decades. Most of the buildings built after the energy crisis have more airtight envelopes and used construction materials of high R-values. Many new buildings and residential houses have adopted solar house designs or have installed various types of solar heating systems.

The present study of bi-directional thermodiodes originated from the smart construction module proposed for next-generation building envelopes.[1] It was motivated to

* 제주대학교
** 한국에너지 기술연구원

* *Cheju National University*
** *Korea Institute of Energy Research*

E-mail : wgchun@cheju.ac.kr

resolve the various problems of the previous works, which deemed quite extricate to overcome. First, the heat transfer rate and dynamic behavior of a bayonet thermodiode were studied both theoretically and experimentally to delve into its operational characteristics. The diode got its name because its cross-section resembles a bayonet with a narrow cavity above or below a rectangular storage tank. Connecting the cavity and the tank is an inclined flow passageway. Despite its simple configuration and construction, the bayonet thermodiode has a major drawback in changing heat flow direction. Chen and Jones[2)-3)] extensively studied the performance of the bayonet thermodiode in the favorable mode of heat flow for winter operations.

The authors have explored the feasibility of using bi-directional thermodiodes for years to harness the solar energy more expeditiously. Different types bi-directional thermodiodes were designed, constructed, and tested in their previous works.[4)-6)] The proposed design introduced in this paper is one of those which deems most efficient and feasible among divers designs of thermodiodes. This paper presents the experimental results of the previous works in such a way that they are put forth for analysis to develop one-dimensional analytical models which are simple but fairly accurate. However, it is not the objective of the present study to make one-on-one comparison of these models but to suggest simple analytical models for the type of energy system which involves rather complicated heat flows depending on its mode of operation.

Fig. 1 shows one of the thermodiode designs with variable surface absorptivity and alterable direction of heat flow. When solar heating is desired, the outdoor-facing surface will turn into a dark color for high radiation absorption during the daytime, but become highly reflective or opaque at night to reduce radiation loss. If the rectangular loops are heated from below during the hours of bright sunshine, natural convection will be induced throughout the loops. As the temperature difference in the loop reaches a critical value, heat can be transferred efficiently across the thermodiode from its front to the rear. In this case, the thermodiode is said to be in the forward biased mode of operation. The bi-directional thermodiode allows heat flow from the outdoors to the indoors, but impedes heat loss in the opposite direction

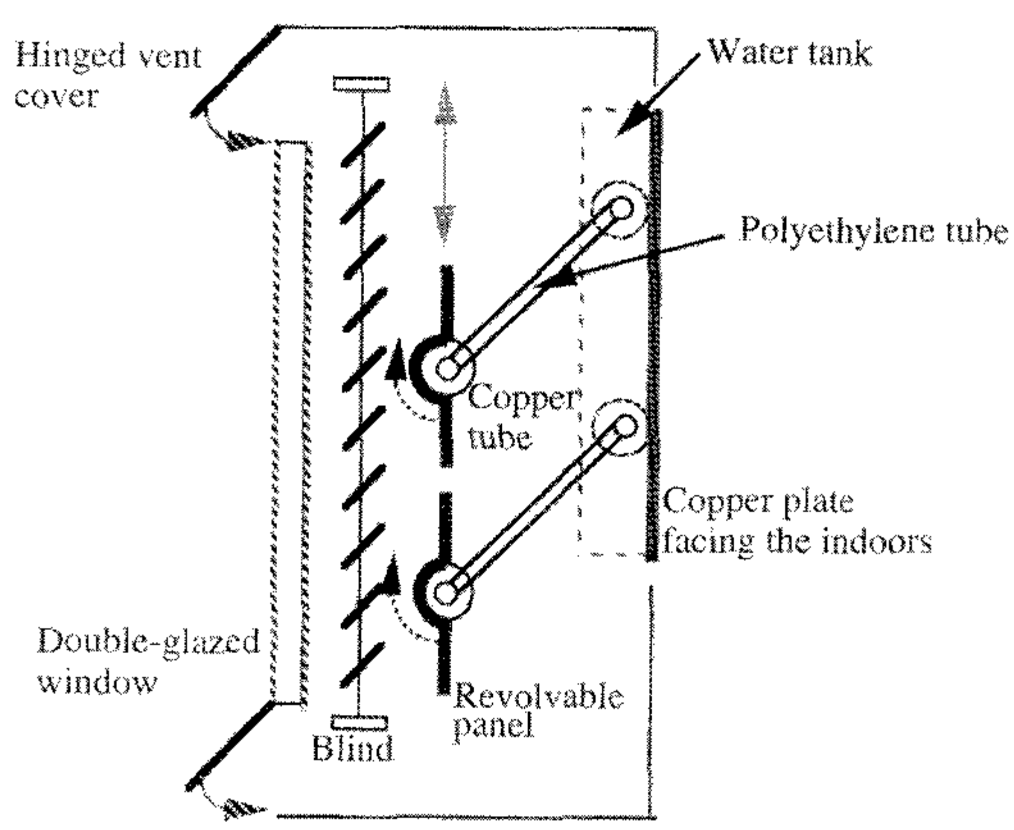

Fig. 1. A thermodiode design with alterable heat flow direction and surface absorption.

when the indoor temperature is higher than that of the outdoors. Operations of the change in surface absorptivity and the thermodiode are opposite for summer cooling application. That is, if the heated horizontal section is above the cooled section of the rectangular loop, only conduction will take place in the loop and the thermodiode is in the reverse-biased operation mode.

Unlike many existing solar houses built with passive or active schemes, the proposed energy-efficient construction module can be manufactured in large quantities in one place and shipped to the construction site for easy and quick assembly. The module design makes it convenient to disassemble and re-use these as necessary. In addition, the module can take the advantage of recent breakthroughs in microprocessor and sensor technologies to optimize and adjust its performance autonomously for different weather conditions and heating/ cooling requirements. Sensors and a microprocessor can be mounted on a master module. The weather data collected by the sensors will be transmitted to the microprocessor where the optimal operations of surface absorptivity, direction of heat flow, and heat capacity will be determined based on the required heating or cooling load. The master module then sends out electrical signals to the slave modules via built-in adapters to control the mechanisms that adjust the surface absorptivity, direction and rate of heat flow, and the module heat capacity.

2. DESIGN AND CONSTRUCTION OF THE BI-DIRECTIONAL THERMODIODE

Fig. 2 shows the major components of the present bi-directional thermodiode (Korean Patent No. 0290228), which include closed rectangular loops and metallic panels attached to the loop surfaces for increasing heat transfer area, DC motors and other mechanisms such as gears and belts for moving or rotating the metallic panels and the tubes. Each loop in the thermodiode system consist of two inclined polyethylene tubes and two horizontal copper tubes. Rotatable (swivel) joints were used between the horizontal and inclined segments of the loops, allowing vertical movements of the metal panels and changes in inclination angle of the polyethylene tubes. Flexible polyethylene tubes were employed in the present design as they could accommodate minor volume changes of the working fluid and, hence, prevent

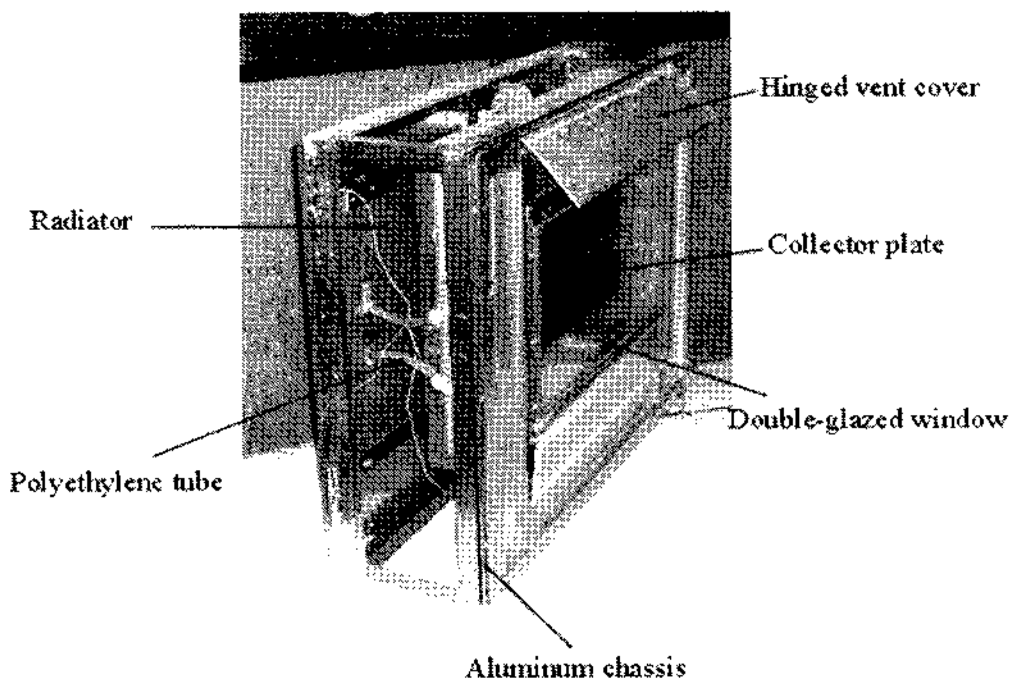

Fig. 2. An oblique view of the present bi-directional thermodiode

leakage. The copper tubes were soldered to 0.0004m-thick copper sheets with a selective paint on their outer surfaces to enhance radiation absorption. Water or a mixture of water and an antifreeze was chosen to be the working fluid.

The metallic panels facing the indoors or the outdoors served as the collector or the radiator for thermal radiation. The rectangular loops were installed between the collector and the radiator, with their horizontal sections in contact with the panels. Moving either the radiator or the collector surface vertically changes the direction of heat flow and the buoyancy of the induced natural convection. The space between the collector and the radiator was filled with glass wool so that the diode system could also function as a regular insulating wall with minimum heat leakage.

Fig. 2 is the oblique view of a thermodiode module built in the present study. The exterior of this module consists of an aluminum chassis, 0.003 m-thick aluminum plates, sus meshes that covers the vent openings to prevent large debris from entering the thermodiode interior, and glazed windowpanes. Wooden panels of 0.006 m thickness were used to seal the back and sides of the aluminum chassis. The interior of the module was filled with glass wool and polystyrene insulation to hinder leakage of heat. The outer diameter of the copper tubes was 0.0127 m with a wall thickness of 0.001 m.

At the rear end of this thermodiode, a slab-shaped radiator joins polyethylene tubes whose thickness is 0.02 m. This radiator can accommodate pressure variation in the working fluid during the heating mode, which might cause some leakage at loosened joints due to excessive pressure buildup within the circulation loop.

3. ANALYTICAL MODELING OF DIODE OPERATION BASED ON EXPERIMENTAL DATA

3-1 Componental Model

A simple analytical model was developed to analyze the diode(with a slab shaped radiator) behavior and performance. The model allows prediction and estimation of diode temperature variations and heat transfer rates cTo reduce the complexity to a manageable level, the average temperatures of individual components were used in the analysis. Assumptions such as constant fluid properties(except for the expansion coefficient in buoyancy force calculations) and constant and uniform heat transfer coefficients or thermal resistances were also made in the model.

In the following analysis subscripts a, r, t, s, and w refer to the ambient air, radiator, copper tubes, copper sheets, and water in the tubes, respectively. The time-dependent heat transfer process is divided into three parts: a start-up phase during which the copper sheets and tubes are heated up very rapidly by the incidence thermal radiation, but natural convection throughout the loops has not been induced yet; a heat-up phase during which heat is transferred primarily by natural convection between the collector and the radiator, and a cool-down phase after the radiative heating ceases. Strictly speaking, there is a fourth part of the heat transfer process. It is the short period between the start-up and the heat-up phases in which the natural convection throughout the loops is first induced. The hot fluid in the tubes begin to enter the radiator and the cold fluid in the radiator begin to enter the tubes, causing the collector temperature to drop and the radiator inlet temperature to rise sharply. This process is very difficult to model analytically and is more suitable for a three-dimensional, time-dependent numerical simulation. Since this process lasts only a few minutes, it is not included in the present analytical study.

During the start-up phase the buoyancy force of water in the heated copper tubes is too weak to penetrate to the radiator. Without a bulk flow in the axial direction, the heat transfer rate from the copper tubes to the fluid in the tubes is very low. The resistance(and also the temperature difference) to heat flow from the copper tube walls to the fluid inside is therefore much higher than that from the copper sheets to the copper tubes. As a result the copper sheet and copper tube temperatures(T_s and T_t) are assumed to be the same in the start-up phase. The energy equations for this phase are: For the copper sheets and tubes:

$$[(\rho cV)_s + (\rho cV)_t]\, dT_t/dt = Q_{in} - Q_{s\text{-}a} - Q_{t\text{-}w} \qquad (1)$$

For water in the copper and polyethylene tubes:

$$(\rho cV)_w\ dT_w/dt = Q_{t\text{-}w} \qquad (2)$$

where ρ, c, V, t, and Q denote density, specific heat, volume, time, and heat transfer rate, respectively.

The rate of incident radiation absorbed by the copper sheets is calculated from:

$$Q_{in} = q_{rad}\ \alpha\tau^2 A_s \qquad (3)$$

where q_{rad} is the radiation flux on the diode surface. α and τ are the absorptivity of the paint on the collector surface and the transmissivity of the glass panels. Average values ($\alpha = 0.85$ and $\tau = 0.85$) of the radiative properties provided by the manufacturers and/or measured by KIER (Korean Institute of Energy Research) were used in our analysis.

Laminar convection correlations of the form:

$$Nu = hL/k = C\ Ra^{1/4} \qquad (4)$$

were first tried to model the heat transfer from the copper sheets to the surroundings($Q_{s\text{-}a}$) and from the copper tubes to the fluid in the copper and polyethylene tubes($Q_{t\text{-}w}$).

It was found however that the copper

tube temperature variation could be modeled more accurately if the two heat transfer terms were calculated from conduction-like relations:

$$Q_{s-a} = (T_s - T_a)/R_{s-a} \tag{5}$$
$$Q_{t-w} = (T_t - T_w)/R_{t-w} \tag{6}$$

Using the experimental results of 600 W/m^2 radiation flux, the two thermal resistances were found to be approximately $R_{s-a} = 0.888$ and $R_{t-w} = 0.393$ W/ °C .

The above analytical model with thermal resistances determined from the 600 W experiment predicts the water temperature at the end of the start-up phase(which should be very close to the measured radiator inlet temperature at the onset of natural convection throughout the loop) within 1 °C for input radiation of 400, 600, and 800 W/m^2. The computed T_t also agrees well with the measured collector temperature variations, with the exception of the last stage of the start-up phase for the 400 W case in which the model under predicts the collector temperature by about 4 °C. The start-up phase ends when the temperature difference between the fluid in the tubes and the radiator reaches about 17 °C.

During the heat-up phase, the energy equation for the radiator temperature can be expressed as:

$$(\rho c V)_r \, dT_r /dt = Q_{t-r} - Q_{r-a} \tag{7}$$

The heat loss term consists of natural convection and radiation losses of the indoor-facing surface:

$Q_{r-a} = H_r W_r$ (height x width of the indoor facing surface)

$$[h_r (T_r - T_a) + \sigma \varepsilon_r (T_r^4 - T_a^4)] \tag{8}$$

The emissivity of the dark gray paint on the indoor facing surface is 0.84. The convection coefficient h_r was determined from laminar natural convection on a vertical isothermal plate:

$$h_r H_r /k_a = C_r Ra_r^{1/4} = 0.59 Ra_r^{1/4} \tag{9}$$

Since the Nusselt number for laminar natural convection generally depends on the ¼ th power of the Rayleigh number, the heat transfer rate from the heated copper tubes to the radiator was assumed to be :

$$Q_{t-r} = C_t (T_t - T_r)^{5/4} \tag{10}$$

Experimental results of 3 radiation fluxes indicate a nearly constant temperature difference, $T_t - T_r$, during the heat-up phase. For the diode we tested, this temperature difference can be approximately estimated from:

$$T_t - T_r \text{ (in °C)} = (\text{radiation flux on the diode surface, in } W/m^2/34)^{4/5} \tag{11}$$

The above relation allows determination

of the copper tube temperature at the beginning of the heat-up phase from the radiator initial temperature and the input radiation flux.

Using the results of the 600 W experiment and Eq.(11), C_t was determined to be around 1.55 and Q_{t-r} was about 27.35 W. Since $T_t - T_r$ remained nearly constant during the heat - up phase, the effectiveness of the diode can be determined from :

Diode effectiveness = Q_{t-r}/(Conduction through the insulating materials and the stagnant water in the four polyethylene tubes for the same temperature difference) = 93 for 600 W/m^2 radiation input. (12)

The copper tube temperature, which should be very close to the temperature measured by the thermocouple embedded in the collector plate, can be determined from the following energy equation:

$$(\rho cV)_t \; dT_t /dt = Q_{s-t} - Q_{t-r} \qquad (13)$$

Since Q_{t-r} remained nearly constant and the variation of T_t was observed to be almost linear during the heat-up phase, the average value of Q_{s-t} was employed in our analysis. Matching the theoretical solution for T_t with the experimental result of 600 W, we found:

$$Q_{s-t,\ ave} = 27.6 \text{ W} \qquad (14)$$

Using the average heat transfer rates for Q_{s-a} and Q_{s-t}, the energy equation for the copper sheets during the heat-up phase can be expressed as:

$$(\rho cV)_s \; dT_t /dt = q_{rad}\alpha\tau^2 A_s - Q_{s-a,\ ave} - Q_{s-t,\ ave} \qquad (15)$$

The mass of the copper sheets is very small. The left-hand-side of Eq.(15) is therefore close to zero. As a result the average heat loss rate of the copper sheets is approximately:

$$Q_{s-a,\ ave} = q_{rad}\, \alpha\tau^2 A_s - Q_{s-t,\ ave} \qquad (16)$$

During the cool-down phase the radiator temperature is calculated from:

$$(\rho cV)_r \;\; dT_r /dt = - Q_{r-a.cool\text{-}down} \qquad (17)$$

The equation for the heat loss term is identical to Eq.(8), but the convection coefficient was determined from experimental data. The C_r value that matched the cool-down phase data of 600 W was found to be around 1.5. This value is much higher than that of natural convection on an isothermal vertical plate. Possible reasons for the difference are:

(1) The first term on the right hand side of the energy equation in fact includes natural convection on the indoor-facing surface as well as heat losses from the back and sides of the radiator during the cool-down

phase.

(2) The thermocouple that measured the radiator surface temperature was only attached to, not embedded into the copper plate. The temperatures it measured therefore were lower than the average radiator temperature due to exposure of the thermocouple bead to the airflow. Experimental measurements during the heat-up phase show that the radiator temperature was a few degrees lower than the arithmetic average of the radiator inlet and outlet temperatures. Since the temperature difference over most of the cool-down phase was small, a few degrees difference in T_r measurement results in a large decrease in Cr value.

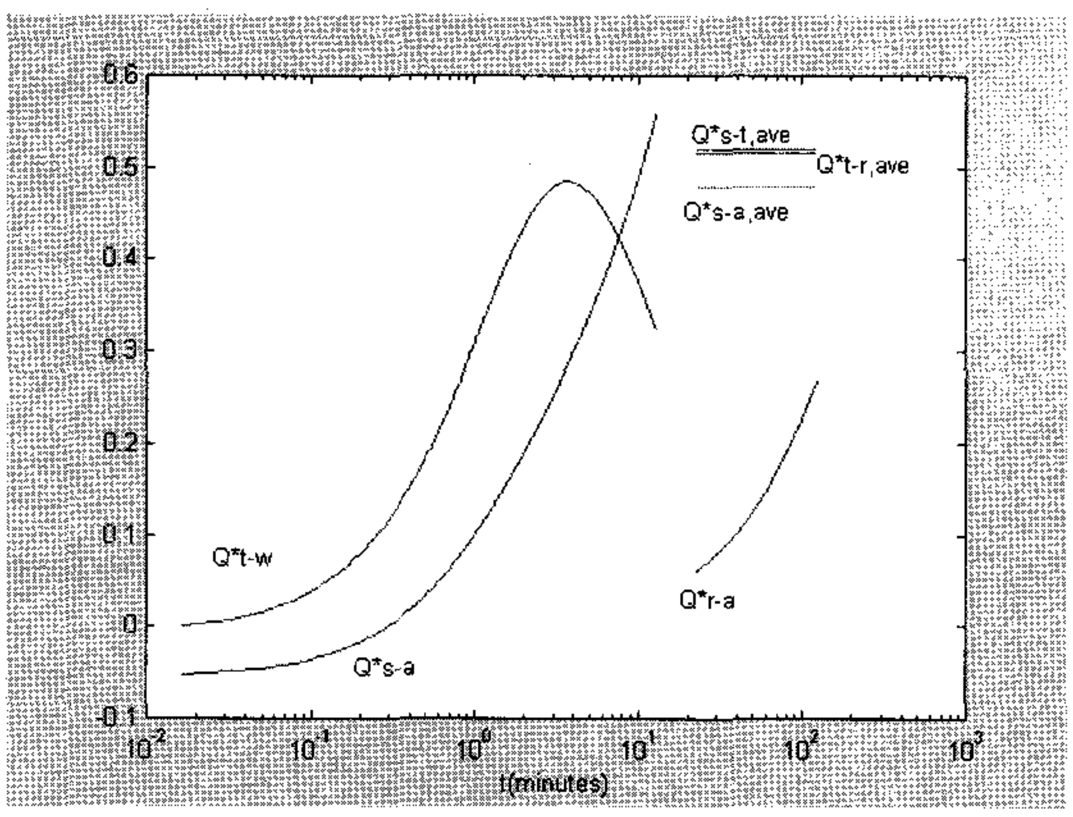

Fig. 3 Normalized heat flows during heat-up phase

Results of the analytical model applied to the case of 600 W/m^2 are presented in Fig. 3. Heat transfer rates in this figure were normalized by the radiation absorbed by the copper sheets, which is only 61% ($= \alpha\tau^2$ in Eq.(3)) of the radiation incident on the diode surface. During the start-up phase the rate of energy transfer from the heated copper tubes to the fluid in the tubes first increases faster than the rate of heat loss to the surroundings. However the later increases continuously while the former reached a maximum value of about 50 % before dropping to 33 % at the end of the start-up phase. In the heat-up period about 50% of the energy absorbed by the copper sheets is transferred to the copper tubes, and then to the radiator by natural convection throughout the diode. The rest is lost primarily from the collector surface to its surroundings. The heat transfer rate from the radiator to the indoors increases steadily during the heat-up phase. The normalized Q_{r-a} reaches 28 % after 2 hours of heating.

3-2 Lumped Model

When the bi-directional thermodiode was forward-biased, it had a short start-up period. During this period the buoyancy force of the working fluid in the heating section was insufficient for natural convection to be induced throughout the loop. Small natural convection cells first appeared in the heating section, then penetrated to the branches connecting the heating and cooling sections. Before the onset of throughflow in the loop, it is very difficult to obtain an analytical solution for the flow and heat transfer in the diode. However this start-up phase of our thermodiode system lasted only about 15 minutes. After that the simple analytical model outlined below can predict the time variation of the average water temperature during the heat-up and cool-down phases.

If the thermodiode is simplified to a lumped system, the time variation of the average water temperature, T, can be calculated from:

$$[(\rho cV)_{water} + (\rho cV)_{copper}]\ dT/dt = Q_{in} - Q_{loss} - Q_{out} \quad (18)$$

where ρ, c, and V are the density, specific heat, and total volume of the water in the diode or the copper tubes and sheets. Q_{in} is the rate of radiation absorbed by the collector surface. Q_{out} includes natural convection and radiation heat transfer from the radiator surface to the indoors. Heat losses from the diode surfaces other than the radiator surface are included in Q_{loss}.

The rate of radiation absorbed by the collector surface can be calculated from:

$$Q_{in} = q_{rad}\ \alpha\tau^{2}\ A_{collector} \quad (19)$$

where q_{rad} is the radiation flux on the diode surface, and α and τ the absorptivity of the paint on the collector surface and the transmissivity of the glass panels. According to manufacturers specifications and the tests performed at KIER (Korean Institute of Energy Research), the average values of the above radiative properties are:

$$\alpha = 0.85,\ \text{and}\ \tau = 0.85 \quad (20)$$

The heat loss term was approximated by a natural convection relation of the form:

$$Q_{loss} = h\ A_{diode}\ (T - T_{ambient}) = (T - T_{ambient})^{1.25}/R_{convection} \quad (21)$$

A convection resistance of 1.2 W/°C was found to fit the experimental data of 800 W/m^2 quite well.

The emissivity of the dark gray paint on the radiator surface, ε, is about 0.84. For laminar natural convection on a vertical isothermal plate, the average heat transfer coefficient can be calculated from Incropera and DeWitt[7]:

$$h = 0.59\ (k/H)\ Ra_H^{\ 0.25} \quad (22)$$

In the above equation k is the thermal conductivity of the indoor air, H the height of the radiator surface, and Ra_H the Rayleigh number. The rate of heat transfer from the radiator to the indoors was calculated from:

$$Q_{out} = A_{radiator}\ [h\ (T - T_{indoor}) + \sigma\varepsilon\ (T^4 - T_{indoor}^{\ 4})] \quad (23)$$

Equation (1) can also be used to predict water temperature variation during the cool-down phase by dropping the heat input term Q_{in}. A comparison of the analytical solution and the experimental results of 800 W/m^2 is given in Fig. 4. Considering the fact that the analytical model described above is rather a very simple one, the agreement between these two is quite remarkable. After the first 20 minutes of heating, the calculated water temperatures are distributed fairly close to the average of radiator inlet and outlet temperatures during heat-up. Its rate of increase is almost identical with the measured ones. This trend is also observed during the cool-down phase, Fig. 5,

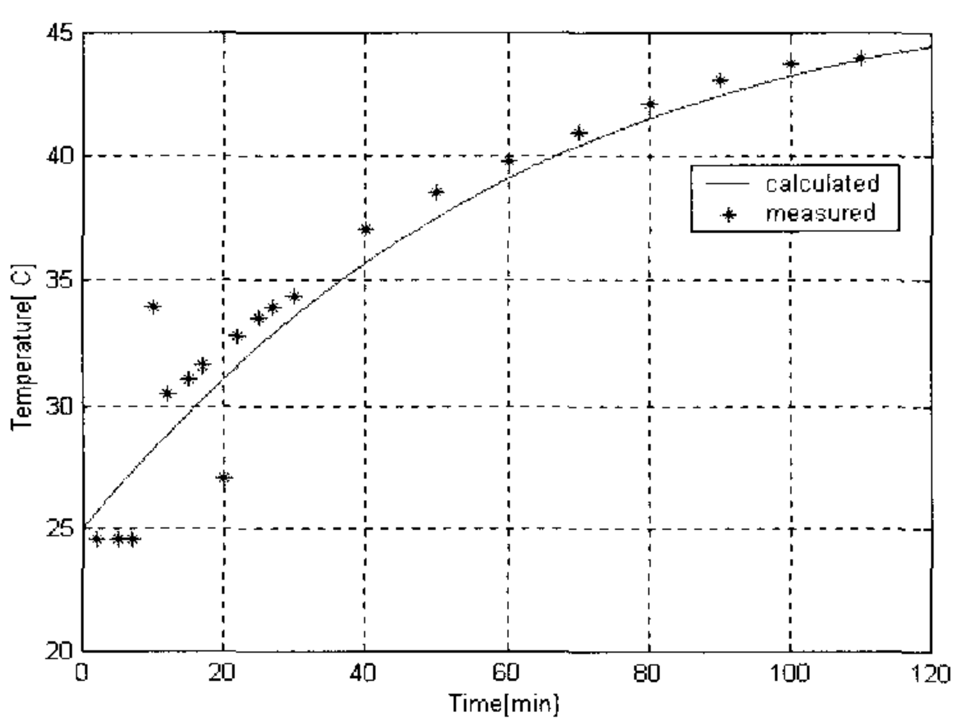

Fig. 4. A comparison of the analytical model and experimental results(heating phase)

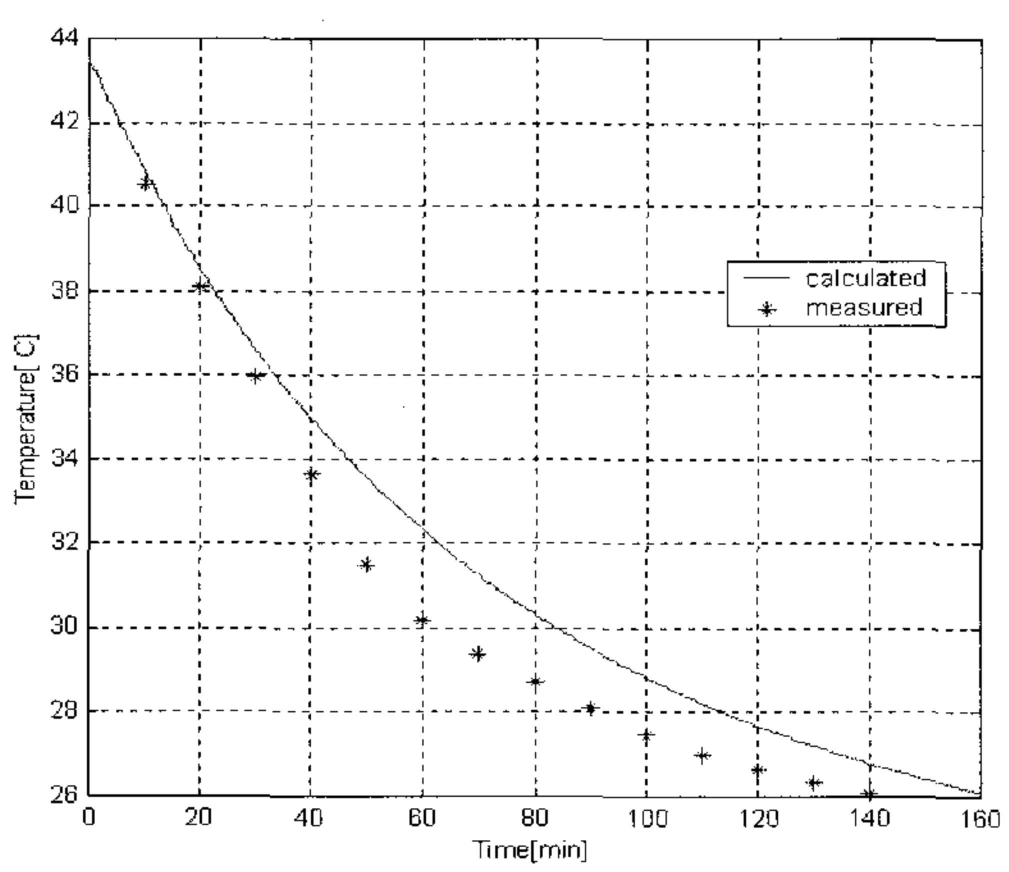

Fig. 5. A comparison of the analytical model and experimental results(cooling phase)

where the calculated water temperature shows a monotonic decrease from the beginning as time elapses. The good agreement between the analytical solution and experimental data validates the accuracy of the simple analytical model for heat transfer analysis of the present thermodiode system.

4. Conclusions

A theoretical study was carried out to investigate the heat transfer characteristics of a new bi-directional thermodiode design. The thermal diode is made using several rectangular loops filled with a fluid(water) and mounted between two vertical plates. Rotatable joints are used between the horizontal and inclined segments of the loops to alter the direction of heat transfer, when required. Two simple one-dimensional analytical models were developed to analyze the thermal diode(with a slab shaped radiator) behavior and performance. Of these, one presents the heat flows between different system components whereas the other is a lumped model to resolve the time variation of the average water temperature in the diode.

REFERENCES

1. Chen, K., Shorthill, R. W., Chu, S. S., Chailapo, P., Narasimhan, S., An Energy-Efficient Construction Module of Variable Direction of Heat Flow, Heat Capacity, and Surface Absorptivity, USAF

Report WL-TR-95, 3046, 1995.

2. Chen, K., Chailapo, P., Chun, W. G., Kim, S., and Lee., K. J., The Dynamic Behavior of a Bayonet-Type Thermal Diode, Solar Energy, Vol. 64, Nos. 4-6, pp. 257-263, 1998.
3. Jones, G. F., Heat Transfer in a Liquid Convective Diode, ASME Journal of Solar Energy Engineering, Vol. 108, pp. 163-171, 1986.
4. Chun, W. G., Lee, Y. J., Lee, J. Y., Chen, K., Kim, H. T., and Lee, T. K., "Application of the Thermal Diode Concept for the Utilization of Solar Energy," Proceedings of the 31st Inter-society Energy Conversion and Engineering Conference, Vol. 3, pp. 1709~1714, 1996.
5. Chun, W., Kang. Y. H., Kwak, H. Y. and Lee, Y. S., "An Experimental Study on the Utilization of Heat Pipes for Solar Water Heaters," *Applied Thermal Engineering*, Vol. 19, pp. 807 ~817, 1999.
6. K. J. Lee and Chun, W., "Development of Energy Efficient Smart Module with Variable Direction of Heat Flow," Heat Capacity and Surface Absorptivity, Vol. 18, No. 3 pp. 119~128, 1998.
7. Incropera, F. P. and DeWitt, D. P., *Fundamentals of Heat and Mass Transfer*, 4th edn, pp. 492-494. John Wiley & Sons, Inc., New York, NY, 1996.

E-mail : wgchun@cheju.ac.kr