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수직이중관형 잠열축열장치의 성능분석

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Performance Analysis of a Vertical Double Pipe Heat Exchanger for Latent Heat Storage

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요 약

고밀도 잠열축열장치의 최적설계와 효율적인 작동을 위해서는 그 전열특성과 축열효율이 규명되어야 한다. 본 연구에서는 수직이중관형 잠열축열장치의 방열과정에서의 전열특성을 이론 및 실험적으로 분석하였으며 두 결과는 잘 일치하였다. 그리고 컴퓨터 시뮬레이션을 통하여 방열효율에 대한 설계 및 작동파라미터의 영향을 분석하였다.

ABSTRACT

For the optimal design and the efficient operation of the double pipe type latent heat storage equipment, the effect of the parameters of the system were analysed. The statistical analysis showed that the theoretical and the experimental results of the volume change rate and the temperature variations were well agreed. Therfore, this theoretical model is reasonable to analyze two dimensional moving boundary problems.

In the analysis of the effects of the parameters, the heat extraction fraction and the water outlet temperature of the system as function of the time were analysed depending on the initial temperature of PCM, water inlet temperature, water mass flow rate and the dimension of the inner tube.

I. Introduction

For the efficient utilizaton of the solar thermal energy, an efficient heat strorage technique, especially high density heat storage system is necessary. Among the several heat storage methods, latent heat storage system has some advantages that it can store higher density heat at constant temperature.

For the optimal design of the latent heat storage system by using the PCM and efficient operation of the system, heat transfer characteristics of latent heat storage systems have to be analyzed.

In this study, the effect of parameters for the optimal design and operation of the double pipe type latent heat storage equipment were analyzed. The theoretical and the experimental analysis of the heat transfer characteristics of the system during heat discharging process were already performed by us[1,4]

II. Theroretical Analysis

The latent heat storage system for the computer simulation and the experiments is shown in Figure 1, which is the double pipe heat exchanger. The phase change material, i.e. calcium chloride hexahydrate, is filled in the outside of the inner tube. And water flows through the inner tube inside.

In this case, the temperatures of water and the phase change material vary depending on time and distance during heat charging and discharging process.

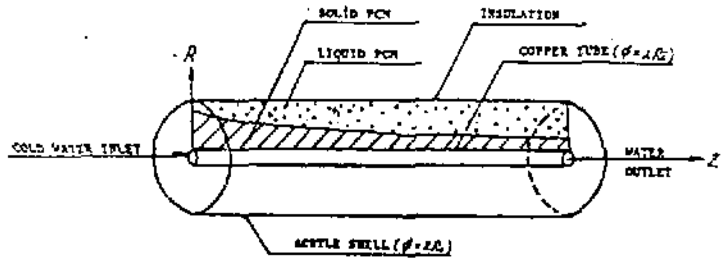

Fig. 1 Configuration of the double pipe heat exchanger

For the optimum operations and the design of the system, the temperature variation, the location of the moving boundary interface, the heat storage efficiency and the solidification rate, et cetera, depending on time during heat charging and discharging process have to be analyzed.

The computer simulation starts with setting of the governing equation, i.e. energy equations of the phase change material and the coolant. And the set of the initial conditions and the boundary conditions, setting of the finite difference equations by the finite difference method, solving the coefficients matrix, calculating the temperature history depending on time, solidification rate and the heat storage efficiency by the numerical integral, et cetera, are performed and calculated.[1]

After taking the energy balance equation about the finite control volume in the tube, the governing equation in the inner tube can be expressed as follows.

$$\frac{\partial Tw}{\partial t} = -\frac{\partial Tw}{\partial Z} \cdot U + \frac{Kw}{\rho_w \cdot Cpw} \cdot \frac{\partial^2 Tw}{\partial Z^2} + \frac{2 \cdot Hw}{\rho_w \cdot Cpw \cdot Ri} \cdot (Tt - Tw) \tag{1}$$

The governing equations of the phase change material consisted of two parts of the liquid phase and the solid phase because the thermophysical properties of the phase change material are different at each phase.

The basic equations can be expressed as follows

$$\text{Liquid region:} \ \rho l \cdot Cpl \cdot \frac{\partial Tl}{\partial t} = Kl \cdot \left(\frac{1}{R} \cdot \frac{\partial Tl}{\partial R} + \frac{\partial^2 Tl}{\partial R^2} + \frac{\partial^2 Tl}{\partial Z^2}\right) \tag{2}$$

$$\text{Solid region:} \ \rho_s \cdot Cps \cdot \frac{\partial T_s}{\partial t} = Ks \cdot \left(\frac{1}{R} \cdot \frac{\partial T_s}{\partial R} + \frac{\partial^2 Ts}{\partial R^2} + \frac{\partial^2 T_s}{\partial Z^2}\right) \tag{3}$$

The governing equations need the initial conditions and the boundary conditions for the unique solution of the problem. The initial and the boundary conditions are summarized as follow.

Initial conditions ; $t=0$ (4)

$Ri<R<Ro$: $Tl=Ti$

$Z=0$: $Tw=Twin$, $\partial Tl/\partial Z=0$

$R=Ri$: $Tt=Ti$,

$Kl*(\partial Tl/\partial R)=Hw*(Tt-Tw)$

$R=Ro$: $\partial Tl/\partial R=0$

$Z=Zmax$: $\partial Tl/\partial Z=0$

Boundary conditions ; $t>0$ (5)

$Z=0$: $Tw=Twin$, $\partial Ts/\partial Z=0$

$\partial Tl/\partial Z=0$

$Z=Zmax$: $\partial Ts/\partial Tl/\partial Z=0$

$R=R_0$: $\partial Tl/\partial R=0$

$R=Ri$: $Ks*(\partial Ts/\partial R)=Hw*(Tt-Tw)$

$R=Rm$: $Ts=Tl=Tm$

$$Kl\cdot\left(\frac{\partial T}{\partial R}\cdot Alr+\frac{\partial T}{\partial Z}\cdot Alz\right)-$$
$$Ks\cdot\left(\frac{\partial Ts}{\partial R}\cdot Asr+\frac{\partial Ts}{\partial Z}\cdot Asz\right)$$
$$=\rho m\cdot Hf\cdot Amr\cdot\frac{d\delta s}{dt}$$

Ⅲ. Experiments

For the two dimensional analysis of the double pipe heat exchanger for latent heat storage, the experimental equipments were constructed as shown in Figure 2.

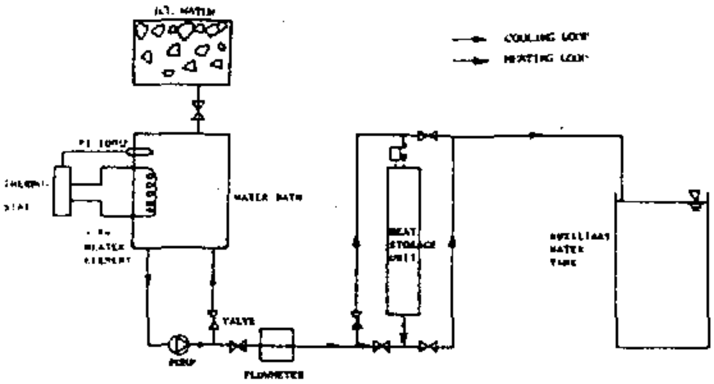

Fig. 2 Schematic diagram of the experimental system

The tube is made of copper which thermal resistance is low, and its thickness is 1mm and outer diameter is 10mm. The outer pipe is made of acrylic resin whose thermal conductivity is similar to that of calcium chlodride hexahydrate and helpful for the insulation. The dimension of the outer pipe is taken as the thickness of 3mm, the outer diameter of 100mm and the height of 300mm, the total volume was 2080 cubic centimeter.

The auxiliary container, whose inner diameter is 3mm, is installed on the upper part of the shell, which is purposed for the volume expansion of the phase change material during the phase change and checking the solidification rate. The auxiliary container may be installed on the side of the shell in the horizontal shell and tube type[2].

In this case, the temperature of the phase change material may be failed to be axisymmetric distribution.

The density difference between the solid phase and the liquid phase state causes the volume expansion during the heat charging process and this gives stresses to the system if there were not for the auxiliary container.

The mineral oil, which is lighter than the phase change material and not reacted with it, is filled in the auxiliary container and therefore, forms free surface.

The solidification rate can be calculated by checking the height of the mineral oil. A drain cock is installed at the bottom of the outer pipe. The outside of the container is efficiently insulated by the insulation material (Toilon).

The temperature measurement system is consisted of E-type thermocouple and the reader.

The phase change material for the experiments is calcium chloride hexahydrate which is imported from Switzland. It is regent grade and the melting point is 28℃. The thermophysical properties are studied at the former part of this paper. The density of the solid phase were 1,415 kg/m³ and that of the liquid phase is 1,680 kg/m³, which are measured. The weight and the volume of the phase change material were measured accurately for the density.

The mineral oil is used for the compensation of the air hole which would be arised in the solid state.

Although the specific heat and the latent heat capacity may be measured by the energy conservation law, most of the properties are referred to the reference(3) because the material is regent grade.

The experiment is started with melting the phase change material at 37℃ which is higher than the melting point.

The liquid material is poured cautiously in the PCM container.

The fine straw has to be used for the air ventilating because the container system is a closed system without the entrance of the auxiliary mineral oil container.

In a few minutes for the temperature regulation, the coolant is supplied through the inner tube from the constant temperature bath. The temperature of the coolant is 7℃, which is produced in the water bath.

By releasing the valve, the coolant flows through the inner tube from the lower part to the upper part of the tube. The direction of the flow has to be considered because the solidification from the lower part can give the free surface on the upper part of the tube.

The solidification from the upper part may arise the vacuum in the PCM container because the center of the system would be empty by the volume depression caused by the density difference between the solid and the liquid phase.

Also the water flow direction of the heat charging process has to be from the upper part to the lower part of the tube.

If the phase change material were liquified from the lower part of the tube, the volume expansion from the density change with the phase change would give the stresses to the system.

From the instant of the valve releasing, the temperature of the phase change material, the coolant and the height of the mineral oil in the auxiliary container were measured at every five minutes.

Ⅳ. Results and discussion

1. Comparison of the theoretical predictions and the experimental results

The experiment was performed to find the heat transfer characteristics of the latent heat storage system during heat discharging process and for the validation of the theoretical model and the numerical analysis. The first validation can be done by the temperature history of the phase change material, i.e. calcium chloride hexahydrate. The temperature histories of the four points in the phase change material are shown in Figure 3.

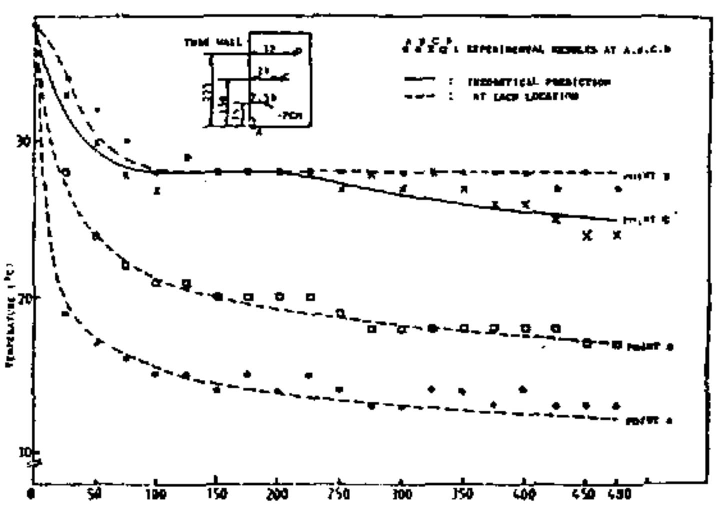

Fig. 3 Experimental and theoretical temperature history of the PCM as function of time on two dimensional system

The experimental results and the theoretical predictions were conformable.

The chi-square test showed that of the experimental data were well agreed with the theoretical data at the 0.05 level of significance.

The effects of the latent heat capacity were more clearly seen at long distance from the inner tube wall.

The duration of the latent heat discharging period was about two hours at the C point.

The second validation was done by the solidification rate of the PCM depending on time. The solidification rate of phase change material is defined as follow.

Solidification rate = (Solidification volumex density of solid PCM) / (Total weight of PCM)

The solidification rate of the experiment was well agreed with that of the theoretical analysis[4]. Therefore, the theoretical model and the numerical analysis for the doublepipe heat exchanger in this study for the latent heat storage are resonable.

2. Effect of parameters

For the optimal design and operation of the latent heat storage system, the effect of the the system have to be investigated.

For this purpose, the effects of several parameters on the double pipe heat exchanger for the latent heat storage are studied.

The parameters which have to be analysed are initial temperature of PCM, water inlet temperature, water mass flow rate, diameter and length of tube.

The basic conditions for simulation are as follows: initial temperature of PCM, Ti=35℃, water inlet temperature, Twin=15℃, water mass flow rate, $\dot{m}$=0.075 ㎏ / sec, diameter of tube, Di=40 ㎝, length of tube, Zmax=1m.

Each of the above mentioned parameters corresponds with the dimensionless parameter of Stefan number, Stanton number, aspect ratio.

One of the object parameter of the system is the heat discharging rate. According to the purpose of the system, the object parameter may be the heat extraction fracton, water outlet temperature, et cetera.

2.1 Initial temperature of PCM

Two levels of initial temperature, Ti=45℃ and Ti=30℃, were taken of the analysis of its effect to the system performance. The heat discharging rate, which is the rate of discharged heat quantity to the maximum discharging one, was not very dependent upon the initial temperature of the PCM as shown in Figure 4.

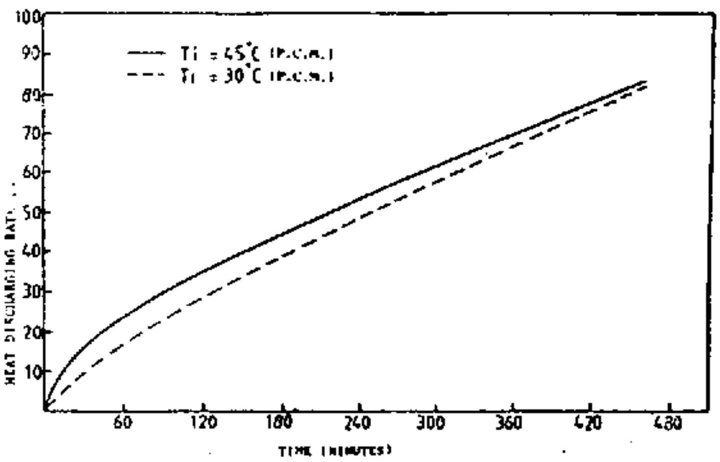

Fig. 4 Heat discharging rate vs.time as function of initial temperature of PCM, Ti

Even though the initial temperature was considerablely high, it did not influence the heat discharging rate.

And the solidification rate of the case of Ti=30℃ became 100% in about 450 minutes.

About 80% of the maximum dischargeable heat was discharged in 7.5 hours at both temprature.

But the amount of the discharged latent heat of the case of the initial temperature 30℃ was bigger than that of the initial temperature 45℃

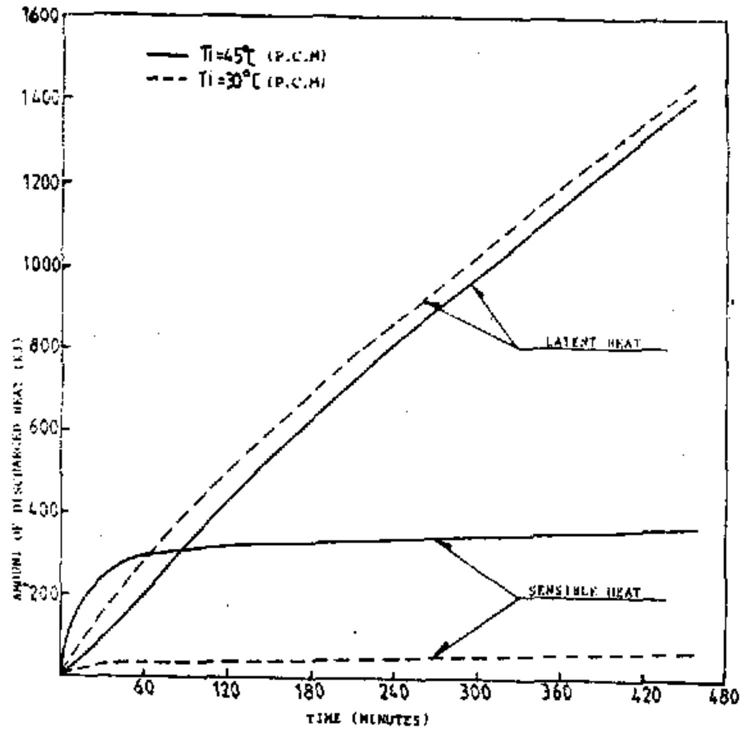

Fig. 5 Amount of the discharged heat as function of time on the reference conditions

as shown in Figure 5.

Also in both cases, the amount of the discharged latent heat was bigger than that of the sensible heat.

The sensible heat was increased very rapidly at beginning of discharging process and became nearly constant in about an hour.

Here, the effect of the latent heat storage was obvious.

Therefore, the initial temperature of PCM was recommended to be near the melting point in the latent heat storage system.

2.2 Water inlet temperature

Depending on water inlet temperature, the heat discharging rates were considerablely different as shown in Figure 6.

When the inlet temperatrue of water was 10℃, the solidification rate was nearly 100% and the heat discharging rate was 83% in 5.5 hours.

In the case of Twin=15℃, the solidification rate was nearly 100% and the heat discharging rate was approximately 85% in 480 minutes.

Therefore, heat discharging rate was higher at the lower inlet temperature.

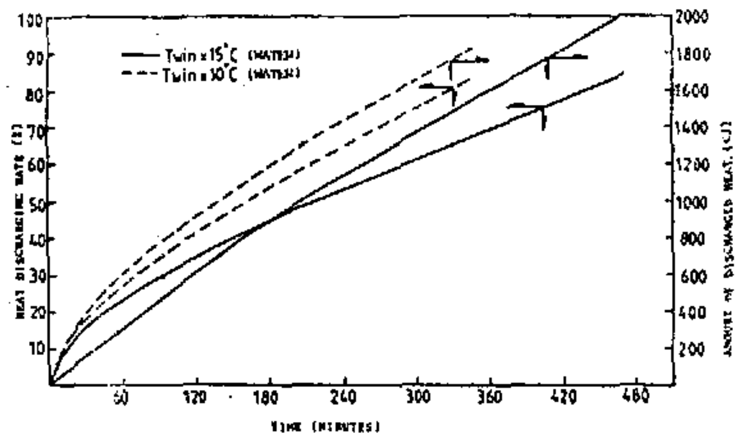

Fig. 6 Heat discharging rate and amount of discharged heat vs. time as function of the water inlet temperature

2.3 Water mass flow rate,

In order to find the effect of the water mass flow rate, both cases of 0.075 kg / s and 0.015 kg / s were compared. As shown in Figure 7, the heat discharging rate of the case of $\dot{m}=0.075$ kg / s was slightly higher than that of $\dot{m}=0.015$ kg / s.

The discharging heat of both cases were nearly same as 85% in 480 minutes. Therefore, the mass flow rate may not be so much impor-

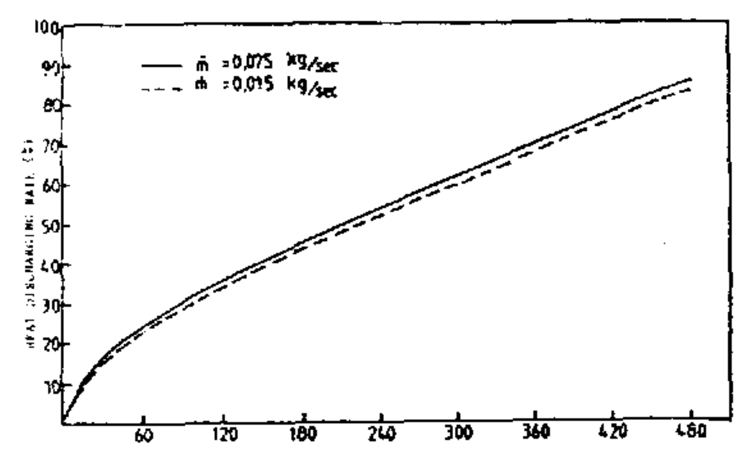

Fig. 7 Heat discharging rate vs. time as function of the water mass flow rate, $\dot{m}$

tant factor to the operation of the system.

The outlet temperature was considerablely dependent upon the water mass flow rate. When the water inlet temperature was 15℃ in both cases of $\dot{m}=0.075$ and $=0.015$kg / s, the outlet temperatures became 15.1℃ and 15.5 ℃ at each case.

Therefore, the smaller the mass flow rate became, the higher the water outlet temperature became.

2.4 Diameter of tube

The diameter of the tube is related to the surface area of the tube, which is the convection boundary of the PCM. As shown in Figure 8, heat discharging rate of the case of radius Ri =20 mm, was about 85% in 450 minutes and that of Ri=10 mm was about 55%.

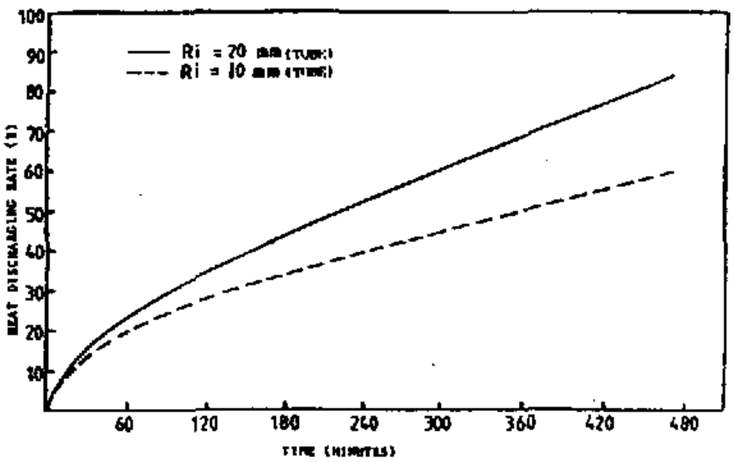

Fig. 8 Heat discharging rate vs, time as function of radius of tube, Ri

The solidification rate of the case of Ri=20 mm became nearly 100% in about 450 minutes, and that of Ri=10 mm became 65% as shown in Figure 9.

Although the amount of PCM in the case of Ri=10 mm was bigger than that of Ri=20 mm, the amount of discharged heat 1824 KJ of the case of Ri=20 mm was bigger than 1518KJ of the case of Ri=10 mm in 450 minutes.

Therefore, the convection boundary surface, by which heat was transferred from coolant or heating media to PCM, was recommended to be as bigger as possible.

Especially in the case of the simulation conditions, the radius of the tube was appropriate in viewpoint of the maximum utilization of the heat of PCM.

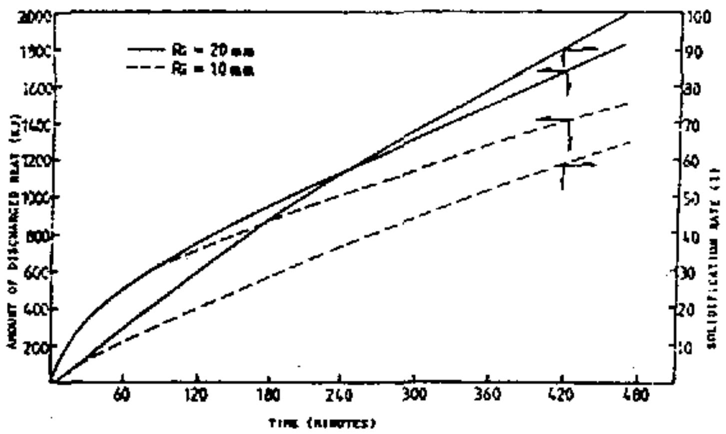

Fig. 9 Amount of discharged heat and solidification rate vs. time as function of the radius of the tube, Ri

2.5 length of heat storage unit.

The length of heat storage unit did not have big effect to the heat dicharging rate. In both cases of the length Zmax=1 m and 10 m, the heat dicharging rates were not so much different as shown in Figure 10.

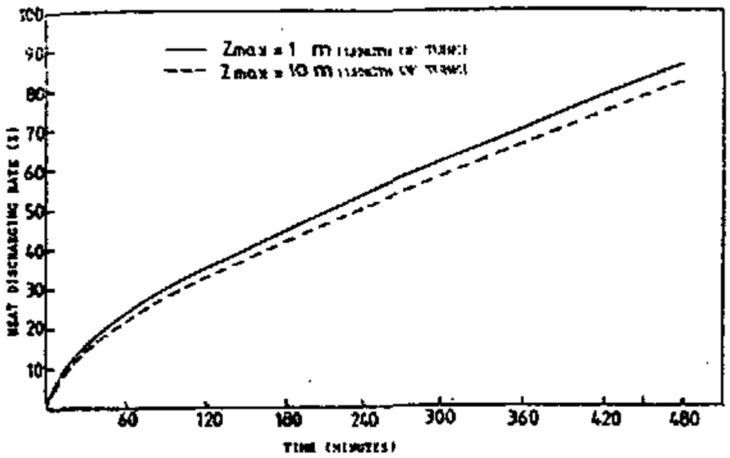

Fig. 10 Heat discharging rate vs. time as function of the length of the heat storage unit, Zmax

The amount of the discharged heat of the case of Zmax=10m was about 16900 KJ and that of Zmax=1m was about 1760 KJ in about 480 minutes.

Water outlet temperature is important when the heat storage system is used for air heating systems or domestic purpose.

When the water inlet temperature was 15℃ and the length of the tube was 1m, the outlet temperature became 15.1℃. When the length of the tube was 10m, it became 16.2℃.

The more the water mass flow rate became or the longer the system length became, the higher the water outlet temperature became in the heat discharging process.

V. Conclusion

For the optimal design and the efficient operation of the double pipe type latent heat storage equipment, the effect of the parameters were analyzed. By the theoretical and the experimental analysis, the results of the study can be summarized as follows.

1. The statistical analysis showed that the theoretical and the experimental results of the volume change rate and the temperature variations were well agreed.

Therefore, this theoretical model is reasonable to analyze two dimensional moving boundary problems.

2. In the analysis of the effect of the parameters,

1) Heat discharging rate was not so much dependent upon the initial temperature of PCM. The initial temperature was recommended to be near the melting point.

2) The lower the water inlet temperature became, the higher the heat discharging rate became.

3) Water mass flow rate was not seriously influent on the heat discharging rate. But the smaller the water mass flow rate became, the higher the water outlet temperature became.

4) In the analysis of the effect of the diameter, the convective boundary surface was recommended to be as bigger as possible to improve the heat discharging rate.

5) Heat discharging rate was not changed very much by the length of the heat storage system. But the longer the system length became, the higher the water outlet temperature became during the heat discharging process.

NOMENCLATURE

Alr, Alz, Amr, Ast, Asz : Area of the infinitesimal control volume

Cp	:	Specific heat
Hf	:	Latent heat capacity
Hw	:	Convection heat transfer coefficient
k	:	Thermal conductivity
R	:	Radius
Ri	:	Inner radius of the inner tube
Ro	:	Inner radius of the outer pipe
t	:	Time
T	:	Temperature of PCM
U	:	Water velocity
Z	:	Axial coordinate
Zmax	:	Length of the double pipe heat exchanger

Greek Symbols

δs	:	Distance of moving interface from the outer diameter of tube.
ρ	:	Density

Subscripts

i	:	initial or inner
l	:	liquid
m	:	melting
o	:	outer
s	:	solid
t	:	inner tube
w	:	water
win	:	water inlet

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