Heat Transfer Characteristics of a Latent Heat Storage Unit with a Finned Tube*

(2nd Report : Numerical Analysis of Solidification Process)

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A three-dimensional numerical analysis, in which the axial variations of the temperature of a heat exchanger fluid and a solidification layer are taken into consideration, is made to obtain detailed information on the heat transfer characteristics of a latent heat storage unit with a finned tube. The calculated results of the timewise variations of the heat flux and the solidification fraction are in good agreement with prior experimental data, so that the validity of the proposed numerical method is verified. Also, using this method, the effects of the thickness and the number of fins on the heat transfer characteristics are shown.

Key Words : Thermal Conduction, Phase Change, Numerical Analysis, Latent Heat Storage, Solidification, Finned Tube

1. Introduction

Since the thermal conductivities of phase change materials (abbreviated as PCM) used as heat storage materials of a latent heat storage unit are generally small, the heat flux abruptly decreases with increasing time. Among various methods proposed to eliminate this defect, the use of a finned tube is very effective, and several studies^{(1)~(5)} have been made. However, the effects of the finned tube on the heat transfer characteristics of the latent heat storage unit have not yet been well understood.

In the preceding report⁽⁶⁾, experiments were performed on the solidification and melting processes, and it was shown that the heat flux considerably increases if a finned tube is used. In this report, a three-dimensional calculation method is proposed, by which detailed information is obtained on the solidification process.

Aboul-Enein, et al.⁽⁴⁾ and Humphries⁽¹⁾ have performed two-dimensional analyses for latent heat storage units with longitudinal and lateral fins. However, the effects of important parameters, B_i (Biot number) and NTU (the number of transfer

units), on the heat transfer characteristics have not been shown. Saitoh, et al.⁽⁵⁾ recently proposed a new latent heat storage unit with fins in order to reduce the decrease of the heat flux, and they conducted experiments and an analysis on that unit, but the effects of B_i and NTU have not been clarified.

Shamsundar and Srinivasan⁽⁷⁾ have performed an analysis for a bare tube in which the axial temperature variation of the heat exchanger fluid is taken into account, and the effects of B_i and NTU have been clarified. According to their report, when appropriate values of B_i and NTU are selected, it is possible to keep the heat flux and the heat transfer effectiveness large. However, when a bare tube is used, the tube length has to be very large to obtain a high and constant heat flux since the thermal conductivities of PCM are very small.

The purpose of the present study is to clarify the effects of B_i and NTU on the heat transfer characteristics of a latent heat storage unit with a finned tube, and to obtain the data for the design of heat exchangers with high efficiency for latent heat storage. In this report, a three-dimensional numerical analysis is performed to obtain detailed data on the solidification process in which the heat flux is smaller than that in the melting process. The calculated results are compared with the previous experimental values⁽⁶⁾ and their validity is checked. Furthermore, using this method, the effects of the thickness and the number of fins are studied.

2. Basic equations and a calculation method

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Figure 1 shows the configuration and dimensions of a latent heat storage unit with a finned tube for which analyses were performed. A tube with 4 fins is representatively shown in the figure. The outer diameter, thickness and length of the copper tube are 45 mm, 4 mm, and 1 m, respectively. A shell made of an acrylic resin cylinder has an outer diameter of 105 mm and a thickness of 4 mm. PCM fills the space between the tube and the shell, and water is used as a heat exchanger fluid. As shown on the left side in Fig.1, longitudinal fins made of copper are soldered to the tube, and the calculations are conducted for various thicknesses δ and the numbers N_F of fins. When N_F is changed, δ is also varied in such manner that the mass of the fins becomes the same as that in the case of $N_F=4$ and $\delta=2$ mm. The physical properties of n-Eicosane $C_{20}H_{42}$ used as PCM are as follows (subscripts s and l denote solid and liquid phases of PCM, respectively).

Fusion temperature $T_f = 34^\circ\text{C}$ (measured value)

Latent heat $H = 2.45 \times 10^5 \text{ J/kg}$

Specific heat $\begin{cases} c_s = 2.25 \times 10^3 \text{ J/(kg}\cdot\text{K)} \\ c_l = 2.46 \times 10^3 \text{ J/(kg}\cdot\text{K)} \end{cases}$

Thermal conductivity $\begin{cases} \lambda_s = 0.313 \text{ W/(m}\cdot\text{K)} \\ \lambda_l = 0.151 \text{ W/(m}\cdot\text{K)} \end{cases}$

Density $\begin{cases} \rho_s = 853 \text{ kg/m}^3 \\ \rho_l = 777 \text{ kg/m}^3 \end{cases}$

A calculation domain and boundary conditions in k cross section are shown in Fig. 2 for the case of the tube with 4 fins. Assuming that symmetry condition are satisfied, the calculations are performed for the domain bounded by symmetry axes OA

($J=2$) and OB ($J=N_J+2$), an arc AB ($I=N_I+3$), and the inlet and outlet cross sections $K=2$ and $K=N_K+2$ in Fig.1. It is also assumed that the heat transfer is mainly through conduction except between the heat exchanger fluid and the tube. Considering the energy balance, basic equations are derived for control volumes $a \sim e$ representatively shown in Fig.2, and they are transformed into non-dimensional forms. First, for the control volume a [nodal point (i,j,k)] in PCM, the following non-dimensional finite difference equation is derived,

$$\frac{\Delta r'_{i+1}(r'_i + G_2/G_1 - \Delta r'_i/2)T'_{i-1,j,k} - (r'_i + G_2/G_1)(\Delta r'_i + \Delta r'_{i+1})T'_{i,j,k} + \Delta r'_i(r'_i + G_2/G_1 + \Delta r'_{i+1}/2)T'_{i+1,j,k}}{\Delta r'_i \Delta r'_{i+1}(\Delta r'_i + \Delta r'_{i+1})[r'_i + G_2/G_1 + (\Delta r'_{i+1} - \Delta r'_i)/4]/2} + \frac{\Delta \theta'_{j+1}T'_{i,j,k-1} - (\Delta \theta'_j + \Delta \theta'_{j+1})T'_{i,j,k} + \Delta \theta'_j T'_{i,j,k+1}}{\pi^2(r'_i + G_2/G_1)\Delta \theta'_j \Delta \theta'_{j+1}(\Delta \theta'_j + \Delta \theta'_{j+1})[r'_i + G_2/G_1 + (\Delta r'_{i+1} - \Delta r'_i)/4]/2} + \frac{T'_{i,j,k-1} - 2T'_{i,j,k} + T'_{i,j,k+1}}{(\Delta z'_k/G_1)^2} = P_1 \frac{\Delta T'_{i,j,k}}{\Delta F_0} \quad (1)$$

where π is the ratio of the circumference of a circle to its diameter and Δ is increment. The temperature T , the radius r , the angle θ , the axial distance z and the time t are non-dimensionalized as follows.

$$T'_{i,j,k} = \frac{T_{i,j,k} - T_f}{T_{in} - T_f}, \quad r'_i = \frac{r_i - r_{co}}{r_{ai} - r_{co}}, \quad \theta'_j = \frac{\theta_j}{\pi}, \quad z'_k = \frac{z_k}{L}, \quad F_0 = \frac{a_s}{(r_{ai} - r_{co})^2 t}$$

where T_{in} is an inlet temperature, r_{co} the outer radius of the tube and r_{ai} the inner radius of the shell. The non-dimensional parameters G_1 , G_2 and P_1 are defined as follows in which a_s and a_l denote the thermal diffusivities of solid and liquid PCM, respectively.

$$G_1 = \frac{r_{ai} - r_{co}}{L}, \quad G_2 = \frac{r_{co}}{L}, \quad P_1 = \frac{a_s}{a_l} \quad (\text{liquid region}) \text{ or } 1 \quad (\text{solid region})$$

Mesh sizes in r and θ -directions are non-uniform and small near the fin and the tube, but that in z -direction is uniform. The numbers of mesh, N_I , N_J and N_K , are 14, 6 ~ 12 and 6 in the directions r , θ and z , respectively. The time increment ΔF_0 is varied from 7.23×10^{-4} ($\Delta t = 3 \text{ s}$) to 1.45×10^{-2} ($\Delta t = 60 \text{ s}$). Next, the control volume b in Fig.2 representatively shows the volume containing a nodal point being adjacent to the fin. In this

Fig.1 Latent heat storage unit and calculation domain (finned tube with 4 fins)

Fig.2 Detailed calculation domain and boundary conditions

control volume, the equation is derived in the same way as Eq.(1) for the volume a , and the heat transfer rate $Q_{c,b}$ from the fin (volume c) to PCM (volume b) is derived from the following relation.

$$Q_{cb} = \frac{\lambda}{r_i \left[\Delta\theta_3 - \sin^{-1} \left(\frac{\delta}{2r_i} \right) \right]} \frac{\Delta r_i + \Delta r_{i+1}}{2} \Delta z_k (T_{i,2,k} - T_{i,3,k}) \dots \dots \dots (2)$$

where δ is the thickness of the fin and λ the thermal conductivity of PCM in the volume b . The equation in the fin (volume c) is expressed as follows.

$$\begin{aligned} & \frac{\Delta r'_{i+1} T'_{i-1,2,k} - (\Delta r'_i + \Delta r'_{i+1}) T'_{i,2,k} + \Delta r'_i T'_{i+1,2,k}}{\Delta r'_i \Delta r'_{i+1} \frac{\Delta r'_i + \Delta r'_{i+1}}{2}} + \frac{P_5 (T'_{i,3,k} - T'_{i,2,k})}{\frac{G_3}{2G_1} \left(r'_i + \frac{G_2}{G_1} \right) \left[\pi \Delta\theta'_3 - \sin^{-1} \left\{ \frac{G_3/G_1}{2(r'_i + G_2/G_1)} \right\} \right]} \\ & + \frac{T'_{i,2,k-1} - 2T'_{i,2,k} + T'_{i,2,k+1}}{(\Delta z'_k/G_1)^2} = P_2 \frac{\Delta T'_{i,2,k}}{\Delta F_0} \dots \dots \dots (3) \end{aligned}$$

The parameters in this equation are defined as

$$G_3 = \frac{\delta}{L}, \quad P_2 = \frac{a_s}{a_c}, \quad P_5 = \frac{\lambda_s}{\lambda_c} \text{ (solid region) or } \frac{\lambda_l}{\lambda_c} \text{ (liquid region)}$$

where the subscript c denotes copper. For the volume d in the tube, we have

$$\begin{aligned} & \frac{B_i P_4 G_4/G_1 \cdot T'_{1,j,k} - \{B_i P_4 G_4/G_1 + P_5/\Delta r'_3 \cdot (G_2/G_1 + \Delta r'_3/2)\} T'_{2,j,k} + P_5/\Delta r'_3 \cdot (G_2/G_1 + \Delta r'_3/2) T'_{3,j,k}}{(G_2 - G_4)/G_1 \cdot (G_2/G_1 + G_4/G_1)/2} \\ & + \frac{\Delta\theta'_{j+1} T'_{2,j-1,k} - (\Delta\theta'_j + \Delta\theta'_{j+1}) T'_{2,j,k} + \Delta\theta'_j T'_{2,j+1,k}}{(G_2/G_1 + G_4/G_1)^2 \pi^2 (\Delta\theta'_j + \Delta\theta'_{j+1}) \Delta\theta'_j \Delta\theta'_{j+1}/8} + \frac{T'_{2,j,k-1} - 2T'_{2,j,k} + T'_{2,j,k+1}}{(\Delta z'_k/G_1)^2} \\ & = \left\{ P_2 + \frac{\Delta r'_3 (G_2/G_1 + \Delta r'_3/4)}{(G_2/G_1)^2 - (G_4/G_1)^2} P_1 P_5 \right\} \frac{\Delta T'_{2,j,k}}{\Delta F_0} \dots \dots \dots (4) \end{aligned}$$

Biot number B_i , P_4 , and G_4 in Eq.(4) are defined as

$$B_i = \frac{h_w (r_{ai} - r_{ci})}{\lambda_s}, \quad P_4 = \frac{\lambda_s}{\lambda_c}, \quad G_4 = \frac{r_{ci}}{L}$$

where r_{ci} is the inner radius of the tube. The heat transfer coefficient h_w is evaluated from the experimental equation (3.8) in Ref.(10) in which the velocity and temperature entrance regions and the secondary flow are taken into consideration. The second term on the right hand side of Eq.(4) relates to the thermal capacity of PCM in the volume d , and for the volume e , it is replaced by the thermal capacity of the fin. Furthermore, for the heat exchanger fluid, the following equation is obtained neglecting the axial heat conduction.

$$\frac{1}{NTU} \frac{T'_{1,j,k-1} - T'_{1,j,k}}{\Delta z'_k} + T'_{2,j,k} - T'_{1,j,k} = \left(\frac{G_4}{G_1} \right)^2 \frac{P_3}{N_{uw}} \frac{\Delta T'_{1,j,k}}{\Delta F_0} \dots \dots \dots (5)$$

where

$$NTU = \frac{h_w 2\pi r_{ci} L}{\dot{m} c_w}, \quad N_{uw} = \frac{h_w 2r_{ci}}{\lambda_w}, \quad P_3 = \frac{a_s}{a_w}$$

and $\dot{m}$ is the mass flow rate of water and the subscript w denotes water. Governing equations consist of Eqs.(1) ~ (5), and they contain the non-dimensional parameters. That is, the geometric parameters $G_1 \sim G_5$ and the physical property parameters $P_1 \sim P_5$, and these are constant if the configuration and materials of an apparatus, and PCM are determined. Biot number B_i , the number of transfer units NTU and Nusselt number N_{uw} vary if $\dot{m}$ and h_w are changed. Generally, the thermal resistance in the shell side is larger than that in the tube side, and so the variation of N_{uw} does not significantly affect the heat transfer characteristics. The initial and boundary conditions are given as follows.

(Initial condition : $F_0 = 0$)

- (i) $T'_{i,j,k} = \{ (T_f + 2^\circ\text{C}) - T_f \} / (T_m - T_f)$ ($2 \leq I, J, K \leq N_I + 2, N_J + 2, N_K + 2$)
- (ii) $T'_{i,j,k} = 1$ ($2 \leq J, K \leq N_J + 2, N_K + 2$)

(Boundary condition : $F_0 > 0$)

- (i) $T'_{N_I+3,J,K} = \{ (T_f + 2^\circ\text{C}) - T_f \} / (T_m - T_f)$ ($2 \leq J, K \leq N_J + 2, N_K + 2$)
- (ii) $T'_{i,j,2} = 1$ ($2 \leq J \leq N_J + 2$)
- (iii) $T'_{i,3,K} = T'_{i,3,K}, T'_{I,N_J+3,K} = T'_{I,N_J+1,K}$ ($2 \leq I, K \leq N_I + 2, N_K + 2$)
- (iv) $T'_{i,j,1} = T'_{i,j,3}, T'_{I,J,N_K+3} = T'_{I,J,N_K+1}$ ($2 \leq I, J \leq N_I + 2, N_J + 2$)

The boundary condition (ii) prescribes the inlet temperature of water, and (iii) and (iv) represent insulating conditions. Equations (1) ~ (5) are solved by ADI method⁽¹¹⁾ under

the condition described above. Since the variations of the temperature of water in θ -direction obtained by this calculation are small, they are averaged to obtain bulk temperature. The development of the solidification layer is determined by the enthalpy method⁽¹²⁾, in which after the temperature of a control volume in PCM decreases to T_f , it is fixed at T_f ($T'_{I,J,K}=0$) until the control volume releases the latent heat H . Therefore, Stephan number $Ste=c_s|\Delta T|/H$ ($\Delta T=T_{in}-T_f$) does not apparently appear in Eqs.(1)~(5), but this is also one of the parameters governing the heat transfer in the latent heat storage unit.

3. Calculated result and discussion

3.1 Comparison with quasi-analytical solutions in the case of the bare tube

To check the validity of the present method, calculations are performed for the bare tube, and the results are compared with quasi-analytical solutions obtained by Shamsundar's method⁽¹³⁾. Figure 3 shows comparison between these results for $\Delta T=-20$ °C, $\dot{m}=0.0333$ kg/s ($Ste=0.184$) and $\Delta T=-10$ °C, $\dot{m}=0.00833$ kg/s ($Ste=0.092$) taking heat flux $q=c_w\dot{m}(T_{out}-T_{in})/A_b$ as the ordinate and time t as the abscissa. Where, A_b is the outer surface area of the bare tube and T_{out} the outlet

Fig.3 Comparison between present calculation results and quasi-analytical solutions

temperature of water, and for simplicity, it is assumed that h_w does not change with t . As shown in Fig.3, the calculated values by the present method are in good agreement with those by Shamsundar's method, and therefore the present method is considered valid.

In the case of a larger Ste ($\Delta T=-20$ °C), the value by the present method is slightly larger for small t than that by Shamsundar's method. This is probably because the latter neglects the sensible heat in PCM.

3.2 Comparison between calculated results and experimental ones

Figure 4 shows comparison between calculated results of the heat flux q and experimental ones⁽⁶⁾ in the case that $\dot{m}$ is fixed at 0.0167 kg/s and ΔT is varied for the bare tube and the finned tubes with 4 fins (4-fins tube) and with 8 fins (8-fins tube). The thickness δ of the fin is 2 mm for the 4-fins tube and 1 mm for the 8-fins tube, and so the masses of the two finned tubes are the same. From Fig.4, it is shown that q increases with ΔT or with the number N_F of fins. The decreasing rate of q with t becomes smaller for larger N_F in the region of small t . However, for large t the decreasing rate of q

Fig.4 Comparison between calculated results and experimental values of heat flux ($\dot{m}=0.0167$ kg/s)

Fig.5 Comparison of developments of solidification layer [$\Delta T=-20$ °C, $\dot{m}=0.0167$ kg/s, Figs.(a) ~ (c) correspond to the points a ~ c in Fig.4, respectively]

for the finned tube is larger than that for the bare tube. The calculated results are in good agreement with these experimental facts, and thus the validity of the present method is also proved by comparison with the experimental values.

Figure 5 shows comparison between calculated values and experimental ones about the development of solidification and temperature distributions at points $a \sim c$ in Fig. 4. Broken lines indicate the positions of the interface between solid and liquid layers, and figures represent temperature distributions obtained from the experiment. Although the interface obtained from the experiment is located outside of the calculated interface, which is the boundary between temperature regions $T \leq 34^\circ\text{C}$ and $T = T_f = 34^\circ\text{C}$, the shapes of the two interfaces are similar.

To clarify the effects of the mass flow rate $\dot{m}$ of water on the heat flux q , the calculations are carried out for $\dot{m} = 0.00833 \text{ kg/s}$ and 0.0333 kg/s in the cases of the bare tube, the 8-fins tube and a 16-fins tube ($\delta = 0.5 \text{ mm}$). Figure 6 shows the results for $\Delta T = -15^\circ\text{C}$. The effects of $\dot{m}$ are larger for small t or for large N_F , and this fact coincides with the experimental results. The reason will be explained in the following.

Now, the heat flux q is approximately evaluated from the following equation if the heat capacities are neglected.

$$q = \frac{1}{\bar{R}[1 + (1/\bar{h}_w)/\bar{R}]}(T_f - \bar{T}_w) \quad (6)$$

where $1/\bar{h}_w$ is the axial mean thermal resistance of the tube side, $\bar{R}$ that of the shell side and $\bar{T}_w$ the axial mean temperature of water. The magnitude of $\bar{R}$ depends on N_F and the development of the solidification layer. The thermal resistance $1/\bar{h}_w$ and $\bar{T}_w$ are strongly affected by the change in $\dot{m}$. Hence, from Eq. (6), it is shown that the extent of the effect of the change in $\dot{m}$ on q depends on the ratio $(1/\bar{h}_w)/\bar{R}$ of the thermal resistance of the tube side to that of the shell side, and

Fig. 6 Effects of $\dot{m}$ on timewise variations of heat flux (calculated values for $\Delta T = -15^\circ\text{C}$)

on the change in the temperature difference $T_f - \bar{T}_w$. Considering the energy balance, the following relation is derived.

$$\frac{(1/\bar{h}_w)}{\bar{R}} = \frac{\bar{T}_c - \bar{T}_w}{T_f - \bar{T}_c} \quad (7)$$

To evaluate the ratio $(1/\bar{h}_w)/\bar{R}$ at points $a \sim d$ having almost the same time in Fig. 6, the axial distributions of the temperatures of the tube and water (T_c and T_w) which are averaged values in each cross section, are shown in Fig. 7. Since the temperatures T_c and T_w at the points $a \sim d$ nearly linearly change in the axial direction, taking the arithmetic mean of the inlet and outlet temperatures as the axial mean value, the values of $(1/\bar{h}_w)/\bar{R}$ are obtained. The values for the 16-fins tube are 1.24 at the point a and 1.78 at the point b , and these are larger than 1.0. On the other hand, the values for the bare tube are 0.367 at the point c and 0.508 at the point d , and these are much smaller than 1.0. Therefore, in the case of the bare tube, the thermal resistance of the shell side predominates. But, in the case of the 16-fins tube, since the thermal resistance of the shell side is smaller than that of the tube side, the effects of the change in $\dot{m}$ or $\bar{h}_w$ are significant. Furthermore, for the bare tube, the value of $T_f - \bar{T}_w$ in Eq. (6) decreases by 1.6°C from the point c to the point d , and for the 16-fins tube, it decreases by 2.4°C from the point a to the point b . Hence, the decrease of $T_f - \bar{T}_w$ for the 16-fins tube as a result of the decrease of $\dot{m}$ is larger than that for the bare tube. This is because the heat flux of the 16-fins tube is large and the axial increase of T_w due to the decrease of $\dot{m}$ is large, while T_f is fixed. It is concluded that the reasons why the effects of the change in $\dot{m}$ on q for the finned tube are larger than those for the bare tube are twofold as mentioned above.

Figure 8 shows comparison between calculated results and experimental ones about the variation of the solidification fraction $\bar{Q} = \int q \cdot A_b dt / [M(1 + c_s |\Delta T|)]$ with non-dimensional time $\tau = F_0 \cdot St_e$. Where, M is the mass of PCM. Although the calcula-

Fig. 7 Axial variations of temperatures of water and tube [subscripts $a \sim d$ correspond to the points $a \sim d$ in Fig. 6, respectively]

tions were performed for various ΔT in the case of $\dot{m}=0.0167$ kg/s, there are little difference between these results, and so the result for $\Delta T=-15$ °C is representatively shown in the figure. The values of NTU and B_i vary with time, and they are representatively indicated for $t=0$. In the figure, the calculated results agree well with the experimental facts that the effects of S_{te} on $\bar{Q}$ are very small for both tubes, the effects of the change in $\dot{m}$ (or NTU and B_i) are significant, and the increasing rate of $\bar{Q}$ with τ for the finned tube is considerably larger than that for the bare tube.

Figure 9 shows the relation between the heat transfer effectiveness $\eta_T = (T_{out} - T_{in}) / (T_f - T_{in})$ and the non-dimensional time τ , and parameters in the figure have the same meanings as those in Fig.8. The calculated values agree well with the experimental ones, and it is seen that η_T of the 8-fins tube is quite larger than that of the bare tube, and the decreasing rate of η_T of the 8-fins tube is greater for large t .

3.3 Effects of the number and thickness of fins on the heat flux

The validity of the present calculation method is verified in the previous sections. Next, using this method, calculations are performed to make clear the

Fig.8 Timewise variations of the solidification fraction

Fig.9 Timewise variations of the heat transfer effectiveness

effects of the number N_F and thickness δ of fins on the heat transfer. In calculations, when N_F is varied, δ is also changed in such manner that the mass of the fins is the same as that of the fins with $N_F=4$ and $\delta=2$ mm. Figure 10 shows the relation between q and N_F taking t as a parameter in the case of $\Delta T=-15$ °C and $\dot{m}=0.0167$ kg/s. For small or quite large t , the heat flux q first increases with N_F and then becomes nearly constant. Therefore, it is not effective to have a too large number of fins. However, in the region of $1000 < t \leq 4000$ s, q increases with N_F even if N_F is quite large. These facts are also recognized from comparison of the experimental values between the 4 and 8-fins tubes in Fig.4.

The effects of δ for the 8-fins tube on q are shown in Fig.11 for $\Delta T=-15$ °C and $\dot{m}=0.0167$ kg/s. For comparison, the variation of q for the bare tube and the values $\bar{Q}$ at $t=4500$ s are also indicated. From the figure, the effects of fin is considerably large even if its thickness is very small. For $\delta=0.2$ mm, the ratio of $\bar{Q}$ of the 8-fins tube to that of the bare tube is about 1.6 for $t=4500$ s. Although the heat flux increases with δ , its increasing rate with δ is small if δ is very large.

The numerical calculations were

Fig.10 Effects of the number of fins on heat flux (calculated results for $\Delta T=-15$ °C, and $\dot{m}=0.0167$ kg/s)

Fig.11 Effects of the thickness of fin on heat flux (calculated values for $\Delta T=-15$ °C, and $\dot{m}=0.0167$ kg/s)

carried out using FACOM-M200 at the Computer Center, Kyushu University, and the computation time was 15-20 minutes for one job.

4. Conclusions

A three-dimensional calculation method was proposed in order to clarify the heat transfer characteristics of a latent heat storage unit with a finned tube in the solidification process. The results were compared with the previous experimental data⁽⁶⁾ to check the validity of the calculation method. The results obtained are summarized as follows:

(1) The variation of heat flux q with time t and the development of solidification layer on the finned tube calculated by the present method, in which ADI method as a calculation method and the enthalpy method for the determination of interface were used, are in good agreement with those obtained from the experiments.

(2) The effects of the change in the mass flow rate $\dot{m}$ of a heat exchanger fluid on the heat flux q increase with the number N_F of fins. The reason is explained from temperature distributions of the heat exchanger fluid and the tube obtained from the present calculations.

(3) The calculated results agree well with the experimental facts that the variations of the solidification fraction and the heat transfer effectiveness with a non-dimensional time are slightly affected by the change in Stephan number, but strongly affected by the number of transfer units and Biot number.

(4) In the region of small or very large t , the heat flux q first increases with N_F and then becomes nearly constant. But, for intermediate time, q increases even if N_F is very large. The effectiveness of fin is very large even if the thickness δ of fin is small, but the increase of the effectiveness with δ is limited.

The present calculation method will

be useful for the design of heat exchangers for latent heat storage if calculations are performed for a wide range of the non-dimensional parameters appearing in the basic equations. These results will be published in the next report.

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