PHASE CHANGE MATERIAL SOLIDIFICATION IN A FINNED CYLINDRICAL SHELL THERMAL ENERGY STORAGE
An approximate analytical approach

by
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Results are reported of an investigation of the solidification of a phase change material in a cylindrical shell thermal energy storage with radial internal fins. An approximate analytical solution is presented for two cases. In case 1, the inner wall is kept at a constant temperature and, in case 2, a constant heat flux is imposed on the inner wall. In both cases, the outer wall is insulated. The results are compared to those for a numerical approach based on an enthalpy method. The results show that the analytical model satisfactory estimates the solid-liquid interface. In addition, a comparative study is reported of the solidified fraction of encapsulated phase change material for different geometric configurations of finned storage having the same volume and surface area of heat transfer.

Key words: thermal energy storage, phase change material, analytical, finned cylindrical shell storage, enthalpy method

Introduction

Energy use varies significantly during the day and night according to the demands of industrial, commercial, residential and other activities, especially in extremely hot and cold climates. As for in thermal energy grow increasingly, energy use is correspondingly increasing. Thermal energy storage (TES) systems are an important technology for reducing the dependency of society on fossil fuels. TES can be accomplished using either sensible heat or latent heat storages. Latent heat thermal storage (LHTS) is particularly attractive since it provides a high energy storage density and has the capacity to store energy at a constant temperature corresponding to that of the phase transition. The storage of energy by latent heat during the melting process of phase change materials (PCM) occurs in many applications: thermal comfort, electronic cooling, solar water heating, etc. A comprehensive reference related to PCM storage has been reported by Dincer and Rosen [1], which contains a complete review of the types of materials which have been used, their advantages and disadvantages and various experimental techniques used to determine the behavior of PCM in melting and solidification processes.

The major disadvantage of PCM is that most have an acceptably low thermal conductivity, leading to low melting and solidification rates. Heat transfer enhancement techniques are therefore required for most LHTS applications. Velraj et al. [2] studied experimentally and
numerically various heat transfer enhancements for a solar thermal storage system. Among the enhancement procedures reported, the finned system has been reported as one of the most practical. A good review has been presented by Agyenim et al. [3] detailing various PCM investigated over the last three decades, the heat transfer and enhancement techniques employed in PCM to effectively charge and discharge latent heat and the formulation of the phase change problem.

Heat transfer in a PCM storage is a transient, non-linear phenomenon with a moving solid-liquid interface. Analytical solutions for these problems are known only for a few physical situations with simple boundary conditions and/or simple geometries. Some analytical approximation methods for solving melting/solidification problems have been presented by Alexiades and Solomon [4]. Dutil et al. [5] presented a good review on PCM, particularly for mathematical modeling and simulation including analytical, numerical, and experimental simulations.

Velraj et al. [6] developed theoretical and experimental work for LHTS consisting of a cylindrical vertical tube with internal longitudinal fins; this tube assembly is, in turn, placed inside another cylindrical vessel containing water. It was concluded that this configuration, which forms a V-shaped enclosure for the PCM, gives maximum benefit to the fin arrangement. Lamberg et al. [7] investigated numerically and experimentally melting/solidification in a rectangular PCM storage with internal fins. The numerical methods utilized were an enthalpy method and an effective heat capacity method. Both numerical methods provide good estimations for the temperature distribution in both the melting and solidification process. Talati et al. [8] studied analytically and numerically solidification of a finned rectangular PCM storage with an imposed constant heat flux as a boundary condition. The results showed that when the length of fin to the height of storage ratio is smaller than unity, the PCM solidified more quickly in the storage. Lamberg and Siren [9] developed a simplified analytical model for solidification to predict the solid-liquid interface location in a rectangular PCM storage with internal fins where the PCM initial temperature was equal to the solidification temperature. The analytical and numerical results showed good agreement. Lamberg [10] continued the 1-D analytical approach of the 2-D heat transfer problem in a finned PCM storage for the case where the storage was initially at a higher temperature than the PCM solidification temperature. Erek et al. [11] investigated numerically and experimentally an LHTS incorporating phase change around a radial finned tube. The results showed that the stored energy increases with increasing fin radius and decreasing fin space. Zhang and Faghri [12] studied the heat transfer enhancement in the LHTS using a finned tube. They solved the heat conduction problem in the tube wall, fins and phase change process of the PCM outside the tube by the temperature transforming model proposed by Cao and Faghri [13]. The results showed that the height of the fin significantly influences the solid-liquid interface at the two sides of the transverse fins. But the solid-liquid interface at the axial position between fins is affected very little by the height of the fin.

The main purpose of this paper is to present an approximate analytical model for solidification process in 2-D cylindrical shell PCM storage with radial internal fins. The PCM is solidified isothermally via two different cases: in case 1, the inner wall is kept at a constant temperature and, in case 2, a constant heat flux is imposed on the inner wall. In both cases, the outer wall is insulated. The analytical model is required to predict the solid-liquid interface location during the solidification process to determine the solidification (discharge) duration of the storage. The approximate analytical model is compared to a 2-D numerical model using the enthalpy method. In addition, a comparative study for the total solidification time is performed for finned cylindrical shell and finned rectangular storages in order to determine the best performing configuration.
Mathematical formulation

Figure 1 illustrates the storage investigated in the present work. The storage is filled with a PCM in liquid phase at the melting/solidification temperature, $T_m$. In case 1, the temperature of the inner wall of the cylindrical shell storage is lowered at time $t = 0$ and fixed at the temperature $T_w$. In case 2, at time $t = 0$, a constant heat flux is imposed on the inner wall of cylinder shell storage. The outer side of the storage is insulated.

Sensible heat is released from the storage first as the liquid PCM cools. Then the latent heat of fusion is released during phase change and finally sensible heat is transferred from the solid PCM as it cools further until it reaches to the inner wall temperature. The main heat transfer mode in the solidification process is conduction. Although natural convection occurs in the solid-liquid interface due to temperature differences in the liquid PCM, its effect on the solid-liquid interface location is negligible compared to the effect of heat conduction [14]. Heat transfer in PCM storage is a transient, non-linear phenomenon with a moving solid-liquid interface that is generally referred to as a moving boundary problem (MBP). The non-linearity is the principal challenge in MBP and analytical solutions for these problems are known only for a few physical situations with simple boundary conditions and/or simple geometries. In the present investigation, the following assumptions are introduced:

- the thermophysical properties of the PCM are constant. But they are different for the solid and liquid phases,
- the phase change process in the PCM is isothermal,
- the thermophysical properties of the fin are constant; also, the temperature distribution in the fin is considered to be 1-D because the fin is thin and its conductivity is high,
- the PCM is homogeneous, and
- heat conduction in the PCM can be considered 2-D, i.e., varying in axial and radial directions.

A simplified model is introduced to determine analytically the location of the solid-liquid interface during solidification by dividing the storage into two regions (fig. 2). In region 1, the fin does not influence the solidification process. Therefore, heat is transferred from the wall in the $r$-direction only. In region 2, the fin has a notable effect on heat release from the storage.

Analytical approach for region 1

Considering the heat transfer process to involve only conduction, the conduction equation for the solid and solid-liquid interface can be written as

$$
\frac{1}{r} \frac{\partial}{\partial r} \left[ r \frac{\partial T_s}{\partial r} \right] = \frac{1}{\alpha_s} \frac{\partial T_s}{\partial t}, \quad a < r < R(t), \quad t > 0
$$

(1)
\[-\rho_s L \frac{dR(t)}{dt} = -k_s \frac{\partial T_s[T(t), t]}{\partial r}\]  

(2)

The initial and boundary conditions are:

\[T_s(r, 0) = T_m, \quad a < r < b \]  

(3)

\[R(0) = 0 \]  

(4)

\[-k_s \frac{\partial T_s(a, t)}{\partial r} = q''_w, \quad q'' < 0 \quad \text{for case 2} \]  

(5)

\[-\frac{\partial T_s(b, t)}{\partial r} = 0 \]  

(6)

where \(R\) is the location of the solid-liquid interface in the \(r\)-direction. Equations (1)-(6) can be solved by the separation of variables method, and the temperature distribution in region 1 is found to be:

\[T_s(r, t) = T_m + \sum_{m=1}^{\infty} \frac{\exp(-\alpha_m^2 t)}{N(\beta_m)} R^2 (J_0(\beta_m R) Y_0(\beta_m R) - J_1(\beta_m R) Y_0(\beta_m R)) \]  

(7)

Here \(J_0\) and \(Y_0\) are Bessel functions of zero order of the first and second kind, respectively. The values of the \(\beta_m\) parameters are the roots of the following transcendental equations:

\[J_0(\beta_m a) Y_0(\beta_m R) - J_0(\beta_m R) Y_0(\beta_m a) = 0 \quad \text{for case 1} \]  

(8)

\[J_1(\beta_m a) Y_0(\beta_m R) - J_0(\beta_m R) Y_1(\beta_m a) = 0 \quad \text{for case 2} \]

The constant \(A\) is equal to:

\[A = \begin{cases} \frac{T_m - T_w}{\ln \left( \frac{R}{a} \right)} & \text{for case 1} \\ \frac{q''_w a}{k_s} & \text{for case 2} \end{cases} \]  

(9)

The norm \(N(\beta_m), \bar{I}_1, \text{and } \bar{I}_2\) are equal to:

\[N(\beta_m) = \frac{2}{\pi^2} \frac{J_0^2(\beta_m a) - J_0^2(\beta_m R)}{\beta_m^2 J_0^2(\beta_m a)} \quad \text{for case 1} \]  

(10)

\[N(\beta_m) = \frac{2}{\pi^2} \frac{J_1^2(\beta_m a) - J_0^2(\beta_m R)}{\beta_m^2 J_0^2(\beta_m a)} \quad \text{for case 2} \]

\[\bar{I}_1 = -\frac{a}{\beta_m R^2} \ln \left( \frac{R}{a} \right) \frac{J_1(\beta_m a)}{\beta_m^2} + \frac{1}{\beta_m^4 R^2} [J_0(\beta_m R) - J_0(\beta_m a)] \]  

(11)

\[\bar{I}_2 = -\frac{a}{\beta_m R^2} \ln \left( \frac{R}{a} \right) \frac{Y_1(\beta_m a)}{\beta_m^2} + \frac{1}{\beta_m^4 R^2} [Y_0(\beta_m R) - Y_0(\beta_m a)] \]  

(12)

The transcendental eq. (8) converges very slowly for small values of time or when \(R \approx a\). Thus, a Laplace transform is applied to the time variable. By introducing \(\theta(r, t) = T(r, t) - T_m\), the Laplace transforms of heat conduction eqs. (1), (5) and (6) become:
The solutions of eqs. (13)-(15) are:

\[
\frac{\bar{\theta}(r,s)}{T_w - T_m} = \frac{1}{s} \left( I_0 \left( \frac{R}{\sqrt{a_s}} \right) K_0 \left( \frac{a}{\sqrt{a_s}} \right) - I_0 \left( \frac{a}{\sqrt{a_s}} \right) K_0 \left( \frac{R}{\sqrt{a_s}} \right) \right)
\]

for case 1 (16)

\[
\frac{\bar{\theta}(r,s)}{q_w^* \sqrt{\alpha_s}} = \frac{1}{s} I_0 \left( \frac{R}{\sqrt{a_s}} \right) K_0 \left( \frac{a}{\sqrt{a_s}} \right) + I_0 \left( \frac{a}{\sqrt{a_s}} \right) K_0 \left( \frac{R}{\sqrt{a_s}} \right)
\]

for case 2 (17)

where \( I_0 \) and \( K_0 \) are modified Bessel functions of zero order of the first and second kind, respectively.

To obtain solutions applicable for times shortly after solidification starts, when the PCM thickness is small, eqs. (16) and (17) are expanded as an asymptotic series [15]. Finally, the inversion of transforms are obtained as:

\[
\frac{T(r,t) - T_m}{T_w - T_m} = \sqrt{\frac{a_s}{r \gamma_n}} \text{erfc}(-\gamma_n) - \text{erfc}(\omega_n)
\]

for case 1 (18)

\[
\frac{T(r,t) - T_m}{q_w^* \sqrt{\alpha_s}} = \frac{3 \alpha_s}{a_s \sum_{n=0}^{\infty} \left[ \text{erf}(-\gamma_n^2) - \text{erf}(-\omega_n^2) \right] - \left[ \gamma_n \text{erfc}(\gamma_n) - \omega_n \text{erfc}(\omega_n) \right]} \]

for case 2 (19)

where

\[
\gamma_n = \frac{2nR - a(1 + 2n) + r}{\sqrt{4\alpha_s t}}
\]

\[
\omega_n = \frac{2R(1 + n) - a(1 + 2n) - r}{\sqrt{4\alpha_s t}}
\]

Analytical approach for region 2

In region 2, the movement of the solid-liquid interface is assumed to occur only in the x-direction. The energy balance for the fin can be written with the initial and boundary conditions as:

\[
\frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial T_f}{\partial r} \right) + \frac{k_s}{k_f} \frac{T_m - T_f}{X \delta} = \frac{1}{\alpha_f} \frac{\partial T_f}{\partial t}, \quad a < r < b, \quad t > 0
\]
\[ T(r,0) = T_w \quad a < r < b \]  
\[ T_t(a,t) = T_w \quad \text{for case 1} \]  
\[ -k_t \frac{\partial T_t(a,t)}{\partial r} = q_w^*, q^* < 0 \quad \text{for case 2} \]  
\[ \frac{\partial T_w(b,t)}{\partial r} = 0 \]  
\[ \frac{\partial T_t(a,t)}{\partial r} = 0 \]

where \( \delta \) is the half thickness of the fin and \( X \) – the location of the solid-liquid interface in the x-direction. The following dimensionless variables are introduced:

\[ \bar{r} = \frac{r}{b-a}, \quad \bar{b} = \frac{b}{b-a}, \quad \bar{a} = \frac{a}{b-a} \]

\[ \lambda^2 = \frac{k_s (b-a)^2}{k_f} \]

The solution for the temperature of the fin is:

\[ T_t - T_m = \sum_{m=1}^{\infty} \frac{1}{N(\zeta_m)} \exp \left[ -\frac{(\zeta_m^2 - \lambda^2)^\alpha t}{(b-a)^2} \right] \psi_0 (\zeta_m \bar{r}) \psi_0 (\zeta_m \bar{r}) d\bar{r} \]

where

\[ \psi_0 (\zeta_m, \bar{r}) = Y_0 (\zeta_m \bar{r}) J_1 (\zeta_m \bar{b}) - J_0 (\zeta_m \bar{r}) Y_1 (\zeta_m \bar{b}) \]

\[ \phi(\bar{r}) = (T_w - T_m) I_0 (\lambda \bar{r}) K_1 (\lambda \bar{b}) + I_1 (\lambda \bar{b}) K_0 (\lambda \bar{r}) \]

\[ \frac{\partial \phi(\bar{r})}{\partial \bar{r}} = q_w^* (b-a) I_0 (\lambda \bar{r}) K_1 (\lambda \bar{b}) + I_1 (\lambda \bar{b}) K_0 (\lambda \bar{r}) \]

Here \( I_1 \) and \( K_1 \) are modified Bessel functions of order 1 of the first and second kind, respectively.

The values of the \( \zeta_m \) parameters are the roots of the following transcendental equation:

\[ J_1 (\zeta_m \bar{b}) Y_0 (\zeta_m \bar{a}) - J_0 (\zeta_m \bar{r}) Y_1 (\zeta_m \bar{b}) = 0 \quad \text{for case 1} \]

\[ J_1 (\zeta_m \bar{b}) Y_0 (\zeta_m \bar{a}) - J_1 (\zeta_m \bar{r}) Y_1 (\zeta_m \bar{b}) = 0 \quad \text{for case 2} \]

The norm \( N(\zeta_m) \) can be obtained as:

\[ N(\beta_m) = \frac{2}{\pi^2} \frac{J_0^2 (\zeta_m \bar{a}) - J_1^2 (\zeta_m \bar{b})}{\zeta_m^2 J_0^2 (\zeta_m \bar{a})} \quad \text{for case 1} \]

\[ N(\beta_m) = \frac{2}{\pi^2} \frac{J_1^2 (\zeta_m \bar{a}) - J_0^2 (\zeta_m \bar{b})}{\zeta_m^2 J_1^2 (\zeta_m \bar{a})} \quad \text{for case 2} \]

For case 2, \( \zeta_0 \) is also an eigenvalue. The corresponding eigenfunction is \( \psi_0 = 1 \) and the norm is \( N(\zeta_0) = (b^2 - a^2)/2 \). Details on solving the integral in eq. (28) were presented by Kerenev [16]. Then, the distance of the solid-liquid interface from the fin in region two can be obtained from

\[ X(t) = \sqrt{\frac{2k_s (T_m - T_t)}{\rho_s L}} \]
2-D numerical approach

Considering the foregoing assumptions, the conservation of energy for the conduction dominated phase change in the PCM can be expressed in term of sensible enthalpy, temperature and solid fraction as [17]:

$$\frac{\partial h}{\partial t} = \nabla \left( \frac{k}{\rho} \nabla T \right) + L \frac{\partial f_s}{\partial t}$$  \hspace{1cm} (34)

where

$$h = \int_{T_a}^T c_d T' \, dT'$$  \hspace{1cm} (35)

For isothermal phase change, the local solid fraction $f_s$ is defined as:

$$f_s(T) = \begin{cases} 
  0 & \text{if } T > T_m \text{ (liquid)} \\
  1 & \text{if } T < T_m \text{ (solid)} 
\end{cases}$$  \hspace{1cm} (36)

For the region of the PCM in fully solid or fully liquid state, incorporating the definition of sensible enthalpy in eq. (35) and the solid fraction in eq. (36) allows eq. (34) to reduce to an ordinary heat conduction equation. The solid fraction $f_s$ in the vicinity of melting strictly lies in the interval $(0, 1)$. So, the fully implicit finite difference equation, eq. (34), for an internal node $(i, j)$ can be written as:

$$f_{i,j} = f_{i,j}^{old} - \frac{k_s \Delta t}{\rho_s L (\Delta r)^2} \left[ \left( 1 + \frac{1}{2i} \right) T_{i-1,j} - 2T_{i,j} + \left( 1 - \frac{1}{2i} \right) T_{i+1,j} \right] -$$

$$- \frac{k_s \Delta t}{\rho_s L (\Delta z)^2} (T_{i,j-1} - 2T_{i,j} + T_{i,j+1})$$  \hspace{1cm} (37)

For completely solid or liquid phases, the last term in eq. (34) vanishes. Applying discretization and using the backward difference approach, the equation for an internal node $(i, j)$ is given as:

$$T_{i,j} = T_{i,j}^{old} + \frac{\alpha_s \Delta t}{(\Delta r)^2} \left[ \left( 1 + \frac{1}{2i} \right) T_{i-1,j} - 2T_{i,j} + \left( 1 - \frac{1}{2i} \right) T_{i+1,j} \right] +$$

$$+ \frac{\alpha_s \Delta t}{(\Delta z)^2} (T_{i,j-1} - 2T_{i,j} + T_{i,j+1})$$  \hspace{1cm} (38)

Results and discussion

An approximate analytical solution of the 2-D heat transfer problem in a finned, cylindrical shell PCM storage has been developed. Note that the obtained approximate solution, although analytical, is not closed form, and it thus requires computational assistance when applied in design. Note also that the solution could be beneficially expressed in a series of design-aid figures, when similar situations are repeatedly encountered.

Although the prime objective of the present study is to obtain an approximate analytical solution of the 2-D heat transfer problem in a finned, cylindrical shell PCM storage, we also investigate the total solidification time of the PCM encapsulated in containers having two geometric configurations: finned cylindrical shell and finned rectangular.

A program was developed in Visual FORTRAN 6.5 to solve the system of equations and evaluate relevant phase change parameters such as the solid-liquid interface location and the time for complete PCM solidification. As a limit for convergence the value of $10^{-3}$ mm is chosen. The step size is 0.1 mm and the time step is 1 s.
The calculated results presented are for the salt hydrate ClimSel C23 as the PCM and aluminum as the fin. Thermophysical properties are listed in tab. 1.

**Table 1. Thermophysical properties of the PCM [9] and fin material [18]**

<table>
<thead>
<tr>
<th>Property</th>
<th>Aluminum fin</th>
<th>Salt hydrate ClimSel C23</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density, ( \rho ) [kg m(^{-3})]</td>
<td>2,770</td>
<td>1,480</td>
</tr>
<tr>
<td>Specific heat capacity, ( c ) [J kg(^{-1}) K(^{-1})]</td>
<td>875</td>
<td>2,660</td>
</tr>
<tr>
<td>Thermal conductivity, ( k ) [W m(^{-1}) K(^{-1})]</td>
<td>177</td>
<td>0.6</td>
</tr>
<tr>
<td>Latent heat of fusion, ( L ) [J kg(^{-1})]</td>
<td>–</td>
<td>148,000</td>
</tr>
<tr>
<td>Solidification temperature, ( T_m ) [°C]</td>
<td>–</td>
<td>23</td>
</tr>
</tbody>
</table>

Moreover, in case 1, the inner wall is kept at a constant temperature \( T_w = 10 \) °C. In case 2, the PCM is solidified by an imposed constant heat flux \( q_w^* = 1.000 \) W/m\(^2\).

1-D analytical and 2-D numerical predictions of the solid-liquid interface location for case 1 are shown in fig. 3 for three storage types and two times. The results for the two methods are compared to find the accuracy and performance of the 1-D analytical method. When the fin is long, it plays the most important role in transferring heat from the storage and the 1-D analytical model exhibits more error compared to other storage types in region 2. In this case, the large-
est difference of the solid-liquid interface calculated using the derived analytical model and the numerical solution occurs when $t = 200$ s and is about 0.4 mm and the error is about 10%.

It can be seen in figs. 3 and 4, the solidification rate decreases with increasing time. Initially the thickness of the solidified layer is small and hence the thermal resistance is small, leading to a high solidification rate. As time passes, the thickness of solidified PCM increases, causing an increase in the thermal resistance and a reduced solidification rate.

The fraction of PCM encapsulated in finned cylindrical shell and finned rectangular storages in case 1 are compared in fig. 5 for varying storage types. The heights of the different geometric storages are assumed equal, while other relevant storage dimensions are determined on the basis of equal volume of the PCM encapsulated and equal surface area of heat transfer. The variation of fraction of solidified PCM with time for the finned cylindrical shell storage is computed using the present analytical model and is compared with that presented by Lamberg and Siren [9] for finned rectangular storage. It is evident from fig. 5 that the total solidification time of the PCM encapsulated in the finned cylindrical shell storage is less than that in the finned rectangular storage. When the inner wall is kept at a constant temperature, the thermal resistance of the solidified PCM increases at a greater rate for the rectangular storage than for the cylindrical shell storage. Therefore, the heat release is slower in a rectangular storage and the total solidification time is longer.

Figure 6 shows a comparison of the fraction of solidified PCM encapsulated in finned cylindrical shell and finned rectangular storages for case 2 and several storage types, on the basis of equivalent volume and heat transfer surface area. The fraction of solidified PCM for the finned cylindrical shell storage is evaluated using the present analytical model and is compared with that presented by Talati et al. [8] for finned rectangular storage. The results show that when a constant heat flux is imposed to the inner wall, the total solidification time for the cylindrical shell storage is considerably more than that for the rectangular storage. Because of constant heat
flux, when the distance between fins is great, a large amount of heat is released from the PCM storage in region 1. Therefore, as shown in fig. 6, the total solidification time of the PCM is considerably less for storage type 3 in both cylindrical shell and rectangular storages.

Conclusions

An approximate analytical method is presented to predict the solid-liquid interface location in the solidification process of a PCM in a finned cylindrical shell storage. The outer wall is taken to be insulated, and two boundary conditions are considered: constant inner wall temperature (case 1) and constant inner wall heat flux (case 2). In addition to the analytical approach, the problem is solved numerically in 2-D based on an enthalpy method. The results for the analytical model and numerical method are in good agreement. The approximate analytical model is expected to be useful in the pre-design stage of LHTS. Additionally, the results are reported of a comparison of the fraction of solidified PCM encapsulated in finned cylindrical shell and finned rectangular storages having the same volume and heat transfer surface area. For case
1, the PCM total solidification time corresponding to all three storage types in a cylindrical shell storage is less than that in a rectangular storage. In case 2, however, the PCM total solidification time is considerably greater for the cylindrical shell storage than the rectangular storage, for all storage types.

Acknowledgments

The support of the Iranian Fuel Conservation Organization (IFCO) is gratefully acknowledged by the authors.

Nomenclature

- $a$ – inner radius, [m]
- $b$ – outer radius, [m]
- $c$ – specific heat capacity, [J kg$^{-1}$ K$^{-1}$]
- $f$ – local solid fraction
- $h$ – specific enthalpy, [J kg$^{-1}$]
- $k$ – thermal conductivity, [W m$^{-1}$ K$^{-1}$]
- $L$ – latent heat of fusion, [J kg$^{-1}$]
- $q^*$ – heat flux, [W m$^{-2}$]
- $R$ – distance of solid-liquid interface in r-direction, [m]
- $T$ – temperature, [°C]
- $X$ – distance of solid-liquid interface in x-direction, [m]
- $z$ – half height of PCM cell, [m]

Greek symbols

- $\alpha$ – thermal diffusivity, [m$^2$ s$^{-1}$]
- $\delta$ – half thickness of fin, [m]
- $\rho$ – density, [kg m$^{-3}$]

Subscripts

- $f$ – fin
- $m$ – melting
- $s$ – solid
- $w$ – wall

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