INFLUENCE OF VOID RATIO ON PHASE CHANGE OF THERMAL ENERGY STORAGE FOR HEAT PIPE RECEIVER

by

Xiaohong GUI [a*], Xiange SONG [b], Tie LI [a], and Dawei TANG [a]

[a] Institute of Engineering Thermophysics, Chinese Academy of Sciences, Beijing, China
[b] Beijing International Studies University, Beijing, China

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The influence of void ratio on phase change of thermal storage unit for heat pipe receiver under microgravity is numerically simulated. Accordingly, mathematical model is set up. A solidification-melting model upon the enthalpy-porosity method is specially provided to deal with phase changes. The liquid fraction distribution of thermal storage unit of heat pipe receiver is shown. The fluctuation of melting ratio in phase change material canister is indicated. Numerical results are compared with experimental ones in Japan. The results show that void cavity prevents the process of phase change greatly. Phase change material melts slowly during sunlight periods and freezes slowly during eclipse periods as void ratio increases. The utility ratio of phase change material during both sunlight periods and eclipse periods decreases obviously with the improvement of void ratio. The thermal resistance of void cavity is much higher than that of phase change material canister wall. Void cavity prevents the heat transfer between phase change material zone and canister wall.

Key words: *void ratio, phase change, thermal energy storage, heat pipe receiver*

Introduction

A heat pipe receiver with high temperature phase change material (PCM) is a very important component for advanced solar dynamic power generation system during traversal of low earth orbit spacecraft through the eclipse phase of their orbit cycles. The main functions of the receiver are the absorption of solar energy, thermal energy storage (TES) for the eclipse period, and the heating of working gas. The working gas is helium-xenon gas mixture. The molar mass for helium and xenon is 4.00 and 131.29 respectively. The working gas with different molar mass can be obtained if helium and xenon are mixed with different proportions. Usually, the molar mass of helium-xenon gas mixture is 39.9 or 83.8 [1, 2].

A heat pipe receiver contains an array of heat pipes with mounted cylindrically toroidal thermal energy storage modules. The heat pipe receiver is composed of a cavity receiver, thermal energy storage and a sodium heat pipe, and aimed at transporting solar heat to a bottoming system with minimized heat loss [3].

High temperature phase change material is commonly used as thermal energy storage for heat pipe receivers in advanced solar dynamic power generation systems [4-7]. In NASA (Na-

* Corresponding author; e-mail: guixiaohong@iet.cn

tional Aeronautics and Space Administration) project, the phase-change salt (LiF) is used. It is assumed that void cavity distributes along the outer wall of thermal storage unit. The worst heat transfer condition in canister is considered by this kind of assumption [4]. Namkoong *et al.* launched an experiment which was the first to melt and freeze a high temperature thermal energy storage material under an extended duration of microgravity [8]. TES containing LiF in an annular volume, performed flawlessly in the 22 hours of its operation. The void behavior of substances undergoing phase change under microgravity was observed. The frozen salt was observed non-destructively by a technique of "tomography". Similar in concept to a medical "cat-scan", this method allowed observing the solid LiF, and therefore the void pattern, without having to physically section the canister. The movement and behavior of the void during the entire freezing and melting cycles were clearly shown. The position of the void can result in either of two extremes when heat impinges upon the containment surface, such as when the space craft emerges from orbital shade to sun. A void adjacent to the heated surface can result in a hot spot. The void remote from the surface, on the other hand, can result in the melting PCM trapped in the still frozen solid. The shape of the interface, showing a somewhat larger void volume at the sector closest to the strip heater, indicated a vestige of the void during the melting period of the cycle.

A PCM canister is a basic thermal storage unit of a heat pipe receiver. Void cavity appears in canister when PCM freezes and shrinks. Void cavity locates initially in those zones that temperatures are higher and prefers the direction that heat flux is highest in PCM canister. Since the cycle fluid in heat pipe can take the heat away by its inner wall, temperatures in inner wall are lowest during eclipse periods. PCM in the thermal storage device (TSD) region freezes firstly in inner wall and extends outward gradually along radial direction. Void cavity forms a columnar annular space on outer wall finally. High temperature phase change material is commonly used as 80.5%LiF-19.5%CaF_2. Its void ratio is 8%-22%. Since the thermal resistance of void cavity is very high, void cavity prevents phase change. Influence of void ratio on phase change of thermal energy storage for heat pipe receiver under microgravity needs to be analyzed in detail. In this paper, the numerical simulation based on the enthalpy-porosity method is built. The research results can be used to guide designing the PCM canister for heat pipe receiver.

Structural design and working principle of a heat pipe receiver

Heat pipes, using a liquid metal such as sodium, can be uniformly utilized to transfer the solar flux to the PCM and circulating fluid in the receiver [3]. The heat pipe receiver comprises a cylindrical receiver cavity, the walls of which are lined with 24 heat pipes along the sides of the cavity. The heat pipes comprise three functional zones, the receiver portion, the thermal storage device (TSD) portion, and the heat source heat exchanger (HSHX) portion. The receiver section is located nearest to the receiver aperture and includes no thermal storage unit. The TSD section of each heat pipe is rounded by 24 thermal storage units. The annular canisters of PCM are axisymetric to the heat pipe axis. The thermal storage units are stacked and brazed to the TSD section of heat pipes. The outer radius of each heat pipe is equal to the inner radius of each thermal storage unit. The thermal storage units are not bonded to each other but are separated by ceramic fiber spacers. The HSHX comprises individually sheathed heat pipes for heat exchange with the engine working fluid.

A heat pipe receiver is operated in the following manner. During sunlight periods, the insolation in the receiver evaporates the heat pipe fluid. Condensation of the heat pipe fluid occurs in the TSD portion, melting the PCM, as well as in the HSHX portion.

During eclipse periods, the receiver portion of the heat pipe is essentially in an adiabatic region though some condensation may occur due to heat losses. The TSD region acts as a heat pipe evaporator, with condensation occurring in the HSHX. In any operating condition, the heat pipe fluid attains a certain temperature and thermal conductance. Since the heat pipe is nearly an isothermal device, almost the same heat pipe fluid temperature exists throughout the entire length of the heat pipe. The internal pressure is the fluid vapor pressure at an existing temperature. The heat pipe temperature and pressure vary with time throughout an orbit.

A portion of the energy from the heat pipe fluid is conducted into the PCM and stored as latent energy while the remaining energy is transported by the heat pipe to the last section of the receiver where a heat exchanger couples the heat pipe to the engine. During a solar eclipse, pressure and temperature in the heat pipe drop, and energy starts to escape from the PCM via the heat pipe fluid, which transports it to the heat exchanger in the last section of the receiver.

Numerical simulation of the PCM canister of the heat pipe receiver

Physical model

The heat transfer of thermal storage unit of the heat pipe receiver [9] includes:

– heat conduction among solid, liquid in PCM zone and saturated vapor in void cavity, heat conduction among thermal storage unit wall, fin and other enhanced heat additions, In order to simulate the periodic flow in the cross wavy primary surface, some assumptions are made,
– radiation among interfaces of the void cavity with different temperature and radiation among different PCM crystals, and
– evaporation and condensation for heat transfer between high- and low- temperature interface in the void cavity.

The described heat transfer process in thermal storage unit is transient. If all the above are considered, it is very difficult to solve the problem. To find easy solutions and simplify the calculations, some assumptions are made as follows when the heat transfer model of the thermal storage unit is built.

(1) The temperature along heat pipe wall is uniform. To match the parametric heat absorption information of the heat pipe system, a thermal model of the TSD region containment canister for the thermally attached heat pipe is created. A single canister is sufficient to model the entire TSD, since all of the canisters have identical temperature profiles. This is due to the assumed symmetry in incident flux (all heat pipes operate identically) and the uniformity in the heat pipe wall temperature in the TSD region (all canisters on a given heat pipe operate identically). The model is simplified by the heat pipe wicking system, which, if operating properly, ensures an essentially uniform temperature around the circumference of the heat pipe. Thus, only a 2-D canister model (axial and radial) is required [3].

(2) The initial void volume of thermal storage unit is fixed. Void cavity distributes along the outer wall of thermal storage unit. During eclipse periods, PCM in the TSD region freezes. PCM releases latent heat to heat pipe. At the same time, PCM shrinks and void cavity appears. Thus void cavity appears as numerous smaller cavities randomly distributed in the canister. Since the temperature of PCM close to canister inner wall decreases more quickly than that of PCM close to canister outer wall, PCM close to canister inner wall freezes earlier. PCM in canister freezes gradually along the radial from inner wall to outer wall. Most of void cavities appear near the outer wall in canister [10]. In NASA project, the similar physical model is built by this kind of assumption [4]. The void ratio of PCM

(80.5%LiF-19.5%CaF_2) is 8%-22%. That is, the minimum of void ratio is 8%. The maximum of void ratio is 22%. During the sunlight periods, the thermal expansion of PCM caused by the melting of most of the PCM happens, and the void ratio must be less than the maximum. During the eclipse periods, the thermal contraction of PCM caused by the solidification of most of the PCM happens, and the void ratio must be more than the minimum.

(3) Heat convection of liquid PCM is ignored. Natural convection disappears under microgravity. Marangoni convection caused by surface tension exists. The order of magnitude for Marangoni convection is much smaller than that for natural convection. At the same time, Marangoni convection appears only near the void cavity when PCM melts during sunlight periods. The flow of liquid PCM caused by Marangoni convection is in small scale under microgravity. During eclipse periods, the temperature differences among different liquid PCM interfaces decrease to about 1 K quickly. So Marangoni convection driven by the temperature gradient of free surface disappears very quickly during eclipse periods. Therefore, heat convection under microgravity can be ignored during both sunlight and eclipse periods.

(4) The contact resistance between the thermal storage unit and heat pipe wall is ignored.

(5) The surfaces of all void cavities are diffuse reflection gray body. The shape of void cavity is usually twisted and irregular. Void cavity is almost full of PCM vapour. Different interfaces shield each other. So the calculation for radiation angle coefficient of void cavity is very complicated. In order to well simplify numerical simulation and save memory for the calculation of radiation heat transfer in void cavity, the surfaces of all void cavities can be considered as diffuse reflection gray body.

The physical model of the thermal storage unit of the heat pipe receiver is shown in fig. 1.

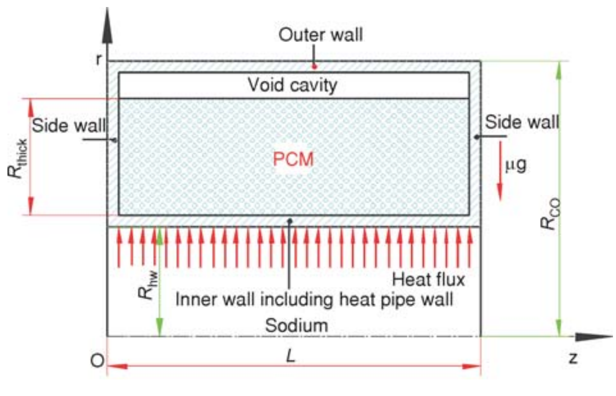

Figure 1. The physical model of PCM canister for the heat pipe receiver

Mathematical model [11]

Energy equation

For solidification-melting problems, the energy equation is:

$$\frac{\partial(\rho H)}{\partial t} + \nabla(\rho\vec{v}H) = \nabla(k\nabla T) + S \tag{1}$$

The enthalpy of the PCM is defined as

$$H = h + \Delta H \tag{2}$$

The sensible enthalpy of the PCM is defined as

$$h = h_{\text{ref}} + \int_{T_{ref}}^{T} C_p \,\text{d}T \tag{3}$$

The liquid fraction of the PCM is defined as:

$$\begin{cases} \beta = 0 & \text{if} \quad T < T_{\text{solidus}} \\ \beta = \dfrac{T - T_{\text{solidus}}}{T_{\text{liquidus}} - T_{\text{solidus}}} & if \quad T_{\text{solidus}} < T < T_{\text{liquidus}} \\ \beta = 1 & \text{if} \quad T > T_{\text{liquidus}} \end{cases} \tag{4}$$

In the case of multi-component solidification with species segregation; *i. e.*, solidification or melting with species transport, the solidus and liquidus temperatures are computed instead of specified.

$$T_{\text{solidus}} = T_{\text{melt}} + \sum_{solutes} K_i m_i Y_i \tag{5}$$

$$T_{\text{liquidus}} = T_{\text{melt}} + \sum_{solutes} m_i Y_i \tag{6}$$

where K_i is the partition coefficient of solute i, which is the ratio of the concentration in solid to that in liquid at the interface, Y_i – the mass fraction of solute i, and m_i – the slope of the liquidus surface with respect to Y_i. It is assumed that the last species material of the mixture is the solvent and that the other species are the solutes.

The solution for temperature is essentially an iteration between the energy equation (eq. 1) and the liquid fraction equation (eq. 4). Directly using eq. 4 to update the liquid fraction usually results in poor convergence of the energy equation. In FLUENT 6.2, the method suggested by Voller and Swaminathan [12] is used to update the liquid fraction.

Momentum equation

The enthalpy-porosity technique treats the mushy region (partially solidified region) as a porous medium. The porosity in each cell is set equal to the liquid fraction in that cell. In fully solidified regions, the porosity is equal to zero, which extinguishes the velocities in these regions. The momentum sink due to the reduced porosity in the mushy zone takes the following form:

$$S = \frac{(1-\beta)^2}{\beta^3 + \varepsilon} A_{\text{mush}} (\vec{\text{v}} - \vec{\text{v}}_{\text{pull}}) \tag{7}$$

The mushy zone constant measures the amplitude of the damping. The higher this value, the steeper the transition of the velocity of the material to zero as it solidifies. Very large values may cause the solution to oscillate.

The pull velocity is included to account for the movement of the solidified material as it is continuously withdrawn from the domain in continuous casting processes. The presence of this term in eq. 7 allows newly solidified material to move at the pull velocity.

Turbulence equation

Sinks are added to all of the turbulence equations in the mushy and solidified zones to account for the presence of solid matter. The sink term is very similar to the momentum sink term (eq. 7):

$$S = \frac{(1-\beta)^2}{(\beta^3 + \varepsilon)} A_{\text{mush}} \phi \tag{8}$$

In continuous casting processes, the solidified matter is usually continuously pulled out from the computational domain, as shown in fig. 2. Consequently, the solid material will have a finite velocity that needs to be accounted for in the enthalpy-porosity technique.

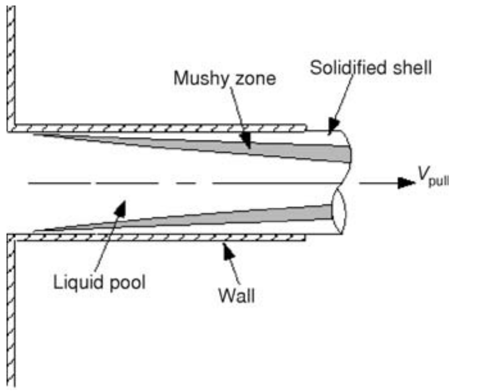

Figure 2. "Pulling" a solid in continuous casting

The enthalpy-porosity approach treats the solid-liquid mushy zone as a porous medium with porosity equal to the liquid fraction. A suitable sink term is added in the momentum equation to account for the pressure drop due to the porous structure of the mushy zone. For continuous casting applications, the relative velocity between the molten liquid and the solid is used in the momentum sink term (eq. 5) rather than the absolute velocity of the liquid. The exact computation of the pull velocity for the solid material is dependent on the Young's modulus and Poisson's ratio of the solid and the forces acting on it. A Laplacian equation is used to approximate the pull velocities in the solid region.

An approach to solution

Computational fluid dynamics (CFD) software (FLUENT 6.2) is used to simulate the heat transfer of thermal storage unit of the heat pipe receiver. Software (GAMBIT 2.1) is used as a pre-processor. In FLUENT 6.2, the separate solver is used to solve the equations. Finite control volume (FCV) method is used to solve governing equations. Thus, non-linear differential equations are linearized. Alternate direction iteration (ADI) method is used as a method of solution.

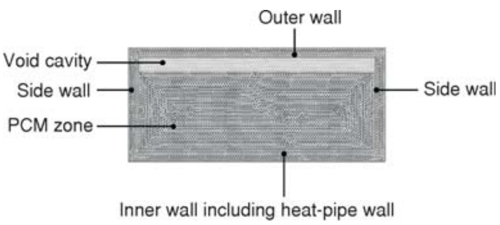

Figure 3. The mesh of the PCM canister for the heat pipe receiver

Mesh

Figure 3 shows the mesh of the PCM canister for the heat pipe receiver. Grid independence test has been carried out in order to ensure the mesh adaptation. Triangle mesh is adopted. Adjacent grid spacing is 0.25 mm. Mesh for PCM zone, void zone and PCM canister wall is 56 × 88, 8 × 88, and 6 × 100, respectively. Variable residuals are monitored to ensure the convergence criterion, which are fixed at 10^{-6} for the energy, and at 10^{-3} as a minimum for the continuity variables.

Boundary conditions

$$z = 0, \Delta H = 0; z = L, \Delta H = 0 \tag{9}$$

$$r = R_{CO,} \ \Delta H = 0; r = R_{hw}, Q = \pm Q(T_{hw}) \tag{10}$$

The boundary condition of heat pipe wall (canister inner wall) is taken as periodic various heat flux. The heat flux is function of heat pipe wall temperature. The heat flux varies with the temperature of heat pipe wall. The heat flux is positive during sunlight period and is negative during eclipse period. The canister outer wall and side wall are adiabatic with the outside. The adiabatic outside temperature (heat sink in orbital space) during these two periods is 200 K. In the void cavity, thermal radiation is considered.

The thickness of canister wall is 1.5 mm. the material of canister wall is Hayness188.

Initial conditions

The initial temperature is set to 950 K. PCM is solid. Each orbit cycle consists of 90 minutes. Each sunlight period consists of 54 minutes. Each eclipse period consists of 36 minutes.

The physical properties of the canister wall and phase change material are constant. The eutectic salt 80.5%LiF-19.5%CaF_2 selected as the PCM in our simulation. The physical properties of 80.5%LiF-19.5%CaF_2 are shown in tab.1.

Table1. Physical properties of phase change material

Physical properties	80.5LiF-19.5CaF_2
Melting point [K]	1040
Latent heat [Jkg^{-1}]	790000
Dynamic viscidity [Pa·s]	$2.21 \cdot 10^{-5}$
Density in solid [kgm^{-3}]	2670
Density in liquid [kgm^{-3}]	2100
Coefficient of heat conductivity in solid [$Wm^{-1}K^{-1}$]	5.9
Coefficient of heat conductivity in liquid [$Wm^{-1}K^{-1}$]	1.7
Specific heat capacity in solid [$Jkg^{-1}K^{-1}$]	1841
Specific heat capacity in liquid [$Jkg^{-1}K^{-1}$]	1970

Results and discussions

Influence of void ratio on phase change of thermal energy storage for heat pipe receiver under microgravity

Figure 4(a-d), shows the liquid fraction distribution in PCM canister at the end of sunlight when void ratio is 0, 8%, 15%, and 22%, respectively. As it can be seen, the minimum liquid fraction of the PCM in the canister is 77%, 74%, 68%, and 65% when void ratio is 0, 8%, 15%, and 22%, respectively. The melting ratio of thermal storage unit decreases as void ratio increases. The utility of PCM decreases as void ratio increases. Void cavity influences the process of phase change greatly. Void cavity prevents the procedure of melting of thermal energy storage. The latent heat of PCM can not be fully utilized at the end of sunlight when void cavity exits. Since the temperature of heat pipe wall is uniform along the axial, the direction of heat transfer in canister is from inner wall to side wall, then from side wall to outer wall. The temperature of canister wall is higher than that of PCM zone along the axial. PCM (close to canister's side wall) melts earlier and proceeds inward along the axial during sunlight periods. The reason is that the thermal resistance of PCM is higher than that of canister wall material (Hayness188 is cobalt-base super alloy). The thermal resistance

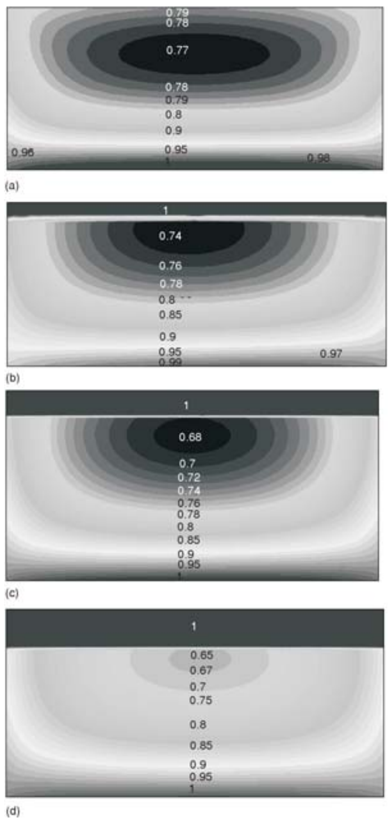

Figure 4. The liquid fraction distribution of PCM canister at the end of sunlight; (a) without void cavity considered, (b) when void ratio is 8%, (c) when void ratio is 15%, (d) when void ratio is 22%

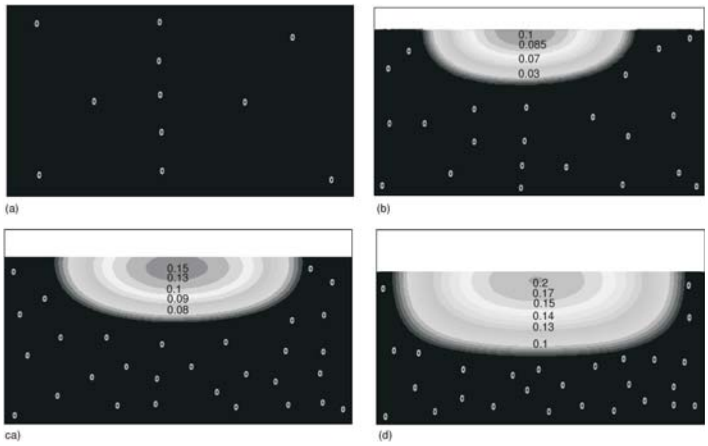

Figure 5. The liquid fraction distribution of PCM canister at the end of eclipse; (a) without void cavity considered, (b) when void ratio is 8%, (c) when void ratio is 15%, (d) when void ratio is 22%

of void cavity is much higher than that of canister's side wall. Canister's side wall has a great effect on heat transfer of PCM canister.

Figures 5(a-d) shows the liquid fraction distribution in PCM canister at the end of eclipse when void ratio is 0, 8%, 15%, and 22%, respectively. As it can be seen, PCM freezes totally without void cavity considered. The maximum liquid fraction of the PCM in the canister of the PCM in the canister is 10%, 15%, and 20% when void ratio is 8%, 15%, and 22%, respectively. The freezing ratio of thermal storage unit decreases as void ratio increases. The thermal storage ability of PCM decreases as void ratio increases. PCM can not release latent heat totally at the end of eclipse when void cavity exits. Void cavity influences the proceeding of phase change greatly. Void cavity prevents the procedure of solidification of thermal energy storage. Since the thermal resistance of void cavity is much higher than that of PCM canister wall, void cavity prevents the heat transfer from PCM zone to canister wall during eclipse periods. Since void cavity has the effect of thermal insulation, PCM in the TSD region freezes firstly on inner wall and side wall, then extends gradually towards the middle of PCM canister.

Void cavity enhances the temperature difference between sunlight and eclipse. Void cavity prevents heat transfer during both sunlight and eclipse periods. The temperature gradient of PCM zone is very significant with the effect of void cavity. So the thermal stress of heat pipe receiver may increase, and the life may decrease.

Comparison and analysis of simulated results and experimental results

Based on the specifications and limitations of the experimental conditions at the national aerospace laboratory (NAL) in Japan, a test model is designed and fabricated to evaluate performances [13]. A test of heating characteristics and thermal cycle of the heat receiver system is performed. The energy storage medium in thermal energy storage to provide thermal energy into bottoming devices during eclipse mode is investigated. TES is coupled at an intermediate position of the heat pipe. LiF-CaF_2 (80.5-19.5 mol%, melting point: 763 °C) PCM is used as a latent thermal energy storage medium for its high latent heats of fusion. TES is divided into 4 canisters connected in series. The periodic sunlight and eclipse modes occur for approximately 40 and 50 minutes, respectively.

Since heat pipe wall is tightly contacted with the outer surface of inner wall the temperature for outer surface of inner wall can be considered as that for heat pipe wall. Figure 6(a) shows a comparison between our numerical results and experimental ones of NAL [13]. Figure 6(b) shows the melting ratio of PCM during 0~20 orbital periods by our simulation. It can be seen that the corresponding temperature differences between our simulation and experiment of NAL in Japan on both heat-pipe wall and PCM-canister outer wall are small. The temperature change trend of our numerical results agrees with experimental ones of NAL in Japan. Therefore, our numerical results are credible. The model of the thermal storage unit for the heat-pipe receiver built in this paper is accurate and reasonable. As it can be seen, a heating cycle is stable, and temperature fluctuation both on heat-pipe wall and in thermal storage unit remains less than 13 K throughout a sunlight and eclipse cycle. These data indicate that the PCM contained in the integrated heat pipe performs an averaging function of heat loadings. The temperature of heat-pipe wall is nearly equal to the values in the PCM-canister throughout an orbital period. The maximum melting ratio is 92% during a sunlight and eclipse cycle. The utility of PCM is essentially improved. Therefore, the thermal performance of the heat pipe receiver is stable and reliable.

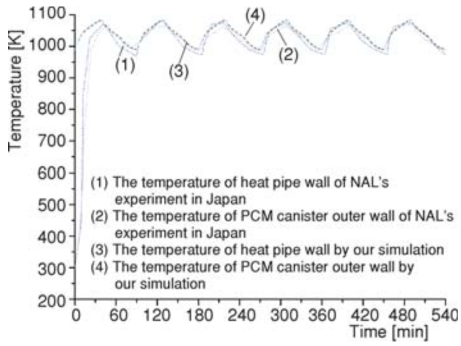

Figure 6(a). A comparison between our numerical results and experimental ones of NAL in Japan[13]

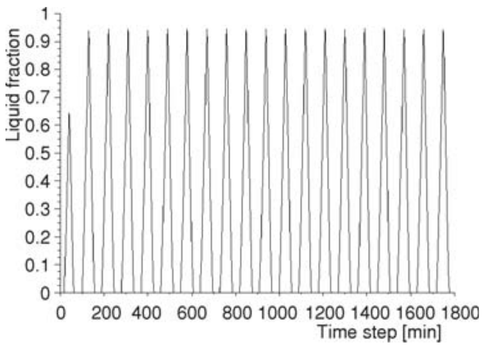

Figure 6(b). Melting ratio of the PCM in the canister by our simulation

Conclusions

In summary, influence of void ratio on phase change of thermal storage unit for heat pipe receiver under microgravity is numerically analyzed. Numerical results are compared with the experimental ones of NAL in Japan.

- In the comparison, it can be seen that the corresponding temperature differences and temperature-change trend of our numerical results agree with the experimental ones of NAL in Japan. Therefore, the numerical simulation of the thermal storage unit for the heat pipe receiver in this paper is accurate and reasonable.
- The melting ratio during sunlight periods and the freezing ratio during eclipse periods of thermal storage unit decrease as void ratio increases. The utility of PCM and the thermal storage ability of PCM decrease with the improvement of void ratio. Void cavity influences the process of phase change greatly. Void cavity prevents the procedure of melting and solidification of thermal energy storage. The latent heat of PCM can't be fully utilized at the end of sunlight and PCM can't release latent heat totally at the end of eclipse when void cavity exits. Since the thermal resistance of void cavity is much higher than that of PCM canister wall, void cavity prevents the heat transfer between canister's wall and PCM zone.

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Nomenclature

A – area, [m^2]
A_{mush} – the mushy zone constant
C_p – specific heat capacity, [Jkg^{-1}K^{-1}]
H – enthalpy, [Jkg^{-1}]
ΔH – latent heat of phase change, [Jkg^{-1}]
h – sensible enthalpy, [Jkg^{-1}]
k – thermal conductivity, [Wm^{-1}K^{-1}]
μg – micro gravitational acceleration, [m^{2}s^{-1}]
P – pressure, [Pa]
Q – heat flux, [W]
R_{thick} – radial thickness of PCM, [m]
r – radial length, [m]
S – source terms, [–]
T – temperature, [K]
t – time, [s]
$\vec{v}$ – fluid velocity, [ms^{-1}]
$\vec{v}_{\text{pull}}$ – the pulling velocity, [ms^{-1}]
z – axial length, [m]

Greek symbols

β – liquid volume fraction
ε – a small number (0.001) to prevent division by zero
ϕ – turbulence quantity
ρ – density, [kgm^{-3}]

Subscripts

CO – canister outer wall
hw – gas
P – constant pressure
Ref – reference

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