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PII: S2451-9049(19)30002-2
DOI: <https://doi.org/10.1016/j.tsep.2019.03.007>
Reference: TSEP 321

To appear in: *Thermal Science and Engineering Progress*

Received Date: 1 January 2019
Revised Date: 19 January 2019
Accepted Date: 8 March 2019

Please cite this article as: M.S. Mahdi, H.B. Mahood, A.F. Hasan, A.A. Khadom, A.N. Campbell, Numerical Study on the Effect of the Location of the Phase Change Material in a Concentric Double Pipe Latent Heat Thermal Energy Storage Unit, *Thermal Science and Engineering Progress* (2019), doi: <https://doi.org/10.1016/j.tsep.2019.03.007>

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Numerical Study on the Effect of the Location of the Phase Change Material in a Concentric Double Pipe Latent Heat Thermal Energy Storage Unit

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Abstract

Latent heat thermal energy storage (LHTS) represents an intriguing solution to the problem of variability of supply that exists in solar thermal power systems. One such storage system consists of a double pipe, where a phase change material (PCM) is enclosed in either the central pipe, or the annulus surrounding it. The heat transfer fluid fills the other void in the system. Whether the PCM is used in the central pipe or the annulus could, potentially, significantly alter the thermal performance of the system. Thus, a comparison between the PCM mounted in the annulus (case A) and the inner tube (case B) was conducted numerically, to investigate the advantages and disadvantages of each case with regard to the melting and solidification performance. A horizontal double pipe latent heat thermal energy storage device was considered. The numerical simulation solved the transient balance equations in a two-dimensional system. The enthalpy-porosity method was used to simulate the phase change and the Boussinesq approximation, which accounts for the small changes in density that drive natural convection, was applied. The effect of the initial temperature of the heat transfer surface (HTS) on the sensible and latent heat changes of the model PCM, RT-50, was tested for both the melting and solidification processes. Aiming to assess the differences in the storage performance, the average PCM temperature, the liquid fraction, and the average velocity of both the melting and solidification processes were examined. The results of the simulation showed that for both cases, convection was dominant after only a short period of the melting process. The melting time was significantly different in the two cases, i.e. it was shorter in case B than case A by almost 50%. Furthermore, an increase in the temperature of the HTS by 5 °C notably affected the melting time of both cases by as much

as 20%. This effect was more pronounced in case B, which had a melting time which was 41% shorter than case A. Finally, the results revealed that the solidification process in case A was more rapid than case B with the total solidification time of case A being lower by 43.4%.

Keywords: Double pipe LHTS, effect of PCM configuration, numerical technique, velocity distribution, PCM melting and solidification

1. Introduction

The continued development of the world is being seriously restricted by an energy crisis, and its associated environmental issues. According to the U.S. Energy Information Administration, energy use around the world will reach up to 1.8×10^{11} MWh in 2020 and about 2.3×10^{11} MWh in 2035 [1]. Conventional fossil fuels, such as oil, gas and coal, remain the main sources of energy that are currently exploited. Energy conversion from these sources is inefficient (about 31% [2] of the gross amount of fossil fuels is lost during the conversion process), and poses many environmental problems. Chief amongst these is the emission of greenhouse gases like CO₂. It is estimated that the total emissions of CO₂ that result from energy production could increase from the current value of 30.6×10^9 tpa to approximately 43.2×10^9 tpa in 2035 [1]. Furthermore, the price of oil, which is the most significant source of energy from fossil fuels, could rise to 120-140 US \$/ barrel from 2020 onward [1].

To mitigate, or even eliminate, these problems it is essential to utilize sustainable sources of energy. Different alternative or renewable energy resources, such as solar, wind and geothermal could potentially be exploited. Among these sources, solar seems the most promising one for widespread deployment; however, the energy can obviously only be harvested during the daytime. To overcome this limitation, and therefore to create a reliable and stable source of power, energy storage should be coupled with a solar system.

Thermal energy storage (TES) is a concept in which excess thermal energy is stored for use when it is necessary. Generally, thermal energy storage can be either (i) a sensible heat system or (ii) a latent heat system, depending on the storage medium that is used. In sensible thermal energy storage, no phase change occurs in the storage medium over the range of

temperatures required for storage. In spite of their low specific heats, concrete and ceramic are the most common materials that have been investigated as storage media. This is due to their good thermal conductivity and low cost. Molten salts and mineral oils are exploited in concentrated solar applications, such as in solar towers [3, 4]. The principal drawback of such systems is that to store a large quantity of energy usually requires a large volume of the medium.

In contrast to sensible heat thermal energy storage, latent heat thermal storage materials undergo a change of phase during both the charging and discharging of the storage system. Solid-liquid phase change materials (PCMs) have been found to be more efficient than liquid-vapour and solid-solid ones [5]. Although a wide range of materials are known to have a moderate to high heat of fusion (and therefore can absorb a large amount of energy per unit mass), no PCM yet tested has all of the characteristics that are desirable for universal deployment. Therefore, the selection of PCM for each application has to be considered carefully. The main factor limiting the use of PCMs is their low thermal conductivity (typical values are in the range 0.2 to 0.8 W / m K) [1]. Therefore, enhancing the thermal conductivity of PCMs is crucial to improving thermal storage efficiency. Attempts have been made to improve the thermal performance of latent heat thermal storage (LHTS) by improving the thermal conductivity of the PCMs (e.g. adding nanomaterials [6-13]) or by improving the storage design (e.g. adding fines [14-20]). It has been shown that the design based improvement technique is more effective, and can enhance the thermal efficiency of the TES by up to 80%.

The double pipe configuration is a design which has been proposed to enhance the thermal performance of LHTS. Simply put, it provides a large heat transfer area by exploiting an annulus between two cylindrical tubes, which can either be concentric, or offset. Sari and Kaygusuz [21] studied experimentally the thermal characteristics of both the melting and solidification processes of lauric acid in a double pipe thermal energy storage system. They observed that the Reynolds and Stefan numbers influenced the average heat transfer coefficient and the heat flow during the melting process, more so than during the solidification process. Medrano et al. [22] reported experiments which measured the heat transfer and thermal performance of five different designs of small latent heat thermal storage devices during both the charging and discharging processes. They found that double pipe storage filled with PCM and embedded in a graphite matrix was more efficient than the others they studied.

With the aim of understanding the behaviour of the PCM during the melting process in an annular space, Shokouhmand and Kamkar [23] simulated the natural convection and the formation of a melting front in a finned double pipe LHTS. Their results showed that the PCM melting process they considered was significantly improved by using longitudinal fins. Xiaohong et al. [24] investigated, both experimentally and numerically, the thermal performance of a composite PCM in a double pipe LHTS. The composite PCM consisted of 70% by mass paraffin wax and 30% graphite. The results indicated that the thermal energy storage capacity of this composite PCM was 86.6% larger than that when paraffin alone was used. Jesumathy et al. [25] measured the melting and solidification of paraffin wax in a horizontal double pipe thermal storage system. Their results confirmed that convection and conduction were the controlling mechanisms for the melting and solidification processes respectively. Additionally, when there was an increase or decrease in the inlet temperature of the heat transfer fluid (HTF) by 2 °C, the rate of heat transfer during the melting and solidification processes increased by 25% and 11% respectively. The total melting time was improved by 31% when the initial temperature of the HTF rose from 70 °C to 74 °C. Pahamli et al. [26] investigated numerically the melting of RT50 in the gap between the two cylinders in a double pipe LHTS unit. The eccentricity (downward displacement of the inner pipe) and the initial temperature and flowrate of the HTF were tested in their model. The results indicated that the movement of the inner pipe downward resulted in a reduction of the melting time by up to 64%.

Whilst there are a considerable number of studies that have considered the double pipe LHTS, the effect of the PCM location (i.e. in the annulus or central pipe) on the thermal characteristics of the storage has not been adequately discussed. Therefore, the effect of the PCM (paraffin type RT-50) location on the thermal performance of the unit was investigated numerically using ANSYS FLUENT 15. The simulation was performed for the PCM undergoing both melting and solidification. The velocity distribution within the liquid PCM was investigated to elucidate its thermal behaviour. Finally, the effect of HTS temperature on the PCM heat transfer characteristics was studied.

2. Physical model

To investigate the effect of the location of the PCM on the thermal behaviour of the double pipe LHTS, two different storage designs were considered. Firstly, the case where the PCM is in the annulus (case A) was considered. Here, the heat transfer surface is located on the inner

surface of the PCM. Secondly, the case where the PCM is in the central pipe was considered (case B). Here, the same heat transfer surface is obviously on the external surface of the PCM. The two configurations are shown schematically in Figure 1. The two designs have the same dimensions (diameters) and are assumed to be constructed of the same materials. The cross sectional area (and thus volume in a 3D system) occupied by the PCM in both cases was considered to be equal (19.6 cm^2) and the diameter of the inner tube and the shell were therefore set as 5 and 7.07 cm, respectively. Paraffin of type RT-50 was used as the PCM; its thermophysical properties are given in table (1) [27, 28].

Figure 1: A sketch of the layouts considered with the PCM located in the annulus and the PCM located in the central tube

Table 1 The physical properties of the PCM (Paraffin, type RT-50) [27, 28]

Property	Melting temperature range ($^{\circ}\text{C}$)	Latent heat (J/kg)	Specific heat (J/kg K)	Density (kg/m^3)	Thermal conductivity (W/m K)	Dynamic viscosity (Pa s)	Thermal expansion coefficient, β ($1/\text{K}$)
Value	45-51	168000	2000	800	0.2	0.004	0.0006

3. Numerical model

3.1 Governing equations

In this study, the enthalpy-porosity method [29] was used to simulate the phase change. The properties of the PCM were assumed to be constant during the phase change process. The model was transient and buoyancy effects were modelled via the Boussinesq approximation [16, 28, 30]. Thus, the density was considered constant in all terms, barring the buoyancy term in the equations for conservation of momentum, where it was assumed to vary linearly with temperature. This assumes that temperature rises in the system are much smaller than the absolute temperature; this assumption is commonly made in the study of buoyant flows in a wide variety of contexts [31-33].

The governing equations of the model can be written, in Cartesian form, as follows:

Conservation of Mass:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (1)$$

It should be noted that the continuity equation appears in its incompressible form due to the adoption of the Boussinesq approximation.

Conservation of Momentum in the x-direction:

$$\rho \left(\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} \right) = \mu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) - \frac{\partial P}{\partial x} + S_x \quad (2)$$

Conservation of Momentum in the y-direction:

$$\rho \left(\frac{\partial v}{\partial t} + u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} \right) = \mu \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) - \frac{\partial P}{\partial y} + S_y + \rho g \beta (T - T_o) \quad (3)$$

The enthalpy-porosity approach treats the mushy zone as a porous medium. The porosity in each element is assumed to be equal to the liquid fraction of that element. The momentum sink terms S_x and S_y due to the reduced porosity in the mushy zone are [29]:

$$S_x = A_{mushy} \frac{(1 - \lambda)^2}{\lambda^3 + 0.001} u \quad (4)$$

$$S_y = A_{mushy} \frac{(1 - \lambda)^2}{\lambda^3 + 0.001} v \quad (5)$$

where λ is the liquid fraction that varies between 0 and 1, depending on PCM temperature:

$$\lambda = \begin{cases} 0 & T < T_{solid} \\ \frac{T - T_{solid}}{T_{liquid} - T_{solid}} & T_{solid} < T < T_{liquid} \\ 1 & T > T_{solid} \end{cases} \quad (6)$$

187

188 The coefficient A_{mushy} is the mushy zone constant which is fixed at $10^5 \text{ kg/m}^3 \text{ s}$ in this study.

189 This value has been previously shown to yield solutions which compare favourably with
190 experimental measurements [28].

191

192 Conservation of Energy:

193

$$\rho \left(\frac{\partial H}{\partial t} + \frac{\partial}{\partial x}(uH) + \frac{\partial}{\partial y}(vH) \right) = \frac{\partial}{\partial x} \left(k \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(k \frac{\partial T}{\partial y} \right) \quad (7)$$

195

196 In the equations above ρ , P , β , T_o , t , μ , g and k are density, pressure, thermal expansion
197 coefficient, reference temperature, time, dynamic viscosity, acceleration due to gravity and
198 thermal conductivity, respectively. Finally, H is the total enthalpy which is made up of a
199 combination of sensible and latent heat contributions:

$$H = h + \Delta H \quad (8)$$

201

202 where the specific enthalpy is:

203

$$h = h_o + \int_{T_o}^T C_p dT \quad (9)$$

205

206 and the enthalpy variation due to the phase change is:

207

$$\Delta H = \lambda L \quad (10)$$

209

210 where L is the latent heat of fusion of the PCM. Thus, this contribution varies between zero,
211 for solid PCM, and L , for liquid PCM.

212

213 3.2 Initial and boundary conditions

214

215 The initial temperature was assumed to be spatially uniform at 30 °C for melting, and 70 °C
216 for the solidification processes. The HTS temperature was assumed to be uniform at 70 °C,
217 75 °C and 80 °C for the melting process and at 20 °C, 25 °C and 30 °C for the solidification
218 process. The HTS (see fig. 1) has a no slip condition applied and is assumed to remain at a
219 constant temperature. The outer surface of the double pipe was considered to be well-lagged
220 and therefore adiabatic.

221

3.3 Numerical approach

To solve the governing equations numerically the CFD package ANSYS FLUENT R15 was used. A second-order upwinding scheme was applied to solve the energy and momentum equations and the PRESTO Scheme [34] was applied for the pressure. A coupled algorithm [34] was applied to resolve the coupling between velocity and pressure. In order to enhance the convergence stability, the under-relaxation factors of 0.75, 0.75, 0.9 and 1 were considered for the momentum, pressure, liquid fraction and energy respectively.

3.4 Mesh and time step verification

The finite volume numerical method [35] was utilised in FLUENT. Multiple zones were created and the mapped face mesh method was applied to cases A and B. To achieve a reliable solution, the results must be independent of both the mesh density and the time step. The effects of varying mesh density and time step on the average temperature of the PCM, which was performed for both cases A and B but shown only for case B for brevity, are shown in Fig. 2 and Fig. 3, respectively.

In Fig. 2 are shown the curves showing the temporal variation of the average temperature of the PCM for meshes with 550, 1070, 2203 and 4520 elements. It is clear from Fig. 2 that the curves for each mesh are qualitatively similar; however there are significant quantitative differences. It is evident that these quantitative differences become insignificant when the number of elements in the mesh increased to 4520. Figure 3 presents the curves of the evolution of the average temperature generated by the simulation using a time step of 0.2, 0.1 and 0.05 s. Clearly, there is only a slight change in result when the time step is reduced to 0.05 s from 0.1 s. Therefore, a mesh of 2203 elements and time step of 0.1 s were chosen to study the thermal behaviour of the PCM for both cases.

Figure 2: The effect of varying the number of elements in the mesh on the evolution of the average temperature of the PCM, for case B, using a time step of 0.1 s.

Figure 3: The effect of varying the time step on the evolution of the average temperature of the PCM, for case B, using mesh with 2203 elements.

3.5 Model validation

To ensure that the numerical model developed herein is sufficiently accurate to yield significant physical insight, the calculated PCM liquid fractions were compared with previously experimental published results [36] for both the PCM local temperature and liquid fraction, as shown in Figs. 4A and 4B respectively. It is clear that there is excellent agreement between the calculated results and the experimental data of [36]. Thus, the numerical model can be confidently used to simulate the thermal performance of the PCM in the LHTS unit.

Figure 4A: A comparison of the numerical model with the experimental results of [36] of the local PCM temperatures at three different locations used throughout the study [36]; t12: at bottom, t22: at middle and t32: at top of the storage

Figure 4B: Further validation of the present simulation results with the experimental results of [36] for the PCM melting fraction; (a) experimental data of [36], and (b) Present model

4. Results and discussion

4.1 Melting Process

Energy storage systems which exploit latent heat would preferably have a short melting time to be practical. Therefore, investigations have focussed on enhancing this process, to make this time as short as possible. As discussed above, different configurations have been suggested and tested, mainly numerically, elsewhere in the literature.

With the aims of further understanding the fundamental science, and improving the design of the storage system, this work examines the effect of the location of the PCM in the double pipe LHTS on the melting process. In the first case (case A), the PCM is located in the annulus between the concentric pipes. In the second case (case B), the PCM was located only in the internal pipe. In both cases the amount of the PCM (cross sectional area in the 2D case,

corresponding to equal volumes in 3D) is the same. Furthermore, in both cases the PCM and heat transfer fluid are separated by the same wall, i.e. the wall of the inner pipe. Thus, the performance and heat transfer characteristics of each case can be compared directly.

The variation with time of the liquid fraction, in both cases, appears in Fig. 5, which is a colour map of the liquid fraction of the PCM during the melting process when the heating surface is held at a constant temperature (70 °C). The variation of the total liquid fraction with time is shown in Fig. 6, and velocity maps are shown in Fig. 7. When the PCM is found in the annulus (case A), the melting process is first uniformly distributed around the circumference of the heat transfer surface (Fig. 5(a)). Once a sufficiently thick layer of liquid has developed, the temperature gradients within the liquid result in buoyancy forces which move the liquid. The effect of this initiation of convection can be seen in Fig. 6 as a distinct reduction in the rate of melting at $t \sim 1$ min. The warm liquid rises, and thus a thicker layer of liquid can be seen towards the top of the annulus in Fig 5(a). The fluid from the bottom of the annulus flows upwards, and several small recirculation cells can be seen at the top of the annulus in Fig 7(a). It can be seen from the $t = 10$ min image in Fig. 5(a), that eventually the entire PCM at the top of the annulus is completely melted. As the fluid moves down the outer wall, it loses energy. Thus, close to the outer wall, the solid layer reaches higher up in the annulus. This behaviour is characteristic of natural convection in circular or spherical geometry, and is evident in Fig 7(a). For $t > 20$ mins, it is clear that the melted PCM is confined to the bottom of the annulus. There is also evidence of significant horizontal stratification in temperature, which is again characteristic of flows in which natural convection is significant, as is the stagnant, relatively cold zone at the bottom of the pipe [31, 33]. Figure 7(a) also indicates that the liquid towards the top of the annulus is nearly stagnant after it has completely melted, with only minimal flows near to the solid liquid boundary. For case B, the behaviour is somewhat different. The melting process begins by the formation of a thin layer close to the heat transfer surface. As in case A, the thermal gradients induce natural convection, which makes the liquid accumulate towards the top of the pipe. This effect is again clear in Fig 6. Unsurprisingly, given the similarities in the initial development of the thin liquid film on the heat transfer surface in both cases, the initial temporal development of the liquid fraction in both cases is very similar. There is also clear evidence of a horizontal liquid film accumulating towards the bottom of the pipe. Interestingly, Fig 7(b) clearly shows the existence of recirculation cells (and hence natural convection) in this region. Thus, heat is being transferred to the entire melting core by convection. The shape of

the flow field remains similar throughout, with cold fluid descending at the surface of the solid core, and warm fluid rising up the heat transfer surface. Due to this consistency of flow field, the shape of the solid core remains relatively constant until the melting process is complete.

A comparison of the two cases reveals that case A has a longer melting time than case B, with total melting times of 43 min and 18 min, respectively. This is clearly a result of the different flow patterns discussed above. Furthermore, in case A the solid PCM accumulates and is confined to the bottom of the cylinder. As a result, the heat transfer to this cold, unmelted section is controlled purely by conduction. In contrast, in case B the PCM is surrounding by the hot surface which results in the PCM melting circumferentially. Thus, heat transfer to the unmelted core is always by convection, and is therefore more rapid than the conduction controlled melting in case A. This finding is confirmed quantitatively by Fig. (6). This is completely consistent with the shape of the velocity contour of the melted PCM, as shown in Fig. 7 and supported by the calculated value of Rayleigh number ($Ra = \frac{g\beta(T_s - T_m)D_h^3 Pr}{\nu^2}$) of 2.34×10^7 and 1.66×10^6 , for the PCM in the inner tube (case B) and the PCM in the annulus (case A) respectively. This is consistent with the values given by Zhang and Li [37].

Figure 5: Colour maps of liquid fraction for the PCM melting process for a) case A, where the PCM is in the annulus and b) case B, where the PCM is in the inner pipe

Figure 6: The variation of the liquid fraction of PCM with time for a constant heating surface temperature (70 °C) for both cases A and B

Figure 7: Colour maps showing the magnitude of the velocity in the melted PCM during the melting process for both cases; a) case A and b) case B

The variation of the average temperature of the PCM with time is shown in Fig. 8. The figure also includes the development of the average velocity of the melted PCM with time. Both geometries under study are presented for the case where the HTS is held at 70 °C. Obviously, the average temperature of the PCM has a similar trend in both cases, and is quantitatively similar during the first 18 mins. Interestingly, the liquid PCM is slightly hotter for the case where it is in the annulus. Thus, it would appear that more of the heat transferred to the material goes to the sensible heating of the liquid fraction, rather than to the melting process, as compared with the case where the PCM is in the inner pipe. This would again explain why the PCM takes longer to completely melt in the annulus case. It can be seen from Fig. 8, that the PCM in the inner tube is completely melted after ~ 18 min, but the PCM in the annulus continues to have a significant solid fraction. It is interesting to note also, that the velocity of the liquid PCM contained in the inner tube decays rapidly to 0. This would indicate that the temperature is largely spatially uniform within the pipe, and therefore the temperature gradients that do exist are insufficient to overcome the viscous forces and generate significant natural convection. The convection that is present before this is vigorous, and hence supplies heat to the melting core at a more rapid rate. On the other hand, in case A (with the PCM in the annulus) the convection is more rapid initially but was shown in Fig. 7 to decay rapidly in the top half of the annulus, and only to have very limited effect near to the solid/liquid interface towards the bottom of the annular region. This would indicate that the top of the annulus is in thermal equilibrium with the HTS, and therefore the very slow increase in average temperature for $t > 20$ mins that is shown in Fig. 8 is due to the slow heat transfer, primarily by conduction, to the solid region at the bottom of the annulus (see Fig. 5).

Figure 8: Variation of the average temperature of the PCM and the average velocity of the resulting liquid with time for both cases under study (cases A & B) at constant surface temperature (70 °C)

The effect of the temperature of the heating surface on the overall PCM liquid fraction is illustrated in Fig. 9 for case A. It is clear that the higher heat transfer surface temperature results in the PCM melting more rapidly, as would be expected. This effect is more pronounced for case B in Fig. 10. Here, the liquid fraction increases in two, almost linear sections. For small times, where there is no convection and conduction transfers the heat to the melting thin film, then the behavior is similar for all temperatures considered. The curves diverge once convection has initiated, with a greater rate of melting being evident at higher temperatures. This supports the conclusion above, that convection plays a more significant role throughout the melting process in case B, than it does in case A.

Figure 9: The effect of varying the heating surface temperature on the development of the liquid fraction for case A, where the PCM is in the annulus

Figure 10: The effect of varying the heating surface temperature on the development of the liquid fraction for case B, where the PCM is in the central pipe

4.2 Solidification Process

The second phase in any latent heat thermal energy storage system is the solidification phase, when the energy stored during the charging or melting phase is delivered to a process. Thus, efficient solidification is also a requirement for an efficient process.

To assess the storage design, the heat transfer characteristics of the PCM during the solidification process were tested numerically. Fig. 11 shows the development of the local liquid fraction during the solidification process for the same geometric cases discussed above. Shown in Fig. 12 are the temporal developments of the velocity fields. Here, the heat transfer surface was maintained at 30 °C. In case A, it can be seen from Fig. 11 that the solid film begins to form around the heat transfer surface. As the liquid cools, it becomes denser and so begins to sink. Thus, the solid layer is thicker towards the bottom of the annulus. As would be expected, the last remaining liquid accumulates at the top of the annulus. This is of course a mirror image of the melting process. In case B, the solid film again begins to form at the surface. As natural convection becomes more significant, the colder fluid is convected towards the bottom of the pipe, where it solidifies. Thus, the solid layer is thicker at the bottom of the cylinder. Furthermore, the shrinking core of liquid is centred slightly above the centreline. It is interesting to note that the liquid region in Fig 11(b) remains approximately circular throughout. This is in contrast to what was seen in the melting process. The reason for this is apparent from inspection of Fig. 12. It is evident that in both cases, convection actually only plays a significant role in the initial stages of the solidification process. It establishes the general shape of the solidifying region, but then subsequently dies down. Therefore, conduction becomes the dominant mode of heat transfer for the rest of the solidification process.

It is also interesting to note, that unlike in the melting process, case A seems more efficient than case B. Complete solidification of the PCM, as shown in Fig. 13, took 51 min and 90 min respectively. In both cases, the PCM solidifies as layers around an approximately cylindrical surface. These layers act as a heat transfer resistance, due to the poor thermal conductivity of the PCM. Hence, conduction quickly becomes the dominant mode of heat transport and therefore controls the solidification process. However, the thickness of the solidified PCM layer in case A seems more uniform and thinner than that of case B and so there is a lower heat transfer resistance and thus a shorter solidification time. It also, of course, has a greater heat transfer area than the shrinking liquid core in case B.

Figure 11: Colour maps of liquid fraction for the PCM solidification process for a) case A where the PCM is in the annulus and b) case B where the PCM is in the inner pipe

Figure 12: Colour maps showing the magnitude of the average velocity in the solidified PCM during the melting process for both cases a) case A and b) case B

Figure 13: The variation of the liquid fraction of PCM with time for a constant heating surface temperature (30 °C) for solidification in both cases A and B

A further important parameter that can be used to assess the heat transfer characteristics of the PCM is the temperature during the solidification process. This can be seen in Fig. 14, which plots the variation of the average temperature of the PCM for both cases A and B. The figure also shows the variation of average velocity with time during the solidification process. In spite of the velocity profiles of both cases being similar, the time required for complete solidification of case B is longer than that required for case A, which is consistent with the findings above (Fig. 11 and 13).

Figure 14: Variation of average temperature of the PCM and the average velocity with time for both cases under study (case A & B) at a constant heat transfer surface temperature (30 °C)

The effect of the temperature of the cooling surface was found to be an important parameter that can affect the solidification process. Three different cooling temperatures (Fig. 15 for case A and Fig. 16 for case B) were tested so as to distinguish quantitatively how the temperature of the cooling surface can impact the total time required to achieve complete solidification of the PCM. It is obvious from the figures that decreasing the cooling surface temperature results in a shortening of the PCM solidification time, as would be expected.

Figure 15: Variation of PCM liquid fraction with time for the case (A) where the PCM is found in the annulus. Three surface temperatures are shown.

Figure 16: Variation of PCM liquid fraction with time for the case (B) where the PCM is found in the central pipe. Three surface temperatures are shown.

5. Conclusions

In this paper, a numerical simulation using the enthalpy-porosity model and the Boussinesq approximation for phase change and natural convection respectively was proposed to study the effect of the PCM location in horizontal double pipe latent heat thermal energy storage (LHTS) system. Aiming to assess the improvement of the melting and the solidification processes for the cases with the PCM in the outer shell (case A) and the PCM in the inner tube (case B), a comparison of factors including the average PCM temperature, the liquid fraction and the average velocity was performed. From the computational results, the following conclusions can be drawn:

- 1- At the beginning of the melting process, conduction was dominant for both cases under study. Once sufficient liquid was present, natural convection became dominant.
- 2- When compared to case A (the PCM in the annulus), when the PCM is in the inner pipe (case B) the effect of convection is more significant and hence increases the rate of melting. Thus, the melting time of the PCM in case B was shorter than that of case A by almost 50%.
- 3- For both geometric cases, increasing the HTS temperature leads to an increase in the temperature difference between the HTS and PCM. This results in increasing the driving force to melt the PCM and thus shortens the melting time. An increase of the HTS temperature from 70 °C to 75 °C and then from 75 °C to 80 °C results in a decrease in the melting time of 19% and 16% for case A and 20% and 15.4% for case B.
- 4- In the case of the solidification process, convection was dominant for only a short period at the beginning of the process, and then was superseded by conduction. In contrast to the melting process, case A was found to be superior i.e. it had a solidification time that was shorter than that of case B by ~ 43.4%. This was due to the high heat transfer resistance produced by the thicker solid PCM layer that developed in case B.

Given that the melting and solidification processes are favoured by different geometries, the design and optimization of the geometry merits significant further study.

513 **Nomenclature**

514	A_{mushy}	Mushy zone constant ($kg/m^3 s$)
515	C_p	Specific heat ($J/kg K$)
516	D_h	Hydraulic diameter (m)
517	g	Acceleration of gravity (m/s^2)
518	k	Thermal conductivity ($W/m K$)
519	H	Total enthalpy (J/kg)
520	h	Specific enthalpy (J/kg)
521	h_o	Enthalpy at reference temperature (J/kg)
522	L	Latent heat of fusion (J/kg)
523	P	Pressure (Pa)
524	Ra	Rayleigh number
525	S	Momentum source term (Pa/m)
526	T	Temperature (K)
527	T_{liquid}	Temperature of liquid PCM (K)
528	T_m	Melting temperature of PCM (K)
529	T_s	Surface temperature (K)
530	T_{solid}	Temperature of solid PCM (K)
531	T_o	Reference temperature (K)
532	t	Time (s)
533	t_{12}, t_{22}, t_{32}	locations; bottom, middle and top respectively
534	u	x velocity component (m/s)

535 v y velocity component (m/s)

536 x, y Cartesian coordinate (m)

537 Greek symbols

538 β Thermal expansion coefficient ($1/K$)

539 λ Local liquid fraction

540 μ Dynamic viscosity ($kg/m.s$)

541 ρ Density (kg/m^3)

542 ν Kinematic viscosity (m^2/s)

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Research Highlights

- Thermal performance of horizontal double pipe LHTS device was numerically considered
- Comparison between PCM mounted in annulus and inner tube of the LHTS was conducted
- Effect of initial temperature of HTS on PCM, RT-50, in melting and solidification was tested
- Average PCM temperature, liquid fraction, and average velocity of processes were examined
- Melting was shorter in case B than case A by almost 50% and longer in solidification by 43.4%
- Increase temperature of HTS by 5 °C decreases melting time of both cases by as much as 20%.