Sustainable Cities and Society 52 (2020) 101786

Contents lists available at ScienceDirect

Sustainable Cities and Society

journal homepage: www.elsevier.com/locate/scs

Experimental and numerical investigation of laminar heat transfer of microencapsulated phase change material slurry (MPCMS) in a circular tube with constant heat flux

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ARTICLE INFO

Keywords:
Microencapsulated phase change material slurry
Phase change
Heat transfer enhancement
Influence factors
Numerical investigation

ABSTRACT

Due to its large apparent specific heat during the phase change period, microencapsulated phase change material slurry (MPCMS) has been suggested as a medium for heat transfer. In this paper, the convective heat transfer characteristics of MPCMS flowing in a circular tube were experimentally and numerically investigated. The MPCMS is prepared by using the base liquid other than water. The heat transfer experiments and simulations of the slurry flow in the tube at constant heat flux were conducted and the accuracy of the experimental results were examined. The effect of the slurry concentration, Stephen number, inlet subcooling, particle diameter and Re number on heat transfer enhancement were analyzed. It is concluded that the mass fraction of microcapsule particle and the Stephen number have the most important influence on the heat transfer of MPCMS in laminar tube flow. Furthermore, in order to understand the phase change rate of MPCMS in laminar heat transfer from inlet to outlet, considering the influence of wall material of microencapsulated phase change material (MPCM) particles on heat transfer process, the mathematical models of pipeline and phase change particles were established respectively, and the accuracy of the models were verified. The simulation results showed that the phase change rate increases with the decrease of flow rate, the decrease of mass fraction of microcapsule particles and the increase of heat flux.

1. Introduction

Recently, a new technique has been proposed for utilizing phase change materials (PCM) in energy storage systems, heat exchangers and thermal control systems (El-Sebaii, Al-Ghamdi, & Al-Hazmi, 2009; Hawlader, Uddin, & Khin, 2003; Medrano, Yilmaz, & Nogués, 2009) where the PCM is microencapsulated and suspended in a conventional single-phase heat transfer fluid to form phase change slurries. Such slurries have large apparent specific heats during the phase change period, which enhances the heat transfer rate between the fluid and the tube wall. The slurry can serve not only as the thermal storage media but also as the heat transfer fluid, it can significantly improve the heat transfer efficiency, and then improve the energy utilization rate, and contribute to energy saving and emission reduction. Therefore, could have many potentially important applications in the fields of heating, ventilation and air-conditioning (HVAC), refrigeration and heat exchangers, etc, which is of great significance to the sustainable development of cities and society.

The flow and heat transfer characteristics of microencapsulated phase change material slurry (MPCMS) have been investigated in various theoretical and experimental studies (Charunyakorn, Sengupta, & Roy, 1991; Goel, Roy, & Sengupta, 1994; Sohn & Chen, 1981; Yamagishi, Takeuchi, & Pyatenko, 1999). Sohn and Chen (1981) observed enhanced thermal conductivity of solid–liquid slurry at a low flow velocity due to the effects of micro-convection around the solid particles. Charunyakorn et al. (1991b). presented that the main factors of its enhanced heat transfer were mass percent of the particles and the bulk Ste number, and made the conclusion that the heat transfer enhancement ratio of the fluid could be 2–4 times of that of water. Goel et al. (1994) claimed that the reduction of wall temperature rise up to 50% compared with a single phase flow in the same dimensionless conditions and experimental results were compared with those predicted by the numerical simulation of Charaunyakorn et al. (1991b). Based on past observations and research activities, MPCM shells seem not to have a significant impact on the heat transfer process. The experimental data from Yamagishi et al. (1999) showed that MPCMS approximately followed the Blasius equation. They also conducted several heat transfer experiments with constant heat flux and turbulent

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https://doi.org/10.1016/j.scs.2019.101786
Received 8 June 2019; Received in revised form 31 July 2019; Accepted 13 August 2019
Available online 17 August 2019

Nomenclature

C	Specific heat capacity, kJ/ (kg · k)
d	Pipeline diameter, m
h	Convection heat transfer coefficient, W/(m^2·k)
k	Heat transfer coefficient, W/(m^2·k)
L	Latent heat of phase-change microcapsule particles, kJ/kg
N	Number of microcapsules per unit volume
Pe	Berkeley number
Pr	Prandtl number
q	Heat flux, W/m^2
Q	Single microcapsule heat transfer, J
r_0	Pipe radius, m
Re	Reynolds number
S	Internal heat source term
St	Stephen number
T	Temperature, k
$\Delta\dot{T}$	Undercooling at the entrance of MPCMS into the pipeline
λ	Thermal conductivity, W / (m · K)

Subscript

0	Initial time
a	Average
b	Mixed liquid (slurry)
d	Pipeline
e	Effective
f	Based-phase
i	Inlet
P	Particle
w	Shell material
x	Pipeline location

conditions. Some experimental results showed that the phase change slurries enhanced the convective heat transfer with small increases in the pressure drop for some conditions (Choi, Cho, & Lorsch, 1992; Colvin, Mulligan, & Bryant, 1992; Yasushi, Hiromi, Alexander, & Naoyuki, 1999). The fluid could be analyzed as Newtonian when the microcapsule volumetric concentration was less than 25% (Charunyakorn, Sengupta, & Roy, 1991). The Nusselt number for the phase change slurry was from 1.5 to 4 times higher than for single-phase flow under ideal conditions. Experimental results for the dimensionless wall temperature with constant wall heat flux were 45% less than the ideal theoretical prediction because the slurry inlet temperature was less than the PCM phase change temperature and the phase change process occurred over a finite temperature range (Goel, Roy, & Sengupta, 1994; Zhang & Faghri, 1995b).

MPCM slurry has been studied in thermal storage and management systems (Kong, Alvarado, & Languri, 2013; Kurnia, Sasmito, Jangam, & Mujumdar, 2012; Taherian, Alvarado, Tumuluri, Thies, & Park, 2014). Kong et al. (2013) experimentally studied the characteristics of MPCM slurry in a helical coil heat exchanger under turbulent flow conditions. They observed the effects of phase change process in the heat transfer enhancement. Greater viscosity of MPCMs led to greater pressure drop. In addition, they observed that the overall heat transfer coefficient was enhanced by 17% using a 7.4% wt of MPCM slurry instead of water for 4000 De. This shows the contribution of phase change process in the heat transfer. The heat exchanger (HX) effectiveness and MPCM slurry performance are analyzed. The results reveal that when using MPCM slurry, the heat transfer ability improves while the heat transfer performance efficiency coefficient should be evaluated in the increment of MPCM concentration in the base fluid. Taherian et al. (2014) constructed an experimental setup to assess different parameters such as MPCM mass concentration, slurry flowrate, and the type of phase-change materials. They provided a Nusselt number correlation and concluded that the optimum heat capacity can be achieved during the phase change. Kurnia et al. (2012) studied the heat transfer involved in slurry MPCM flow in coiled square tubes in laminar flow regime. They used numerical modeling tool to analyze MPCM slurry flow behavior flowing through various configurations including conical spiral, in-plane spiral, and helical spiral. They reported enhancement in overall heat transfer performance when the MPCM concentrations increased from 1 to 10%.

Sabbah, Seyed-Yagoobi, and Al-Hallaj (2011) performed a numerical study to characterize the thermal boundary layer growth where they studied the influence of MPCM on the growth of the boundary layer at the thermal entry length extension. In another study by Kurnia, Sasmito, Jangam, and Mujumdar (2013), a computational analysis of coiled square tubes was explored. They simulated laminar flow of MPCMs through a square cross section of straight, conical spiral, in-plane spiral, and helical coils. Higher heat transfer performance due to the use of MPCM when compared to water was observed in this computational study. Rennie and Raghavan (2005) conducted an experimental study on the heat transfer characteristics of a coaxial heat exchanger. They found that the overall heat transfer coefficients increased with Dean number. Kong, Alvarado, Terrell, and Thies, (2016) performed experimental tests to investigate the flow and heat transfer characteristics of MPCMs as an enhanced heat transfer fluid in a fully instrumented heated helically coiled tube. Different mass fractions of MPCMs were tested and correlated using the Dean number. Their results showed that although heat transfer enhancement is restricted due to high viscosity and low latent heat of fusion of PCM materials used in MPCMs, MPCMs still shows heat capacity improvements when compared to its base fluid, water.

Although many theoretical and experimental studies have been carried out by experts and scholars at home and abroad, they couldn't form a consensus, or even conflict with each other. For instance, P. Charunyakorn (Charunyakorn et al., 1991a), Goel M (Goel et al., 1994) and Faghri Amir (Zhang & Faghri, 1995a) concluded that the mass fraction of microcapsule particles was an important factor affecting heat transfer, however S.K. Roy (Roy & Avantic, 1997) believed that the mass fraction of microcapsule particles has little effect on heat transfer; As well as, Goel M (Goel et al., 1994), Yasushi Yamagishi (Yasushi, 1999) and Alisetti Edwin L (Alisetti & Roy, 2000) came exactly to that conclusion that *St* is another major factor affecting convective heat transfer, but Wang Li (Wang, 2006) thinks that *St* had little effect on convective heat transfer; Furthermore, P. Charunyakorn (Charunyakorn et al., 1991a) thought that Re had little effect on heat transfer, nevertheless Wang Li (Wang, 2006) thought that Re had a great influence on heat transfer.

Many researchers use thermodynamic cycle system to do similar experiments (Alisetti & Roy, 2000; Charunyakorn et al., 1991a; Chen, Wang, Zeng, Zhang, & Wang, 2008; Choi, Cho, & Lorsch, 1994;Goel et al., 1994; Roy & Avantic, 1997). The heated slurry was cooled by refrigeration system and then flowed to the entrance of the experimental section and heated by heat source. The temperature control system of the refrigeration system indirectly controls the entrance temperature of the slurry. The thermal inertia of the temperature control method is large, and the temperature fluctuation is generally greater than (± 2℃). The stability and accuracy of the experimental data were discussed.

For the sake of further study of the influence of various parameters on convective heat transfer. In this paper, experimental and numerical methods were adopted to analyze the enhanced forced convective heat transfer of MPCMS laminar flow in a circular tube under constant heat flux. In order to ensure the reliability of the experimental results, the non-circulating pipeline was used in the test-bed, and the inlet temperature of the slurry was constant. The MPCMS was prepared by substituting the base liquid with the same density of microcapsule particles for water to improve the stability of MPCMS. In this paper, the heat transfer intensity of pipes with different mass fraction, particle size, *St* and inlet supercooling at the same Re and different Re were studied. Meanwhile, the phase change rate of MPCMS at the exit of pipes was studied by numerical calculation method. Mathematical models were established for the fluid and particle parts of MPCMS respectively, and the temperature field in the pipeline was solved.

2. Investigation of the properties of MPCMS

2.1. Appearance and particle size distribution of microcapsule particles

The MPCM particles used in this paper are commercial products of Microtek Laboratories Inc. The size of MPCM particles is micron, which is difficult to observe with eyes. Its shape can be clearly observed by scanning electron microscopy (SEM, COXEM, EM-30 Plus). Fig. 1 (a), (b) are SEM photos of microcapsule particles. Fig. 1 (a) is the microencapsulated particle with a radius of 5.000 µm and Fig. 1 (b) is the microencapsulated particle with a radius of 20.000 µm. It was observed that the microcapsule particles were smooth and spherical.

The particle size distribution of MPCM was defined as the mass percentage of particles in a certain particle size range. The particle size distribution of microcapsules with two nominal radii can be observed by laser particle size analyzer (Bettersize2000 LD), as shown in Fig. 2.

The experimental results showed that the actual average particle sizes of MPCM with nominal particle sizes of 20.000 and 5.000 µm were 16.360 and 4.216 µm, respectively, and their particle size distributions were concentrated at 16.70–21.28 µm and 3.062–3.902 µm, respectively.

2.2. Preparation of MPCMS

In this paper, the based-phase with a density equal to the phase change particle density of microcapsules (about 0.935 g/cm3) was prepared by mixing n-propanol (about 0.8 g/cm3) and water (about 1 g/cm3) at a volume ratio of 1.34:1. Afterwards, mixed with MPCM particles. The MPCMS was obtained by mixing 1 800 r/min in a shear emulsifier for 15 min and 500 r/min in a constant temperature magnetic stirrer for 3 h. The MPCMS was emulsion white in appearance and had good fluidity. As a result of the density of the two was approximately equal, the influence of gravity was greatly reduced, therefore, the physical stability was good, and the stratification phenomenon did not occur during 24 h, as shown in Fig. 3.

2.3. Measurement and analysis of thermal conductivity

Thermal conductivity is an important parameter of MPCMS as a heat transfer functional thermal fluid. In this paper, the thermal conductivity of MPCMS used in the experiment is measured by hotdisk. At the same time, the temperature of the MPCMS tested is maintained and controlled by water bath pot, so as to obtain the thermal conductivity of MPCMS at different temperature.

The thermal conductivity of MPCMS with mass fractions of 5.0%, 10.0%, 20.0% and 30.0% measured by Hotdisk was shown in Fig. 4.

It can be found that the thermal conductivity of MPCMS decrease with the increase of mass fraction, because the thermal conductivity of polymer and paraffin, which are the materials of microcapsule, are lower than that of based-phase. As well as, the thermal conductivity of MPCMS tend to increase with the increase of temperature.

2.4. Viscosity testing and analysis

Temperature is an important factor affecting the viscosity of MPCMS. The viscosity of MPCMS with different mass fractions at different temperatures was measured with rheometer (Anton Paar MCR102) when the diameter of MPCM particles was 5 µm and 20 µm. The specific values were as follows:

The results showed that the viscosity of MPCMS decreases with the increase of temperature and increases with the increase of the mass fraction of MPCM particles, and the viscosity of MPCMS with an average particle size of 20 µm was lower than that of MPCMS with an average particle size of 5 µm at the same mass fraction and temperature (Tables 1 and 2).

2.5. Experimental investigation of laminar heat transfer of MPCMS

2.5.1. The experimental platform construction and verification of accuracy

The laminar heat transfer system diagram in this paper was shown in Fig. 5. The experimental pipeline was stainless steel pipe with inner diameter of 4.0 mm and outer diameter of 6.0 mm. 18 K-type thermocouples were evenly attached to the outer wall of the pipeline at length of 3.5 m. The temperature measurement error of the thermocouples was tested to be within 0.1 ℃. There was a thermometer and a pressure sensor at the inlet and outlet of the pipeline to measure the inlet and outlet temperature and pressure of the slurry. All thermocouples, thermometers and pressure sensors were collected by data acquisition (Aglient34907A) and recorded by computer. The outer wall of the pipeline was uniformly coated by the heating zone with a power of 900.00 W. The ultimate aim is to simulate the boundary conditions of equal heat flux. The heating belt was connected to an endless continuous adjustable transformer to adjust the heating amount.

(Zeng, Wang, Chen, & Zhang (2009) used centrifugal pump as the

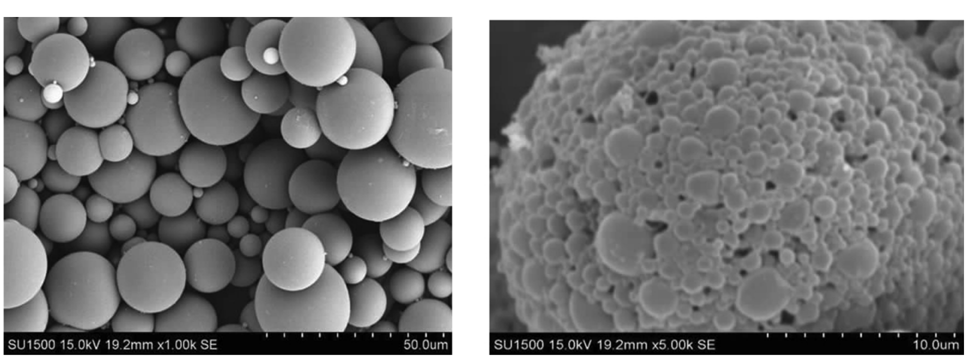
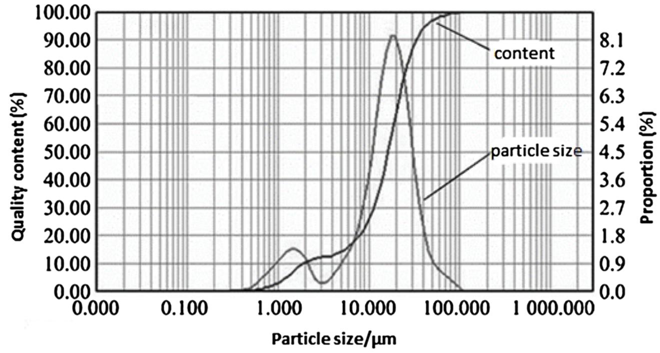
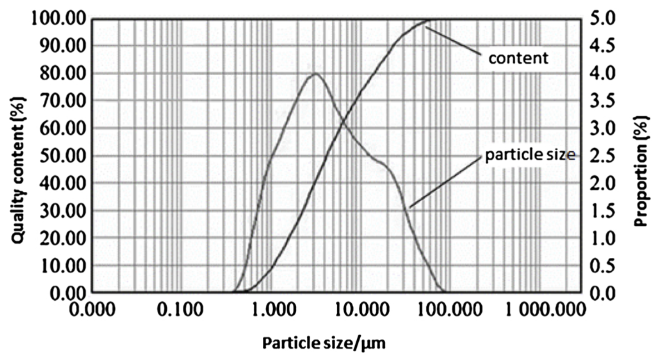


Fig. 1. Electron microscopic scanning image of MPCM particles: (a) Microencapsulated particles with nominal radius of 5.000 µm (b) Microencapsulated particles with nominal radius of 20.000 µm.
Nominal radius is 20.000 µm
Nominal radius is 5.000 µm



particle size/μm	Content %
0.500	0.2
1.000	3.13
2.000	10.21
5.000	14.28
10.000	25.49
20.000	63.60
45.000	95.83
75.000	99.20
100.000	99.94
200.000	100.00

(a) Nominal radius is 20.000μm



particle size/μm	Content %
0.200	0.00
0.500	0.33
1.000	0.68
2.000	25.97
5.000	55.21
10.000	72.79
20.000	86.98
45.000	97.91
75.000	99.90
100.000	100.00

(b) Nominal radius is 5.000μm

Fig. 2. Phase-change microcapsule particle size distribution.

Fig. 3. 24 -h static stability.

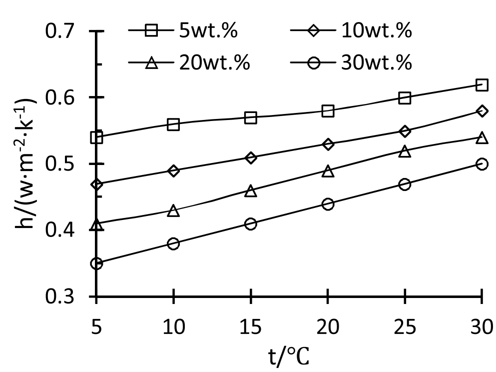


Fig. 4. Measurement of heat transfer coefficient of MPCMS with different mass fraction at different temperatures.

power device of the test-bed. At the beginning of the construction of the test-bed, centrifugal pump was used to provide power. However, during debugging, it was found that the power provided by the pump could not be stable due to some unknown instability of the force between the rotating blade and the liquid. When the pump works on the liquid, the temperature of the liquid rises, and the temperature of the inlet of the

Table 1
Viscosity of MPCMS with 5 μm particle size.

t/°C wt.%	15℃	20℃	25℃	30℃	35℃
5	3.66	3.04	2.64	2.28	2.17
10	6.84	5.55	4.49	3.83	3.13
15	9.31	7.40	6.27	5.54	4.70
25	45.63	36.21	27.86	21.51	17.52
30	152.38	141.51	79.45	71.39	68.33
35	490.06	406.56	315.39	259.79	227.89

Table 2
Viscosity of MPCMS with 20 μm particle size.

t/℃ wt.%	15℃	20℃	25℃	30℃	35℃
5	3.87	3.2	2.68	2.23	1.87
10	4.81	4.07	3.22	2.74	2.31
15	5.15	4.21	3.58	3.05	2.75
25	10.48	8.26	6.52	5.37	4.51
30	14.38	11.51	8.79	7.15	6.06
35	17.15	13.45	10.63	8.56	7.25

pipeline rises obviously with the experiment. This problem can be effectively solved by replacing liquid with gravity as power.

Before the experiment, the laminar heat transfer experiment was carried out with base liquid as heat transfer medium to verify the accuracy of the experimental platform. In the experiment, the local convective heat transfer coefficient between the inner wall of the pipe and the fluid can be calculated according to equation:

$$h_x = \frac{q_w}{T_w - T_x} \tag{1}$$

In the formula: q_w is the heat flux through the pipe wall, T_w is the temperature of the inner wall of the pipe, T_x is the average temperature of the local fluid in the pipe.

Local Nussel number Nu_x can be calculated by local convective heat transfer coefficient h_x, slurry thermal conductivity at local temperature k_b, and pipeline diameter d:

$$Nu_x = \frac{h_x d}{k_b} \tag{2}$$

The classical formula of Nu_x obtained from the experimental results:

$$Nu_x = 5.364\left[1 + \left(\frac{110x^+}{\pi}\right)^{-\frac{10}{9}}\right]^{\frac{3}{10}} - 1.0 \tag{3}$$

In the formula:

$$x^+ = (x/r_0)/(Re_b Pr_{b0}) \tag{4}$$

Re = 249.5, P = 225.00 W; Re = 401.5, P = 225.00 W; Re = 679.3, P = 225.00 W; and Re = 1440.6, P = 225.00 W were used to carry out laminar heat transfer experiments on base liquids at Re 200.0-1600.0. Nu_x was compared with the calculated values from the classical formula, and the results were shown in Fig. 6. When the Re were 249.5, 401.5, 679.3 1440.6, and the P at 225.00 W. The average errors of experimental and theoretical values of local Nu_x along the pipeline were 6.5%, 8.2%, 8.4% and 7.8%, respectively. It can be concluded that the experimental data measured in the laminar flow range of the experimental platform were reliable.

2.6. Experimental results and discussion

2.6.1. Effect of mass fraction of slurry

The dimensionless wall temperature was defined as:

$$\theta_{w,x} = \frac{T_{w,x} - T_i}{q_w r_0 / k_b} \tag{5}$$

The MPCMS with mass fractions of 2.5%, 5.0%, 10.0% and 15.0% were prepared at a heating power of 46.19 W, a mass flow rate of 24.3 kg/h, a flow rate of 0.537 m/s and a Re = 837.0. The $\theta_{w,x}$ along the pipeline were compared, as shown in Fig. 7.

Fig. 7 showed that the dimensionless wall temperature of the MPCMS decreases as the mass fraction of microcapsule particles increases. The conclusion is the same as P. Charunyakorn et al. (1991a), Goel M (Goel et al., 1994) and Faghri Amir Zhang & Faghri (1995a) concluded that the mass fraction of microcapsule particles was an important factor affecting heat transfer.

2.6.2. Heat transfer of slurry with different particle sizes

Experiments were carried out on microcapsules with flow rate of 0.510 m/s, Re = 737.0, P = 516.00 W and mass fraction of 5.0% and particle size of 5.000 and 20.000 μm, respectively. The results were shown in Fig. 8.

Fig.8 showed that the particle size had little effect on the convective heat transfer coefficient of the pipeline. The average convective heat transfer coefficient of the MPCMS of 5.000 and 20.000 μm microencapsulated particles were 3.8% different along the pipeline at the same velocity and heat flux.

2.6.3. Effect of Stephen number on heat transfer

Stephen number (St) was the ratio of sensible heat to latent heat and was defined as:

$$St_b = \frac{c_{p_b}(T_0 - T_i) - L}{L} \tag{6}$$

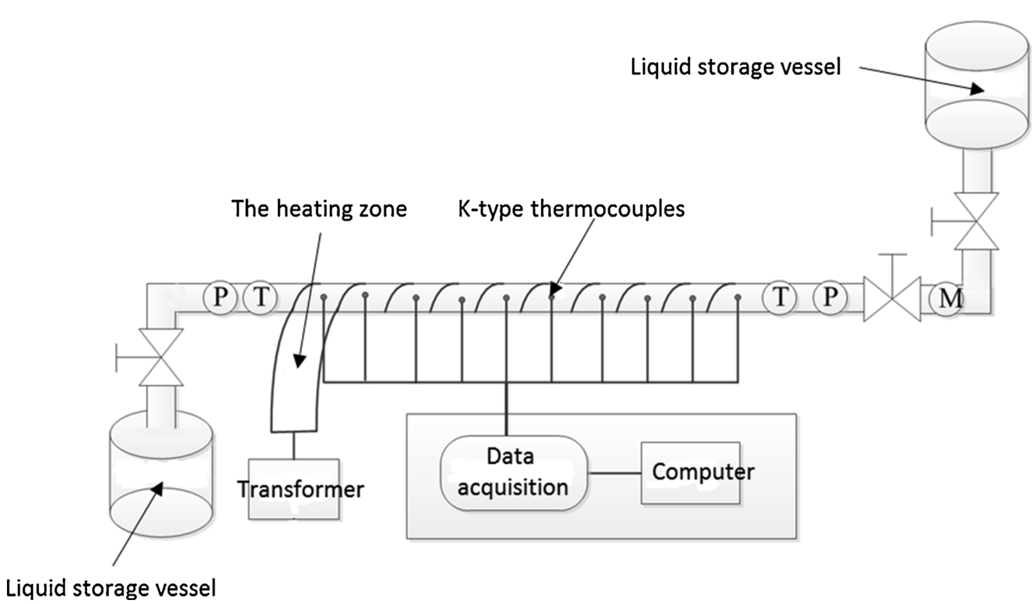


Fig. 5. Layer flow heat transfer system diagram.

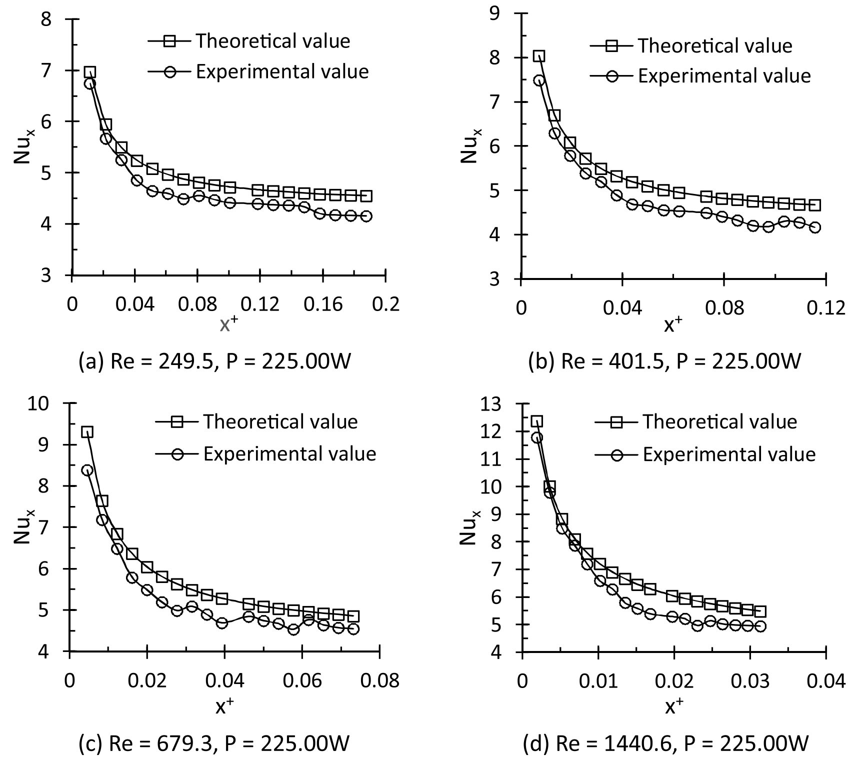


Fig. 6. Verification of the accuracy of the experimental platform.

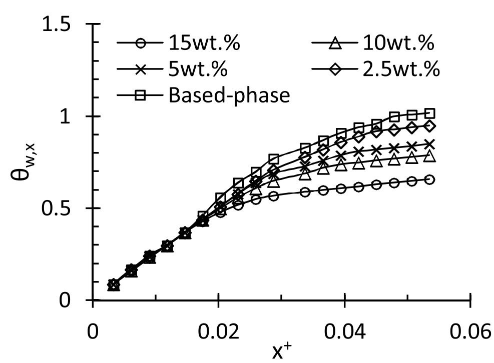


Fig. 7. Effect of mass fraction on heat transfer in pipes.

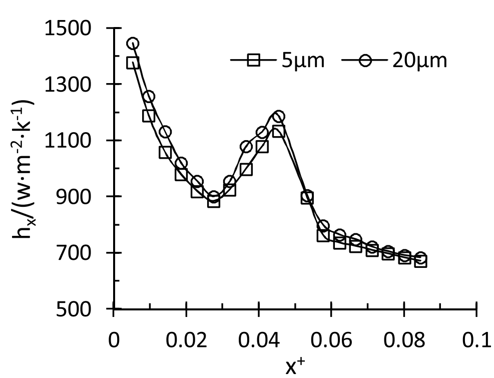


Fig. 8. Effect of particle size on pipe heat transfer.

Table 3
Specific heat capacity calculation of MPCMS.

Name	Specific heat capacity (kJ/(kg. K))
Melamine-Formaldehyde (Wall material)	1.57
Solid Paraffin (Core material)	2.94
MPCM (Wall material: Core material = 15:85)	2.73
Propyl alcohol	2.45
Water	4.2
Based-phase (Propyl alcohol: water = 63:37)	3.55

The specific heat capacity of slurry can be defined as:

$$c_{p_b}\begin{cases} c_{p_{bb}} & T < T_1 \\ c_{p_{bb}} + \dfrac{L}{T_2 - T_1} & T_1 \ll T \ll T_2 \\ c_{p_{b0}} & T > T_2 \end{cases} \tag{7}$$

$c_{p_{b0}}$ it can be calculated from the data in Table 3.

The average convective heat transfer coefficient of the defined pipe was shown in equation:

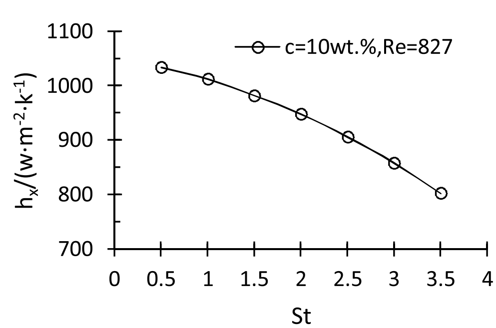


Fig. 9. Effect of Stephen number on heat transfer of pipeline.

$$h_a = \frac{q_w}{T_{w,a} - T_{x,a}} \tag{8}$$

When the mass fraction of the MPCMS was 10.0% and the flow Re was 827.0, there will be different *St* with different heating power.

From Fig.9, it can be seen that the average convective heat transfer coefficient decreases monotonously with the increase of *St*. The result is different from Wang Li (Wang, 2006), thinks that *St* had little effect on convective heat transfer.

2.6.4. Effect of inlet supercooling on heat transfer in pipeline

The different inlet temperature of the pipeline also affects the enhanced heat transfer of the MPCMS. The inlet supercooling was defined as:

$$\Delta\dot{T} = (T_1 - T_i)/(q_w r_o/k_b) \tag{9}$$

In order to investigate the influence of inlet supercooling on heat transfer, the MPCMS with mass fraction of 10.0% and Re of 1426.0 was tested at the inlet temperatures of 22 and 26 °C respectively. Nu_x along the pipeline was obtained under two experimental conditions as shown in Fig. 10.

According to the Fig. 10, the effect of inlet supercooling on the heat transfer of the MPCMS was not obvious. When *St* was 0.2, the average convective heat transfer coefficient of the MPCMS along the pipeline was only 8.7% different from that of the MPCMS $\Delta\dot{T}$ was 0.12 and $\Delta\dot{T}$ was 0.38. The results were influenced by *St*. When *St* changes, the phase change rate of MPCM particles in the whole pipeline was affected by the heating amount, and then the convective heat transfer coefficient of the MPCMS was strongly affected.

2.6.5. Effect of Re on heat transfer

When the mass fraction was 10.0%, the heating power was 418.39 W and Re was 400.0–1300.0, the convective heat transfer coefficient h_x between the pipe wall and the fluid was shown in Fig. 11.

As can be seen from Fig. 11, when the MPCMS with mass fraction of 10.0%, the average convective heat transfer coefficient of Re at 1129.0–1295.0 is 39.1% higher than that at 335.0-484.0. The conclusion is the same as that of Wang Li (Wang, 2006) and contrary to that of P. Charunyakorn (Charunyakorn et al., 1991a). Wang Li (Wang, 2006) thought that Re had a great influence on heat transfer, nevertheless P. Charunyakorn (Charunyakorn et al., 1991a) thought that Re had little effect on heat transfer.

3. Numerical model for convective heat transfer of MPCMS

In order to study the heat transfer of MPCMS in a circular pipe, the temperature value at each point in space and the particle phase change rate at each point in space, the mathematical models of the fluid and particle parts of MPCMS were established and solved numerically by computer.

In order to make the main equation formulaic, the following assumptions were made:

(1) In the study, the maximum volume fraction of microcapsule particles was not more than 25%, so the flow can be regarded as Newtonian fluid (Margon, 1955, Margon & Fok, 1955; Rutgers, 1962). Because the viscosity and thermal conductivity of MPCMS were directly related to its concentration, when the volume fraction of microcapsule particles was more than 25%, it can not be regarded as Newtonian fluid, which was not considered in this paper.

(2) Assuming that the flow in the pipeline was incompressible laminar flow, the melting and cooling of phase change particles were considered to be uniform and fully developed when they enter the heat transfer region. In order to reduce the parameters and simplify the process, it was considered that the MPCMS was not cooled when it entered the heating zone.

(3) The phase change microcapsule particles were spherical in the full sense and were identical with the density of the slurry. The radius of the microcapsule particles was small enough without delamination and precipitation (Fig. 12).

(4) Slurry can be regarded as homogeneous, although the difference in thermal conductivity was a function of local shear force (Ahuja, 1975a, 1975b). This assumption is reliable when the radius ratio of particle to pipeline was small. Radial migration can also be neglected when the radius ratio of particles to pipelines was small. For slurry, small particles were usually easy to coagulate, and their uniformity will increase when dispersants were added.

(5) There may be a wall effect in the actual situation, that is, a free layer of particles will form near the wall, assuming that this phenomenon can be ignored. Because the free layer thickness of particles was about 0.5–1 radius size of particles (Karnis & Goldsmith, 1966; Vand, 1948), this assumption was feasible when the radius ratio of particles to pipes was very small.

3.1. MPCMS fluid control equations

Based on the above assumptions, using the internal heat source model of MPCMS, the phase change effect of microcapsules was equivalent to an internal heat source in the fluid, and the energy equation of the fluid part was:

$$\rho_b c_b u \frac{\partial T_f}{\partial x} = \frac{1}{R}\frac{\partial}{\partial R}\left(k_e R \frac{\partial T_f}{\partial R}\right) + \mathrm{S} \tag{10}$$

In the formula, ρ_b was the density of the slurry, c_b was the specific heat capacity of the slurry and T_f was the temperature of the base liquid were used. The velocity equation of the laminar flow of the slurry in the tube was:

$$u = 2u_{\max}[1 - (R/R_d)^2] \tag{11}$$

In the formula, $u_{\max}$ was the average velocity of the fluid, R_d was the radius of the circular pipe, and R was the axial distance between the point of the required velocity and the center of the pipe.

The boundary condition was:

$$T_f = T_{\text{inlet}} \qquad x = 0, \quad R < R_d \tag{12}$$

$$\frac{\partial T_f}{\partial R} = 0 \qquad R = 0 \tag{13}$$

$$\frac{\partial T_f}{\partial R} = \frac{q_w}{k_e} \qquad R = R_d \tag{14}$$

The determination of internal heat source:

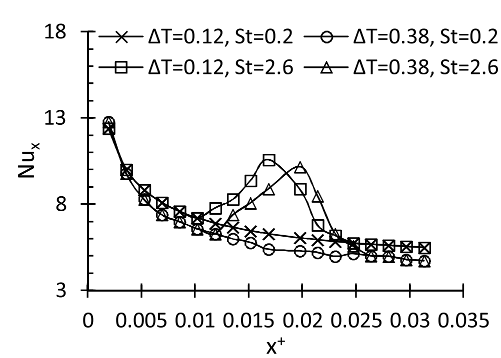


Fig. 10. Effect of inlet supercooling on heat transfer of pipeline.

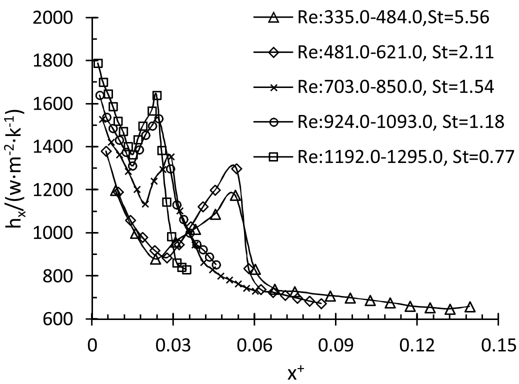


Fig. 11. Influence of Re on heat transfer of pipeline.

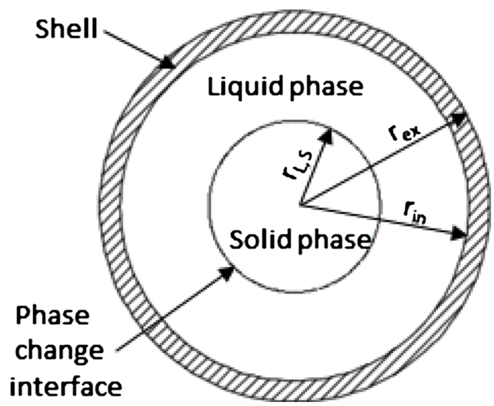


Fig. 12. Schematic diagram of MPCM particles.

Q was defined as the heat transfer capacity of a single microcapsule, and N was the number of microcapsules in a single volume mixture. The calculation formulas were as follows:

$$Q = -4\pi r_{ex}^2 h(\mathrm{T}_p - \mathrm{T}_{\mathrm{f},x_i,R_j}) \tag{15}$$

$$\mathrm{N}= \frac{3c}{4\pi r_{ex}^3} \tag{16}$$

There were:

$$\mathrm{s}= \mathrm{Q}\cdot \mathrm{N}= -\frac{3c}{r_{ex}}h(\mathrm{T}_p - \mathrm{T}_{f,x_i,R_j}) \tag{17}$$

The formula for calculating the effective thermal conductivity k_e can be obtained from the following formula:

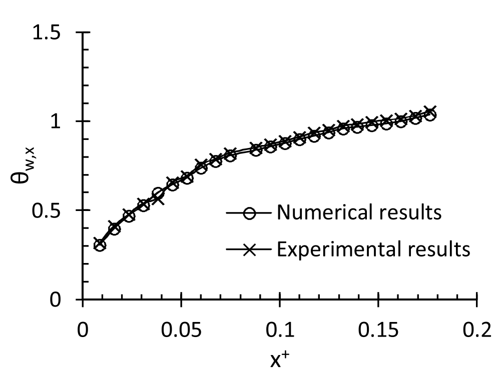


Fig. 13. Verification of accuracy of numerical calculation.

$$\begin{cases} k_{e,i,j} = f\cdot k_b(T_{f,i,j}) \\ k_b(T_{f,i,j}) = k_f \dfrac{2 + \frac{k}{k_f} + 2c\left(\frac{k}{k_f} - 1\right)}{2 + \frac{k}{k_f} - c\left(\frac{k_f}{k_f} - 1\right)} \\ k_f = 0.020833T_{f,i,j} - 0.0275 \\ k = \dfrac{r_{ex}}{\frac{r_{cx}}{k_c} + \frac{r_{ex} - r_c}{k_w}} \\ f = 1 + BcPe^m = 1 + Bc8^m\left[Pe_f\left(\dfrac{r_{ex}}{R_d}\right)^2\right]^m\left(\dfrac{r}{R_d}\right)^m \\ B = 3.0, \quad m = 1.5, \quad Pe < 0.67 \\ B = 1.8, \quad m = 0.18, \; 0.67 \le Pe \le 250 \\ B = 3.0, \quad m = \dfrac{1}{11}, \quad Pe > 250 \\ Pe_f = \dfrac{2R_d u_m}{\alpha_f} \\ Pe = \dfrac{ed_{\mathrm{ex}}^2}{\alpha_f} \\ \alpha_f = \dfrac{k_f}{\rho_f c_f} \\ \mathrm{d}_{ex} = 2r_{ex} \\ \mathrm{e}=\dfrac{du}{dR} = f(u_m, R_{i,j}) \end{cases} \tag{18}$$

3.2. Control equation of particles in MPCMS

MPCM particles consist of core material and wall material. They can be regarded as standard spherical particles. The structure sketch was as follows:

The energy equation of MPCM particles was:

$$r^2\rho c\frac{\partial T}{\partial t} = \frac{\partial}{\partial r}\left(kr^2\frac{\partial T}{\partial r}\right) \tag{19}$$

Thermophysical parameters of core material and wall material were different.

For wall materials:

$$(r_{in} < r \le r_{ex})C_p = C_{p,w} \qquad k_p = k_{p,w} \quad \rho_p = \rho_{p,w} \tag{20}$$

For core materials:

$$(0 \le \mathrm{r}< r_{in})k_p = k_{p,w} \quad \rho_p = \rho_{c,w} \tag{21}$$

$$C_p(T_p)\begin{cases} C_{p,c} & T_p < T_m - \Delta T_p \\ C_{p,c} + \dfrac{L}{\Delta T_P} & T_m - \Delta T_p \le T_p \le T_m \\ C_{p,c} & T_p > T_m \end{cases} \tag{22}$$

Boundary condition:

$$\frac{dr_{L,S}}{dt} = \frac{k_P}{\rho_p L}\frac{\partial T}{\partial r}, \quad r = r_{L,s} \tag{23}$$

$$-\frac{\partial T}{\partial r} = \frac{h_{\mathrm{w}}}{k_{\mathrm{p,w}}}(T_p - T_f), \quad r = r_{\mathrm{ex}} \tag{24}$$

$$\frac{\partial T_P}{\partial r} = 0 \qquad \mathrm{r=0, t} > 0 \tag{25}$$

Initial conditions:

$$T = T_0 \qquad t = 0, \, 0 \le r \le r_{\mathrm{ex}} \tag{26}$$

The numerical results of convective heat transfer of MPCMS containing MPCM particles showed that the average relative error between the dimensionless wall temperature calculated under the condition of equal heat flux and the data measured by K-type thermocouple in this

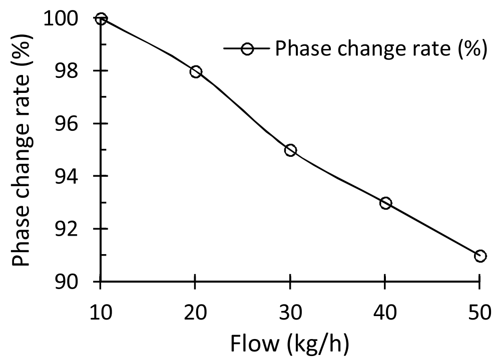


Fig. 14. Phase change rate of slurry at different flow rates.

experiment was less than 5%. As shown in Fig. 13, the model and program can be considered to be reliable.

3.3. Phase change rate of MPCMS in laminar heat transfer

The data showed that the phase change rate of MPCMS from inlet to outlet was related to the flow rate, the mass fraction of MPCM particles and the heat flux density when it was heated in the pipeline. The phase change rate increases with the decrease of flow rate, the decrease of mass fraction of microcapsule particles and the increase of heat flux. Fig. 14 shows the phase change rate of MPCMS with 5% microcapsule mass fraction and $9500\mathrm{J}/(m^2.\ \mathrm{s})$ heat flux was 9% different when the flow rate was 10 kg/h and 50 kg/h.

4. Conclusion

The convective heat transfer characteristics of MPCMS flow in a circular tube were studied experimentally and numerically. The heat transfer enhancement effect of MPCMS, such as mass fraction, St, entrance supercooling, particle size and Re, were analyzed under the condition of equal heat flux. In order to understand the phase change rate of MPCMS in laminar heat transfer from inlet to outlet, considering the influence of wall material of MPCM particles on heat transfer process, the mathematical models of pipeline and MPCM particles were established respectively, and the accuracy of the models was verified. The results by experimental investigation and numerical simulation can be drawn as below:

(1) MPCMS as heat transfer medium can enhance heat transfer.

(2) Mass fraction had a great influence on heat transfer enhancement. Under the conditions of heating power 46.19 W, the rate of flow was 24.3 kg/h, the velocity of flow was 0.537 m/s and Re was 837.0, the average dimensionless wall temperature of MPCMS with mass fraction 2.5% decreased by 5.8% and that of MPCMS with mass fraction 15.0% decreased by 26.5%.

(3) The effect of St on heat transfer enhancement was significant. When the mass fraction of MPCMS was 10.0%, the flow Re was 827.0, and the St was 0.50 and the St was 3.50, the difference of average convective heat transfer coefficient was 22.4%. The smaller the St, the greater the convective heat transfer coefficient.

(4) ΔT had little effect on heat transfer enhancement of MPCMS, and the effect was affected by St.

(5) The particle size of phase change microcapsules had little effect on heat transfer enhancement. When the velocity of flow was 0.51 m/s, Re was 737.0, P was 516.00 W and mass fraction was 5.0%, the difference of average convective heat transfer coefficients of microcapsules with diameter of 5.000 and 20.000 μm were only 3.8%.

(6) When MPCMS was heated in the pipeline, the phase change rate from inlet to outlet was related to the flow rate, the mass fraction of microcapsule particles and the heat flux density. The phase change rate increases with the decrease of flow rate, the decrease of mass fraction of microcapsule particles and the increase of heat flux.

Acknowledgements

The authors would acknowledge our appreciation of financial supports from the EU Marie Curie Actions-International Incoming Fellowships (FP7-PEOPLE-2011-IIF-298093) and Natural Science Foundation of Shanghai, China (17ZR1411300).

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