

Using of Microencapsulated Phase Change Material Suspension to Enhance the Performance of Microchannel Heat Exchanger

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Abstract

The aim of this paper is to improve thermal performance of a counter flow microchannel heat exchanger (CFMCHE) by using microencapsulated phase change material slurry (MEPCM suspension) as a cooling fluid instead of pure fluid. The MEPCM suspension used in this paper consists of microcapsules constructed from n-octadecane as a phase change material (PCM) and the shell material is Polymethylmethacrylat, these capsules are suspended in water in a concentration of (0 – 20) %.

From the results, using of MEPCM suspensions as a cooling fluid leads to modify thermal performance of a CFMCHE by increasing its effectiveness but it also leads to increase the pressure drop. From heat transfer (thermal performance) point of view it is better to use this type of fluid to increase cooling efficiency of a CFMCHE, but due to extra increase in pressure drop it leads to reduce the overall performance compared with pure fluids. Therefore its application depends on the conditions at which this heat exchanger is used.

استخدام معلق المواد متغيرة الطور لتحسين أداء المبادل الحراري المايكروي

المستخلص

الهدف من هذا البحث هو لتحسين الأداء الحراري للمبادل الحراري المتعاكس المايكروي باستخدام معلق مكون من مواد متغيرة الطور كمائع تبريد بدلا من استخدام مائع نقي. المعلق المستخدم في هذا البحث مكون من مائع أساس هو الماء وكبسولات مايكروية مكونة من مادة n-octadecane كمادة متغيرة الطور وقشرة من مادة Polymethylmethacrylat. هذه الكبسولات مخلوطة بالماء بتركيز (0 – 20 %).

من النتائج التي تم الحصول عليها تبين أن استخدام معلق المواد متغيرة الطور كموائع تبريد يؤدي إلى تحسين الأداء الحراري للمبادل الحراري المتعاكس المايكروني من خلال زيادة فعاليته لكنه بنفس الوقت يؤدي إلى زيادة هبوط الضغط. من وجهة نظر الأداء الحراري فإنه يستحسن استخدام هذا النوع من الموائع لزيادة كفاءة التبريد لكن نتيجة للزيادة الكبيرة في هبوط الضغط فإنه يؤدي إلى تقليل الأداء الكلي للمبادل مقارنة مع الموائع النقية. لذلك فإن استخدام هذه الموائع يعتمد على الظروف التي يستخدم فيها هذا المبادل.

1. Introduction

Microchannel heat exchangers are of interest because they can remove large amount of heat over a small volume. This ability makes it well suited for highly specific applications that require compact high heat energy removal solutions such as, biomedical processes, metrology, telecommunications, cooling of high heat flux high density microelectronics, automotive industries, nuclear reactor barriers, fuel processing, aerospace and chemical industries.

Cooling fluids play an important role in all cooling applications, and its thermo physical properties considered as key parameters that affect its cooling abilities. All the observed literature used pure fluids (liquids and gases) as cooling fluids in a CFMCHE.

Using microencapsulated phase change material (MEPCM) suspensions has attracted more and more interest due to their capabilities of enhancing convective heat transfer and thermal storage performance. This heat transfer enhancement results from the latent heat absorption by the PCM in the suspended MEPCM particles during the melting process.

A phase change material PCM is a substance with a high latent heat of fusion which in melting and solidification is capable to absorb and release large amounts of heat. MEPCM offers an opportunity to reduce weight and volume in thermal management systems by utilizing the latent heat of the PCM. In this approach to thermal management, the large latent heat of PCM is coupled with the high heat transfer capabilities of microchannels to achieve a high thermal performance of CFMCHE.

Microchannel heat sink was first proposed by **Tuckerman and Pease (1981) [1]** for electronic cooling. They built a water cooled integral heat sink with microscopic flow channels, and demonstrated that extremely high power density with a heat flux as high as 790 W/cm^2 could be dissipated .

Following the work of Tuckerman and Pease, many researchers have been conducted for microchannels and microchannel heat sinks. **Klein et al (2005) [2]** studied experimentally the effects of surfactants solution on heat transfer of a single phase and boiling flows in

microchannel heat sinks. The surfactant used is Alkyl Poly Glycosides (APG). The results were compared with the heat transfer in water flow under similar conditions. For single phase flow, no significant difference was observed between heat transfer in water and surfactant solutions at various mass concentrations. For boiling flow of surfactant solutions, the optimal value of mass flux was found in which the heat transfer enhancement reached its maximum. The experiment has also revealed that at low mass fluxes, an optimal mass concentration of APG additives may be found for which a two phase flow heat transfer significantly increases. These findings lead to the conclusion that the use of surfactants should be considered as a method for improving two phase boiling flow heat transfer.

Yu et al (2006) [3] experimentally investigated the laminar flow characteristics of suspensions with microencapsulated phase change materials (MEPCM) in water flowing through rectangular cooper minichannels ($D_h < 3 \text{ mm}$) with the aim to develop methods for a more efficient cooling technology for high power electronics. The paper presented the first results of the experimental investigation on the laminar flow frictional characteristics of MEPCM- water suspension with various mass concentrations flowing in rectangular minichannels with ($D_h = 2.71 \text{ mm}$). The mass concentration of the suspension ranged from 0 - 20 %. The experiments were performed to explore the effect of MEPCM mass concentration on the friction factor and pressure drop in the minichannels. The Reynolds number range was from 200 to 2000. From the results, the laminar friction factor of the suspensions increased with increasing the MEPCM concentrations. Compared with the friction factor of water, a slight increase was found for low concentrations of 5 %. However, when the concentration was 10 % or higher, the increase in the friction factor becomes much more distinctive.

Yu et al (2007) [4] investigated experimentally the convective heat transfer characteristics of water based suspensions of (MEPCM) flowing through rectangular cooper minichannels ($D_h = 2.71 \text{ mm}$). MEPCM particles with an average size of $4.97 \mu\text{m}$ were used to form suspensions with mass concentrations ranging from 0 to 20 %. The 5 % suspension always showed a better cooling performance than water resulting in lower wall temperature and enhanced heat transfer coefficients within the range of mass flow rates. The suspensions with higher mass concentrations, however, were more effective only at low mass flow rates. At higher mass flow rates they showed a less effective cooling performance than water.

Farid et al (2007) [5] used the principle of using a suspension of microencapsules in liquid as a heat transfer fluid to improve the performance of microchannel heat sink. The microcapsules contain phase change materials which melt and solidify at a specified range of

temperature. These microcapsules improve the fluid effective specific heat capacity and thermal conductivity due to latent heat effect and micro mixing respectively. The increase in the pumping required to pump the viscous slurry will over shed the benefit in the cooling if the PCM melting range and concentration are not set properly. Hence additional investigation using this technology is required to further develop our knowledge and understanding around this topic .

Lee et al (2007) [6] used a new type of fluids called nanofluid which is a suspension consists of base fluid with some additives of metallic or non metallic solid particles in nano size as a cooling fluid. The nanofluid used in this research was water based nanofluid containing small concentrations of AL_2O_3 . The high thermal conductivity of nano particles is shown to enhance the single phase heat transfer coefficient especially for laminar flow. Higher heat transfer coefficients were achieved mostly in the entrance region of microchannels. However, the enhancement was weaker in the fully developed region, providing that the nano particles have an appreciable effect on the thermal boundary layer development.

Mushtaq et al (2009) [7] used the nanofluids as cooling fluids in a counter flow microchannel heat exchanger. They found that, using of these types of fluids instead of pure fluids lead to increase the thermal performance of this heat exchanger by increasing its effectiveness and the rate of heat transfer without extra increase in the pressure drop across heat exchanger associated with using the nanofluids. Also they found that using of nanofluids lead to increase the overall performance of a counter flow microchannel heat exchanger compared with using pure fluids.

Mushtaq et al (2009) [8] investigated numerically the effect of channels geometry (the size and shape of channels) on the performance of counter flow microchannel heat exchanger and used liquid water as a cooling fluid. They found that with decreasing the size of channels both the effectiveness of heat exchanger and pressure drop were increased and they claimed that the decision of increasing or decreasing the size of channels depends on the application in which this heat exchanger is used. Also they found that the circle is the best shape for the channels of this type of heat exchangers since it gives higher overall performance (including both the hydrodynamic and thermal performance).

It should be pointed out that most of the previous researches done in heat transfer or flow characteristics of MEPCM suspensions focused on flow in macro – channels ($D_h > 5$ mm), mostly in circular tubes [4]. Also these suspensions were not used as cooling fluids in counter flow microchannel heat exchangers up to now as to the knowledge of the authors.

2. Mathematical model

The physical model of the problem shown in Figure. (1) represents the counter flow MCHE which consists of square channels with hot and cold fluids. The difficulties in this model are the flow is developing and heat transfer is conjugated where the 3D energy equation must be solved for the two fluids and solid wall simultaneously. Solving complete heat exchanger numerically needs huge CPU time and is complicated. Therefore, and due to the geometrical and thermal symmetry between channel's rows, individual heat exchange unit which consists of two channels (hot and cold) and separating wall will be considered as shown in Figure.2 to represent the complete CFMCHE and give an adequate indication about its thermal performance [7], [8].

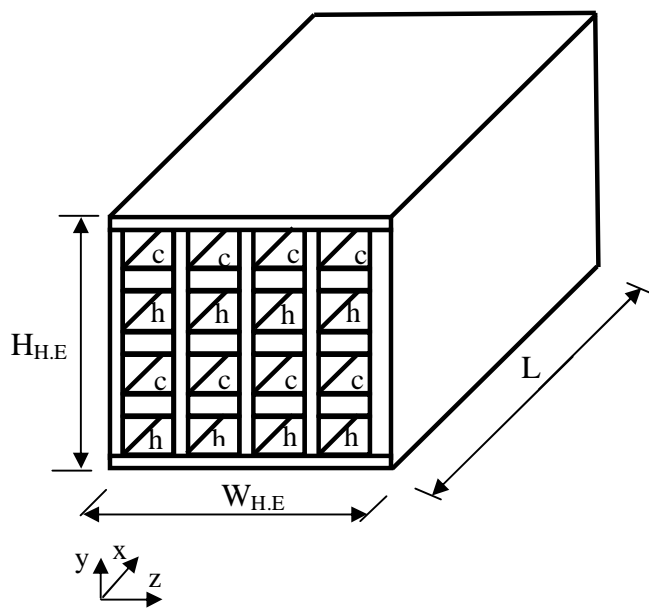
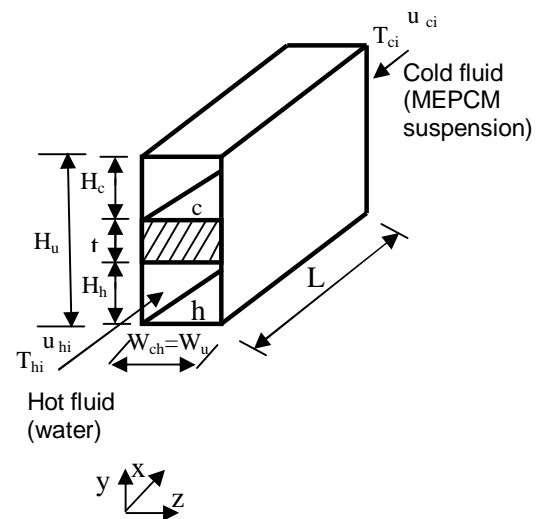


Figure (1). A schematic model of the counter flow MCHE.



Figure(2). A schematic of heat exchange unit .

3. Governing equations

The governing equations used for suspension is the same for pure fluid, the continuity, Navier- stokes and energy equations used with modified fluid properties.

The continuity equation:

$$\frac{\partial u_j}{\partial x} + \frac{\partial v_j}{\partial y} + \frac{\partial w_j}{\partial z} = 0 \quad (1)$$

Momentum equations:

$$u_j \frac{\partial u_j}{\partial x} + v_j \frac{\partial u_j}{\partial y} + w_j \frac{\partial u_j}{\partial z} = -\frac{1}{r_j} \frac{\partial P}{\partial x} + \frac{m_j}{r_j} \left(\frac{\partial^2 u_j}{\partial x^2} + \frac{\partial^2 u_j}{\partial y^2} + \frac{\partial^2 u_j}{\partial z^2} \right) \quad (2)$$

$$u_j \frac{\partial v_j}{\partial x} + v_j \frac{\partial v_j}{\partial y} + w_j \frac{\partial v_j}{\partial z} = -\frac{1}{r_j} \frac{\partial P}{\partial y} + \frac{m_j}{r_j} \left(\frac{\partial^2 v_j}{\partial x^2} + \frac{\partial^2 v_j}{\partial y^2} + \frac{\partial^2 v_j}{\partial z^2} \right) \quad (3)$$

$$u_j \frac{\partial w_j}{\partial x} + v_j \frac{\partial w_j}{\partial y} + w_j \frac{\partial w_j}{\partial z} = -\frac{1}{r_j} \frac{\partial P}{\partial z} + \frac{m_j}{r_j} \left(\frac{\partial^2 w_j}{\partial x^2} + \frac{\partial^2 w_j}{\partial y^2} + \frac{\partial^2 w_j}{\partial z^2} \right) \quad (4)$$

Where $j=h$ and c for hot and cold fluids respectively.

The energy equation based on enthalpy is [5].

$$\nabla \cdot [\vec{V} (r_f H_e)] = \nabla \cdot (k_f \nabla T_f) \quad (5)$$

Or

$$r_f \left[u \frac{\partial H_e}{\partial x} + v \frac{\partial H_e}{\partial y} + w \frac{\partial H_e}{\partial z} \right] = k_f \left[\frac{\partial^2 T_f}{\partial x^2} + \frac{\partial^2 T_f}{\partial y^2} + \frac{\partial^2 T_f}{\partial z^2} \right] \quad (6)$$

The enthalpy of the suspension (H_e) is described by equation (7) and computed as the sum of the sensible heat (h_e) and the latent heat (ΔH) of the PCM.

$$H_e = h_e + \Delta H \quad (7)$$

The sensible heat is described by equation (8), where h_{ref} is the reference enthalpy at T_{ref} [9].

$$h_e = h_{ref} + \int_{T_{ref}}^T C_{p_f} dT \quad (8)$$

The latent heat of the slurry (ΔH) is described by equation (9) as a function of the latent heat of the PCM (L), MEPCM mass fraction (f) and the melted mass fraction (β).

The melted mass fraction (β) is defined as the mass ratio of melted PCM to the total mass of PCM in the slurry. The PCM starts to melt at $T_{solidus}$ and completely melts at $T_{liquidus}$ where the liquid fraction can vary from zero at $T_{solidus}$ to one at $T_{liquidus}$. Equation (10), which is known as the lever rule, describes the melted mass fraction (β).

$$\Delta H = \beta f L \quad (9)$$

Where:

$$\begin{aligned} \beta &= 0 & \text{if} & \quad T_f < T_{solidus} \\ \beta &= 1 & \text{if} & \quad T_f > T_{liquidus} \\ \beta &= \frac{T_f - T_{solidus}}{T_{liquidus} - T_{solidus}} & \text{if} & \quad T_{solidus} < T_f < T_{liquidus} \end{aligned} \quad (10)$$

The energy equation for the heat exchanger solid walls:

$$k_s \nabla^2 T_s = 0 \quad (11)$$

4. Model boundary conditions

The MEPCM – water slurry, which contains micro size particles, enters the channels at a specified temperature and velocity and its temperature increases as it moves through the channels and reaches the melting temperature of PCM. When the PCM melts inside the capsules, the melted PCM remains contained in the capsules and will not mix with the carrier fluid. In other words, there is no mass transfer between the capsules and the carrier fluid. The carrier fluid exhibits lower temperature change when the PCM melts.

The boundary conditions used are:

For lower channels (Hot fluid) (water) ($0 \leq y \leq H_h$)

Location	B.C.	Comments
At $x = 0$	$u_h = u_{hi} , v_h = w_h = 0 , T_h = T_{hi}$	Hot fluid inflow
At $x = L$	$\frac{\partial u_h}{\partial x} = v_h = w_h = 0 , \frac{\partial T_h}{\partial x} = 0$	Hot fluid outflow (fully developed flow, end of channel)
At $z = 0$	$u_h = v_h = w_h = 0 , \frac{\partial T_h}{\partial z} = 0$	No-slip, adiabatic wall
At $z = W_{ch}$	$u_h = v_h = w_h = 0 , \frac{\partial T_h}{\partial z} = 0$	No-slip, adiabatic wall
At $y = 0$	$u_h = v_h = w_h = 0 , \frac{\partial T_h}{\partial y} = 0$	No-slip, adiabatic wall
At $y = H_h$	$u_h=v_h=w_h=0 , -k_h \frac{\partial T_h}{\partial y} = -k_s \frac{\partial T_s}{\partial y} , T_h = T_s$	Fluid -solid interface (no-slip, conjugate heat transfer)

For upper channel (cold fluid) (PCM suspension) ($H_{h+t} \leq y \leq H_{h+t}+H_c$)

Location	B.C.	Comments
At $x = 0$	$\frac{\partial u_c}{\partial x} = v_c = w_c = 0 , \frac{\partial T_c}{\partial x} = 0$	Cold fluid outflow (fully developed flow, end of channel)
At $x = L$	$u_c = u_{ci} , v_c = w_c = 0 , T_c = T_{ci}$	Cold fluid inflow
At $z = 0$	$u_c = v_c = w_c = 0 , \frac{\partial T_c}{\partial z} = 0$	No-slip, adiabatic wall
At $z = W_{ch}$	$u_c = v_c = w_c = 0 , \frac{\partial T_c}{\partial z} = 0$	No-slip, adiabatic wall
At $y = H_{h+t}$	$u_c = v_c = w_c = 0 ,$	Fluid-solid interface

	$-k_c \frac{\partial T_c}{\partial y} = -k_s \frac{\partial T_s}{\partial y}, T_c = T_s$	(no-slip, conjugate heat transfer)
At $y = H_h + t + H_c$	$u_c = v_c = w_c = 0, \frac{\partial T_c}{\partial y} = 0$	no-slip, adiabatic wall

For solid wall separating two channels ($H_h \leq y \leq H_h + t$)

Location	B.C.	Comments
At $x = 0$	$\frac{\partial T_s}{\partial x} = 0$	Adiabatic wall
At $x = L$	$\frac{\partial T_s}{\partial x} = 0$	Adiabatic wall
At $z = 0$	$\frac{\partial T_s}{\partial z} = 0$	Adiabatic wall
At $z = W_{ch}$	$\frac{\partial T_s}{\partial z} = 0$	Adiabatic wall
At $y = H_h$	$-k_h \frac{\partial T_h}{\partial y} = -k_s \frac{\partial T_s}{\partial y}, T_h = T_s$	Fluid - solid interface
At $y = H_h + t$	$-k_c \frac{\partial T_c}{\partial y} = -k_s \frac{\partial T_s}{\partial y}, T_c = T_s$	Fluid - solid interface

The above equations with their boundary conditions were solved by using CFD software FLUENT 6.3 and the distributions of velocity, pressure and temperature were calculated, from which the pressure drop and effectiveness are calculated.

Heat exchanger effectiveness is the ratio of the actual heat transfer to the maximum possible heat that can be transferred:

$$e = \frac{q}{q_{\max}} \quad (12)$$

Where

$$q_{\max} = C_{\min} (T_{hi} - T_{ci})$$

$$\text{And } q = C_h (T_{hi} - T_{ho}) = C_c (T_{co} - T_{ci})$$

$$\text{Where } C_h = \dot{m} C_{p_h} \quad \text{and} \quad C_c = \dot{m} C_{p_c}$$

Total pressure drop across the heat exchange unit is:

$$\Delta P_t = \Delta P_h + \Delta P_c = (P_{hi} - P_{ho}) + (P_{ci} - P_{co}) \quad (13)$$

To calculate the overall performance of a heat exchanger taking into consideration both the thermal and hydrodynamic performance a parameter called performance index used which is the ratio of effectiveness to the total pressure drop [8]:

$$h = \frac{e}{\Delta P_t} \quad (14)$$

5. Properties of microcapsules

In general, MEPCM particles are composed of polymers as the wall material surrounds a core of PCM. The polymer wall is sufficiently flexible to accommodate volume changes that accompanied by solid / liquid phase change.

As shown in Figure (3) a single MEPCM particle consists of two parts: the outer polymer shell and the inner phase change material. The investigated MEPCM particles have an average diameter of 5 μm . The core material is n – octadecane which has a melting temperature of about 28 $^{\circ}\text{C}$, and the shell material is polymethylmethacrylat (PMMA) [4], [10], and [11].

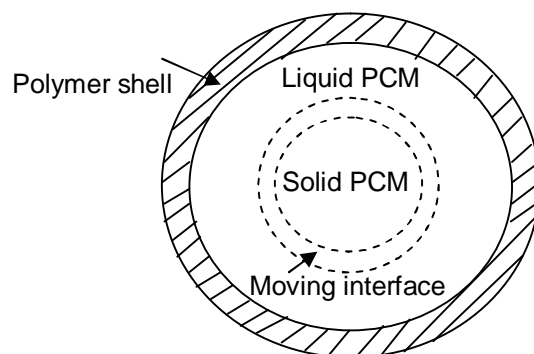


Figure (3). Sketch of a single MEPCM particle.

The core material (PCM) in a single MEPCM particle is about 70 % by volume. The selection of the suspending fluid was governed by a more important factor, its compatibility with n – octadecane and the microcapsule wall. Water was chosen as the suspending fluid because it is easy to handle and has no effect on the PCM or the microcapsule wall.

Since the thermo physical properties of the n – octadecane and the wall material are different, the properties of the microencapsules must be calculated by considering the properties of the individual components. The density and specific heat of the microencapsules were calculated by using mass and energy balance respectively, where the density of n – octadecane was taken as the mean of its solid and liquid densities [12].

$$r_{PCM} = \frac{10}{7} \left(\frac{d_c}{d_{PCM}} \right)^3 r_c \quad (15)$$

$$Cp_{PCM} = \frac{(7 Cp_c + 3 Cp_{wall}) r_c r_{wall}}{(3 r_c + 7 r_{wall}) r_{PCM}} \quad (16)$$

The thermal conductivity of the microcapsules was calculated by using the composite sphere approach. The thickness of microcapsule wall determines the heat transfer resistance of the wall material, while the heat transfer resistance of the core material was evaluated considering the model for a solid sphere in an infinite medium. Thus the thermal conductivity of the microencapsules given by:

$$\frac{1}{k_{PCM} d_{PCM}} = \frac{1}{k_c d_c} + \frac{d_{PCM} - d_c}{k_{wall} d_{PCM} d_c} \quad (17)$$

Where:

PCM : Particle (capsule = core + wall).

c : core material (phase change material).

Wall : polymer wall of the capsule.

d : diameter.

6. Properties of suspension

The bulk properties of suspension are a combination of the properties of the suspending fluid (water) and the microcapsules. Using a mass and energy balance, the density and specific heat are calculated [4], [12].

$$r_f = c r_{PCM} + (1 - c) r_w \quad (18)$$

$$Cp_f = f Cp_{PCM} + (1 - f) Cp_w \quad (19)$$

To calculate the viscosity of the suspension, the following relation was used:

$$m_f = m_w (1 - c - 1.16 c^2)^{-2.5} \quad (20)$$

This relation is valid for concentration up to 20 % and channel – to – particle diameter ratio of 20 – 100 and mean particle diameters of 0.3 – 400 μm [12].

The bulk thermal conductivity of suspension was calculated from the relation:

$$k_f = \frac{2k_w + k_{PCM} + 2c(k_{PCM} - k_w)}{2 + \frac{k_{PCM}}{k_w} - c\left(\frac{k_{PCM}}{k_w} - 1\right)} \quad (21)$$

$$f = \frac{c r_{PCM}}{(r_w + c(r_{PCM} - r_w))} \quad (22)$$

Table (1) gives the values of properties of the microcapsules and the MEPCM – water suspension calculated using the above equations.

Table(1). Physical properties of suspension components and suspension.

fluid	Density (Kg / m ³)	Cp (J / Kg.K)	k (W / m. K)	μ (Kg / m.s)
water	981.3	4189	0.643	0.000598
n – octadecane (MEPCM core)	solid=850 liquid=780	2000	0.18	-
PMMA (MEPCM wall)	1190	1470	0.21	-
MEPCM particles	867.2	1899	0.1643	-
2 % suspension	978	4148	0.6303	0.000629
5 % suspension	975.59	4087.3	0.611	0.000685
10 % suspension	969.89	3984.2	0.581	0.000803
15 % suspension	964.18	3882.1	0.5519	0.00097
20 % suspension	958.48	3774.7	0.523	0.00121

MEPCM volume fractions used in this paper were (2 %, 5 %, 10 %, 15 % and 20 %). Rao [3] used c up to 20 % and mentioned that when c becomes greater than 15 % the suspension behaves as a non Newtonian fluid. While Bernard [13] mentioned that the suspension up to $c = 30$ % becomes a non Newtonian fluid. Farid [5] mentioned that the suspension flow is considered to be Newtonian as long as the MEPCM particles are less than

25 %. Therefore the volume fraction used in this paper is up to 20 % to ensure that the suspension flow is a Newtonian flow.

7. Numerical model

A finite volume method (FVM) is used to convert governing equations to algebraic equations accomplished using an "upwind" scheme. The SIMPLE algorithm is used to enforce mass conservation, and to obtain the pressure field. The segregated solver is used to solve the governing integral equations for the conservation of mass, momentum and energy. CFD package FLUENT 6.3 is used to calculate the distribution of velocity, pressure and temperature in a CFMCH. A mesh was generated by discretizing the computational domain (two channels and separating wall). Mesh independent was studied by using four mesh sizes of hexahedral element and the results for effectiveness and central velocity in fully developed region for different meshes used are listed in table 2 for $Re=50$.

Table (2) shows that the solution becomes independent of grid size and from third configuration further increase in the grids will not have a significant effect on the solution and results of such arrangement are acceptable. Therefore and for more accuracy the grid size of $(25 \times 25 \times 110)$ and $(25 \times 15 \times 110)$ is used.

Table (2). Mesh independent study.

mesh size	$V_{f,d}$ (m /s)	Effectiveness %
(16 x 16 x 90) and (16 x 8 x 90) in z, y, x directions for channels and separating wall respectively	0.6401	42.82
(20 x 20 x 100) and (20 x 10 x 100) in z, y, x directions for channels and wall respectively	0.6481	42.41
(20 x 20 x 110) and (20 x 10 x 110) in z, y, x directions for channels and wall respectively	0.6493	42.26
(25 x 25 x 110) and (25 x 15 x 110) in z, y, x directions for channels and wall respectively	0.6495	42.21

The convergence criteria to control the solution for momentum and energy equations were set to be less than 10^{-6} .

8. Results and discussion

Simulations were conducted first with pure water, and then repeated with MEPCM – water slurry with volume percentages of 2 %, 5 %, 10 %, 15 %, and 20 %. Noting that the T_{solidus} for n- octadecane is $24\text{ }^{\circ}\text{C}$ (297 K) and its $T_{\text{liquidus}} = 29\text{ }^{\circ}\text{C}$ (302 K) also its latent heat $L = 245000\text{ (J /Kg)}$.

To check the accuracy of present numerical model, verification was made by comparing the numerical results of present model with the numerical results of reference [5]. The numerical model presented in [5] is a microchannel heat sink has a width of 5.1 mm, height of 1.5 mm and length of 10 mm. It consists of 25 equally spaced rectangular microchannels each one with 100 μm width, 500 μm height, 10 mm length and 166.6 μm hydraulic diameter. Channels are separated by a 100 μm wall thickness of Aluminum. Due to computational difficulties of modeling the whole heat sink with 25 channels and due to symmetry in geometry of channels, the numerical model used in [5] includes only a half of individual channel with its surrounding of the heat sink metal.

Thermal boundary condition is a constant heat flux of 100 W/ cm^2 acting at the bottom wall of heat sink. The inlet velocity is 1 m/s and inlet temperature of 300 K, the PCM used with melting range of 300 - 305 K.

Figure (4) represents a distribution of bulk temperature of MEPCM-water suspension along microchannel for results of present model and numerical results of [5]. From this figure it can be seen that, the agreement between results of present model and results of [5] is accepted since the mean error for all points is 2.1 %. From these results it can be concluded that, the present model can be safely used to simulate a CFMCHE with MEPCM-water slurry as a cooling fluid.

Figure (5) shows the distribution of bulk temperature of suspension along heat exchanger for 0 % (pure water), 5 %, 10 % and 15 % of MEPCM volume concentrations at $V_i = 0.25\text{ m/s}$. From this figure one can see that the bulk temperature of suspension decreased with the increase of the volume fraction due to increase in the latent heat of melting results from adding more amount of PCM. Since the amount of heat absorbed from hot fluid in case of using MEPCM - suspension consists of sensible and latent heat. Therefore increasing volume

fraction of PCM in the suspension leads to increase the latent heat part and as a result the bulk temperature decreased.

Variation of effectiveness of heat exchanger with volume fraction for MEPCM-water suspension at $V_i = 0.25$ m/s is shown in Figure (6). From this figure one can see that, the effectiveness of the heat exchanger increased with the increase of the volume fraction of MEPCM in the suspension compared with its value for pure water. Despite the decrease in thermal conductivity of suspension with increased volume fraction, the effectiveness increases due to releasing the latent heat of melting of PCM in the suspension, which is responsible for absorbing extra heat from the hot fluid. Since the heat is transferred as a sensible and latent heat in such applications.

The percentage of modification in the effectiveness of a CFMCHE results from using MEPCM-water suspension as a cooling fluid compared with pure water is 15.83 % for volume fraction 20 % and inlet velocity 0.25 m/s.

The effectiveness of a CFMCHE can be increased more than this value by increasing the volume fraction in the suspension more than 20 % but there is a limitation on the increase of volume fraction, since increasing volume fraction more than certain level leads to change the fluid into non-Newtonian fluid therefore the volume fraction used in this paper is up to 20 % [3], [5] and [12].

Figure(7) shows the variation of pressure drop with volume fraction of MEPCM-water suspension at $V_i = 0.25$ m/s. From this figure the pressure drop increases with increasing volume fraction due to the increase in the viscosity. It's clear from this figure that, there is a large increase in pressure drop associated with using this type of fluids for cooling in a CFMCHE, since the percentage increase in pressure drop for MEPCM suspension compared with that for pure water is 50.49 % for 20 % volume fraction and 0.25 m/s inlet velocity. The increase in pressure drop results from using MEPCM-water suspension considered as one of the main disadvantages of using such fluids in this application.

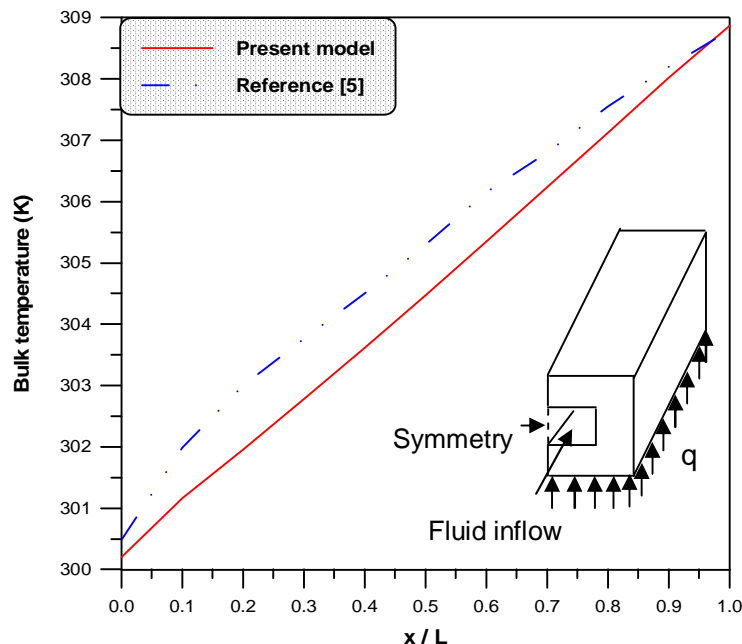
As can be seen from Figures (6) and (7) with using of MEPCM-water slurry as a cooling fluid in a CFMCHE, both effectiveness and pressure drop increased, therefore it is important to find out its effect on the overall performance of this heat exchanger.

The overall performance of a CFMCHE can be represented by a performance index which gives an indication about the overall performance.

Variation of performance index with volume fraction for MEPCM-water suspension at inlet velocity 0.25 m/s is illustrated in Figure (8). From this figure it can be seen that the

performance index for MEPCM suspension decreased with increasing volume fraction compared with that for pure water. Due to that, the increase in pressure drop is higher than the increased effectiveness by adding more amount of PCM. The value of performance index for all ranges of volume fraction used for MEPCM suspension is lower than that for pure water due to extra increase in pressure drop in suspension.

Figure (9) indicates the variation of effectiveness for 0 % (pure water), 5 %, 10 % and 15 % MEPCM volume concentrations with inlet velocity. From this figure the effectiveness decreased with the increase the velocity of flow, also the modification occurred in effectiveness due to adding of more PCM decreased with the increase in flow velocity because some of PCM in slurry did not have enough time to complete its melting inside channels which lead to decrease the latent heat of melting and reduce the cooling performance. This can be proven from Figure (10) which represents the variation of bulk temperature of suspension with inlet velocity for $c = 5\%$. From this figure it can be seen that, the bulk temperature decreases with the increase in the velocity and it reaches (300.9) K at $v_i = (3)$ m/s i.e. the temperature of suspension decreased lower than T_{solidus} . This means that, melting cannot be completed at this velocity.



Figure(4). Distribution of bulk temperature along channel as a comparison between present model and results of reference [5].

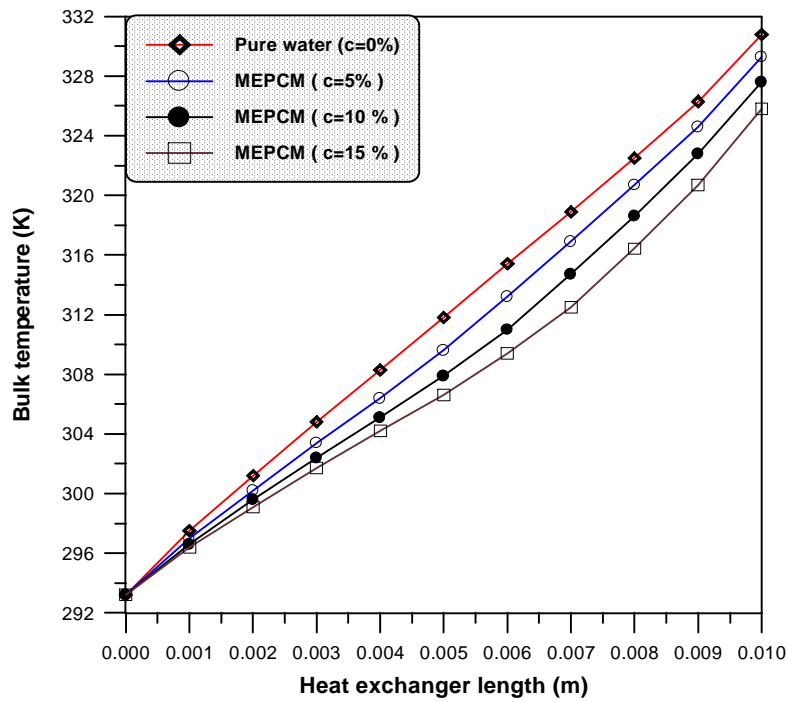
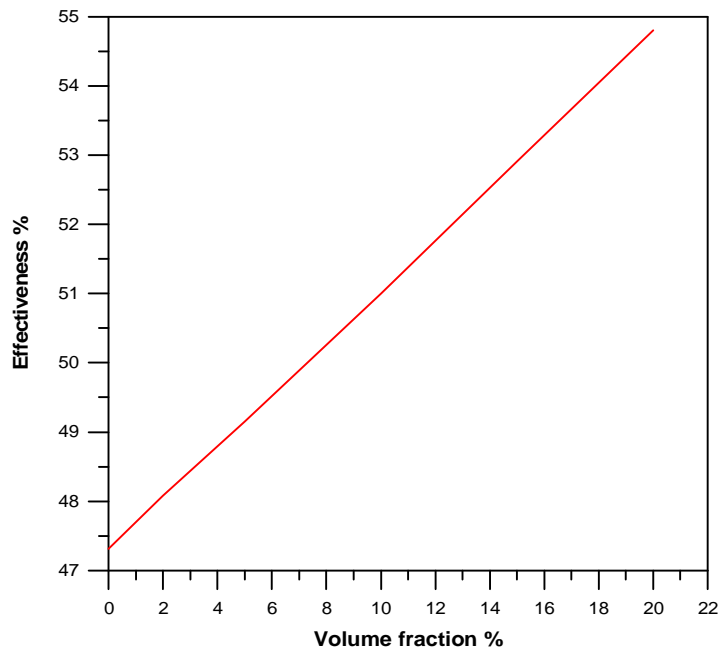
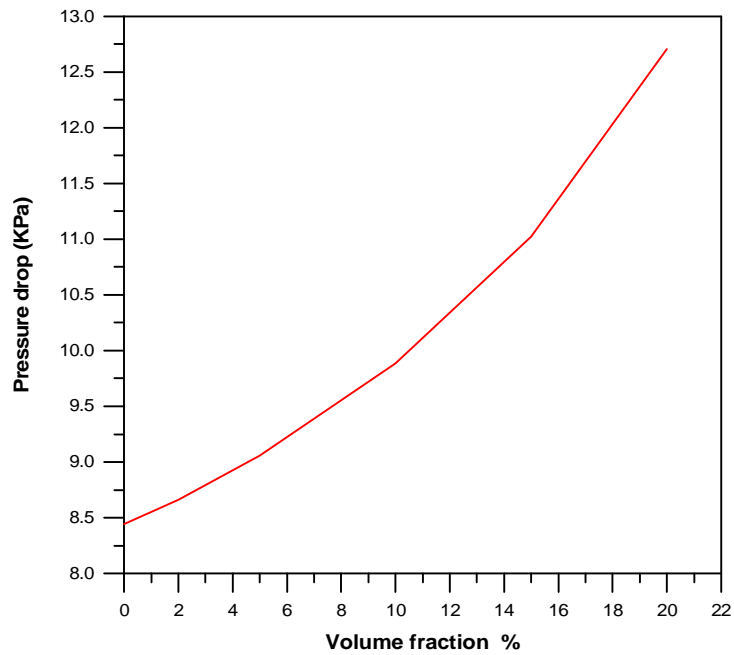


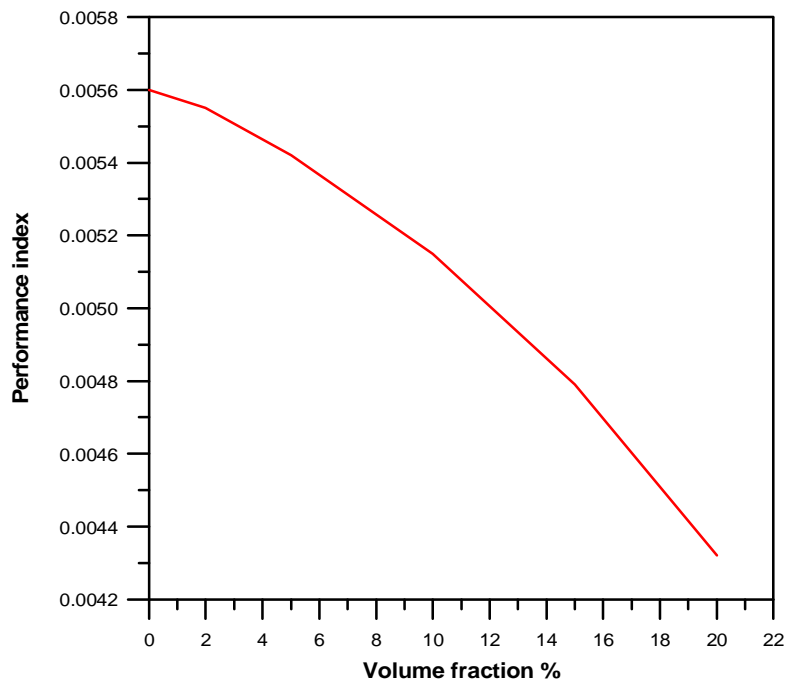
Figure (5). Distribution of bulk temperature of MEPCM suspension for different values of volume fractions ($V_i=0.25$ m/s).



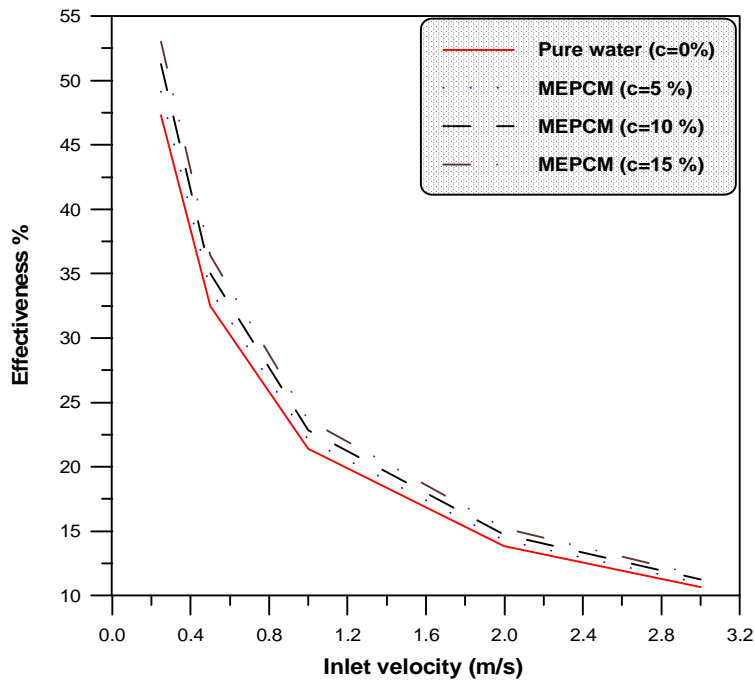
Figure(6). Variation of effectiveness with MEPCM volume fractions at $V_i = 0.25$ m/s.



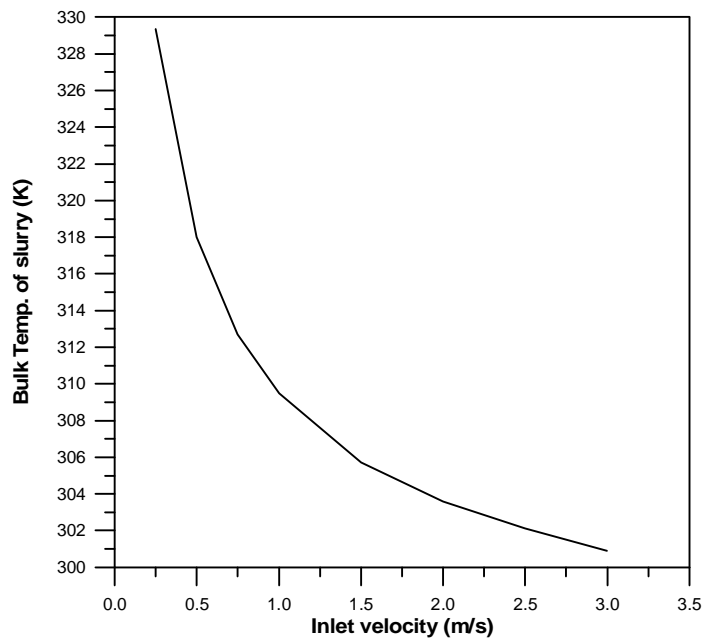
Figure(7). Variation of pressure drop with MEPCM volume fraction at $V_i = 0.25$ m/s.



Figure(8). Variation of performance index with MEPCM volume fraction at $V_i = 0.25$ m/s.



Figure(9). Variation of effectiveness with inlet velocity for different values of volume fraction.



Figure(10). Variation of suspension bulk temperature with inlet velocity for c = 5 %.

9. Conclusions

In this paper the MEPCM-water suspension was used as a cooling fluid instead of pure water in CFMCHE to modify its thermal performance. From the results the following conclusions can be drawn:

- 1- Using this type of suspension leads to modify the thermal performance of a CFMCHE by increasing its effectiveness.
- 2- The use of MEPCM-water suspension leads to increase the pressure drop across the CFMCHE to large values. And as a result its leads to decrease the overall performance of this heat exchanger.
- 3- From thermal performance point of view, the use of MEPCM suspension is desirable since it leads to increase the effectiveness. And from the hydrodynamic performance point of view using this type of suspension leads to extra increase in pressure drop and decrease the performance index. Therefore, using this type of suspension depends on the application in which a CFMCHE is used. For land base applications such as medical laboratories it is preferable to use the suspension to increase thermal performance. While in space applications the extra increase in pressure drop overcome the benefits of heat transfer enhancement.
- 4- It is better to use the MEPCM – suspension at low velocity to get the benefits of melting of PCM and releasing the latent heat.

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11. Nomenclature

A	Cross-sectional area (m ²)
c	Volume fraction
C _p	Specific heat capacity (J / kg K)
D _h	Hydraulic diameter (m)
H	Channel height (m)
h	Convection heat transfer coefficient (W/m ² K)
He	Enthalpy of suspension (W)
he	Sensible heat (W)
k	Thermal conductivity (W/m K)
L	Heat exchanger length (m)
P	Total pressure (Pa)
q	Heat transfer rate (W)
t	Separating wall thickness (m)
T	Temperature (K)
u	Fluid x-component velocity (m/s)

v	Fluid y-component velocity (m/s)
W_{ch}	Channel width (m)
w	Fluid z- component velocity (m/s)
x	Axial coordinate
y	Vertical coordinate
z	Horizontal coordinate
ΔH	Latent heat (W)
ΔP	Pressure drop (Pa)
\dot{m}	Mass flow rate (Kg/s)

Greek letters

ρ	Density (Kg/m ³)
μ	Dynamic Viscosity (m ² /s)
ε	Heat exchanger effectiveness
f	Mass fraction
β	Melted fraction
η	Performance index (1/Pa)

Subscripts

c	Cold
ch	Channel
f	Suspension
h	Hot
i	Inlet
Max.	Maximum
o	Outlet
p	Particle
ref.	Reference
s	Solid
t	total
u	Heat exchange unit
w	Water