OPTIMAL DESIGN VARIABLE CONSIDERATIONS IN THE USE OF PHASE CHANGE MATERIALS IN INDIRECT EVAPORATIVE COOLING

by

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Abstract

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The demand for sustainable, energy efficient and cost effective heating and cooling solutions is exponentially increasing with the rapid advancement of computation and information technology. Use of latent heat storage materials also known as phase change materials (PCMs) for load leveling is an innovative solution to the data center cooling demands. These materials are commercially available in the form of microcapsules dispersed in water, referred to as the microencapsulated phase change slurries and have higher heat capacity than water. The composition and physical properties of phase change slurries play significant role in energy efficiency of the cooling systems designed implementing these PCM slurries. Objective of this project is to study the effect of PCM particle size, shape and volumetric concentration on overall heat transfer potential of the cooling systems designed with PCM slurries as the heat transfer fluid (HTF). In this study uniform volume heat source model is developed for the simulation of heat transfer potential using phase change materials in the form of bulk temperature difference in a fully developed flow through a circular duct. Results indicate the heat transfer potential increases with PCM volumetric concentration with gradually diminishing returns. Also, spherical PCM particles offer greater heat transfer potential when compared to cylindrical particles. Results of this project will aid in efficient design of cooling systems based on PCM slurries.

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Chapter 1

Introduction

The present Data Centers (DC) are state of the art piece of infrastructure which is growing exponentially as need for cloud computing is growing, requiring faster setup and better processing. These data centers are macro modules consisting of many more miniature systems, that are designed with more efficient energy usage in mind, including considerations regarding the external environment. By simply scaling the data centers may solve this crisis but it gives rise another issue with the data centers which is the cost factor. As the size of the data centers so does the energy demand. So several companies such as Hewlett Packard, Google, Yahoo, Amazon, CommScope, etc. are focusing on maximizing the energy effeciency of the data centers by controling the power utilization [1].

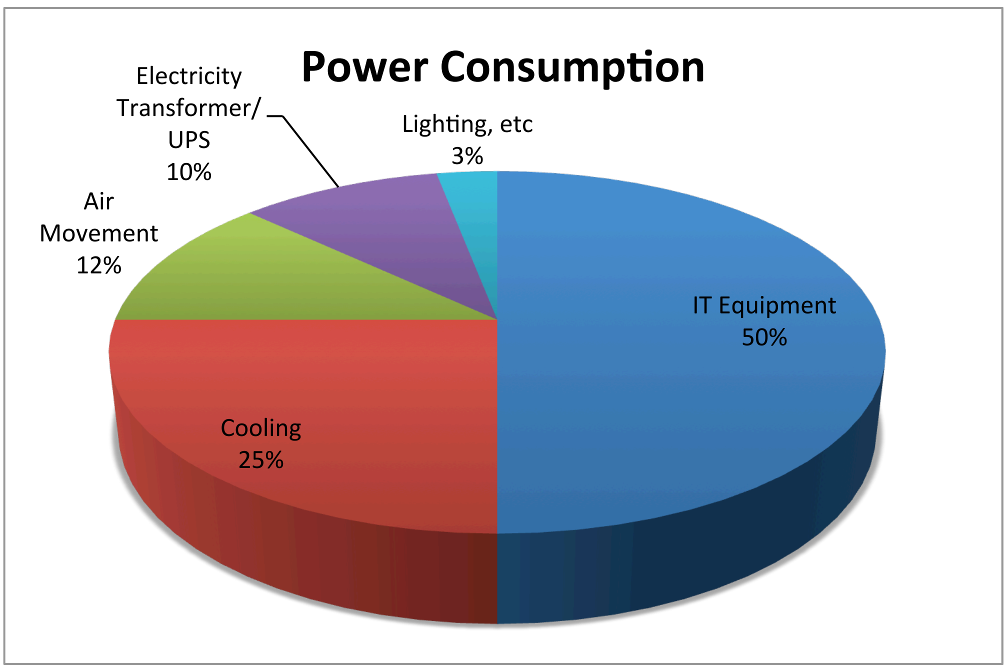

Figure 1-1 Data center power consumption [16]

1.1 Background

Data centers are 40 times more energy intensive than a standard office building and require higher levels electrical consumption [6]. Of this half of the energy is wasted due to lack of technology to make them more efficient. In 2006 a typical commercial 19 inch racks heat load was about 27kW. In 2013, United States data centers consumed an estimated 91 billion kilowatt-hours of electricity which is enough electricity to power all the households in New York City twice over. At this rate it is projected that by 2020, the data center electricity consumption will increase to roughly 140 billion kilowatt-hours annually by 2020. If just half of the wastage or losses can be recovered, electrical consumption could be cut by as much as 40 percent which would reflect a savings of 39 billion kilowatt-hours annually [16],

This has led for the search of innovative hybrid methods for providing sustainable, energy efficient and cost effective heating and cooling solutions through thermal energy storage (TES) by the process of load leveling. Energy storage will help in balancing the losses by energy conservation. Because of the limitations due to size, space power, the choice of cooling systems to embed the TES systems are limited, which are chilled water system, evaporative cooling individually or in combination with free-air cooling system. Of this Evaporative Cooling Unit, which is divided into Indirect/Direct Evaporative Cooling Units, is the most energy efficient option that creates an optimal, sustainable and secure environment for server performance while lowering the data center's Power Usage Effectiveness (PUE). These units are self-contained and the roof-mounted units cool and filter the air delivered to the data center, allowing to easily achieving ASHRAE guidelines for operating conditions [5]. The thermal energy storage process would invole absorbing of the latent heat from the source which in this case would be hot air coming from the rack servers and use of that stored latent heat for other purposes, which in our case be to heat the data center building during colder temperatures. The latent heat storage materials also known as phase change materials (PCM's) are commercially available in the form of microcapsules dispersed in water also known as microencapsulated phase change slurry and have higher heat capacity than water. The composition of the phase change slurry plays a significant role as to how much it is efficient.

1.2 Objective

Objective of this project is to develop a uniform volume heat source model for the simulation and hence study the effect of phase change material particle size, shape and volumetric concentration on overall heat transfer potential of the cooling systems designed with PCM slurries as the heat transfer fluid (HTF).

1.3 Methodology

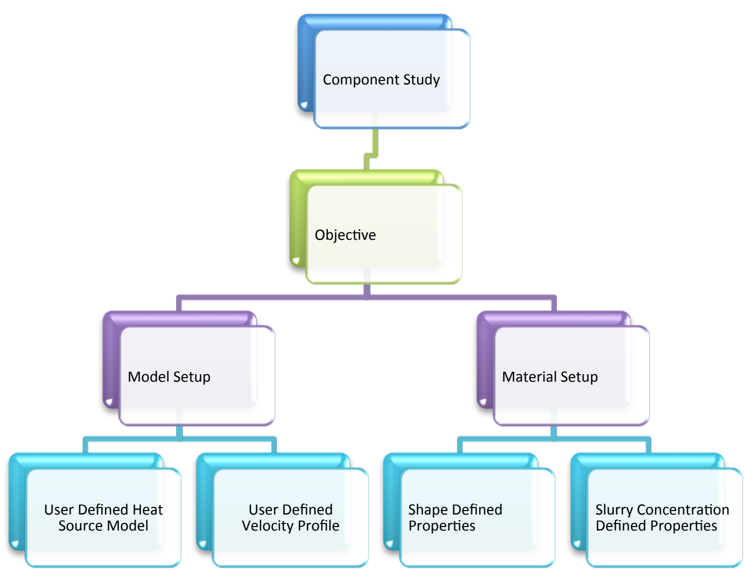

Figure 1-2 Proposed study flow

Chapter 2

Literature review

2.1 Evaporative Cooling System

Evaporative cooling system works on the principle of vapor-compression or absorbtion refrigeration cycle. A typical evaporative cooling system consists of a supply fan to flow the hot air from the rack units through the cooling unit, a sump and a pump for for the heat transfer fluid and a refrigerant compressor. Evaporative cooling systems are classified into two types of system; direct evaporative cooling system and indirect evaporative cooling system. The direct evaporative cooling system works on the phenomenon of adiabatic cooling. The cooling process takes place when the water comes in contact with air and vaporizes because of the tendency of the air to equalize temperature and vapor pressure. The Indirect evaporative cooling system uses coils carrying heat transfer fluid like water, kind of like a heat exchanger to cool the air, which helps in avoiding addition of moisture to the room air stream and also reduces both dry bulb and wet bulb temperatures achieving wet bulb depression efficiency of upto 75-80%. For example cooling from 95°F to 75°F when ambient wet bulb temperature is 65°F or lower [7] can be achieved.

2.2 Thermal Energy Storage

As part of ongoing efforts to cut down the costs and save power in data centers, thermal energy storage techniques are widely being explored. Storing of thermal energy takes place in the form of change in the internal energy of a material either as sensible heat, latent heat or as a combination of both. In sensible heat storage, change in temperature of the material takes place during the process of charging and discharging without any phase change. In latent heat storage, capture or release of heat takes place when a material experience a phase change from solid to liquid or liquid to gas; such materials are termed phase change materials (PCM). The amount of heat stored depends on the mass of the heat storage medium (m), its average specific heat (Cp) and temperature change

$$Q = mC_p\Delta T \qquad (1)$$

There are three types of thermal energy storage process, namely sensible heat storage, latent heat storage and thermo-chemical storage. Latent heat storage materials that are used to store thermal energy through change of state are known as phase change materials (PCMs). Latent heat based TESs (LHTESs) show advantages of high storage density and small temperature swing. Furthermore the wide variety of PCMs' phase change temperatures makes it possible to tailor each of the specific applications with suitable working conditions.

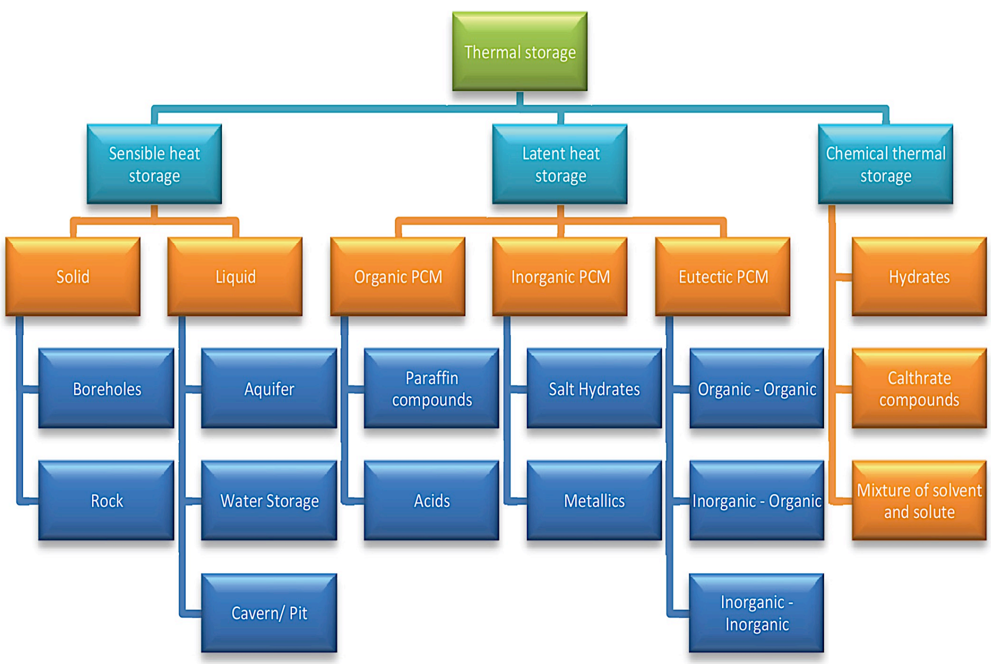

Figure 2-1 Thermal Energy Storage System Classification [17]

2.3 Phase Change Materials

The evaporative cooling systems use chilled water as the heat transfer fluid (HTF). Water takes up or rejects heat sensibly. Recently lot of research and resources are being expended in the investigation of alternative, which would increase the heat capacity and efficiency. One such option is the use of phase change materials (PCM). The use PCM's as thermal energy storage system has been of interest since 1940s. PCM's absorb latent heat as the ambient temperature rises up to their melting temperature. These materials unlike water which is only a single phase flow fluid, have the ability to change phases like for instance solid to liquid, liquid to vapor and vice versa. These phase change materials have high heat of fusion which store and release thermal energy dusing the phase change process.

The phase change materials are classified into namely three types, organic, inorganic and eutectic. Each of them have their advantages and disadvantages. Organic PCMs have high thermal stability, high heat of fusion, self nucleating properties but they are flammable and have low thermal cunductivity, low enthalpy. Whereas Inorganic PCMs are non flammable and have hight thermal conductivity and thermal storage capacity but they are corrosive and have low stability. Eutectic PCM's have high volumetric storage density, but their availability is limited. Inspite of the large diversity of PCM's, the number of chemical materials suitable for TES is still limited. So selecting the appropriate PCM for the job depends desired working temperatures such as meting and solidifing temperatures, density, chemical stability, thermal conductivity, viscosity, low cost, etc. The size of these Micro-encapsulated phase change materials (mPCM) ranges from 1 to 1000μm.

The drawbacks of some of these phase change materials can be overcomed by micro-encapsulating them. Micro-encapsulation is a process of coating and enclosing small spherical or rod-shaped particles within a thin layer of high molecular weight polymeric film. Micro-encapsulating the phase change materials has certain advantages. The thermal conductivity of the PCMs can be enhanced improving heat transfer rate, corrosion can be avoided, particles can be suspended withing a continuous phase like water thereby improving the thermal performance of the combined slurry and solves contamination problem by isolating the core material from the entire system when it changes its phase.

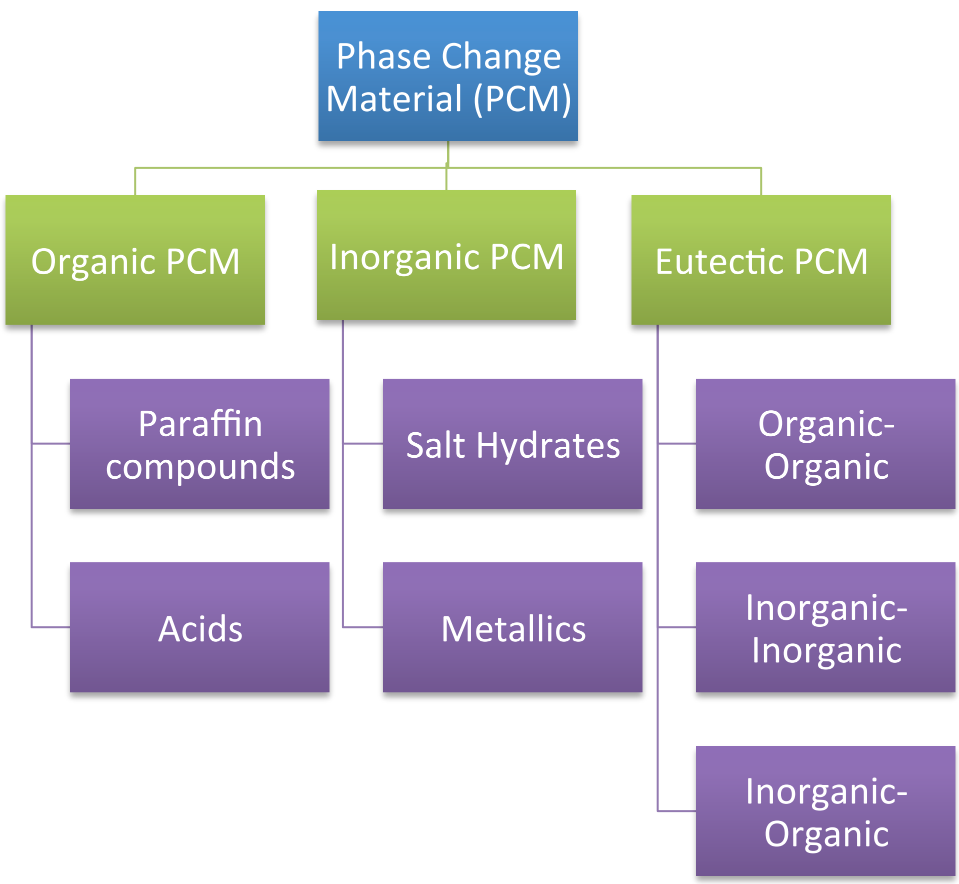

Figure 2-2 Classification Phase change materials

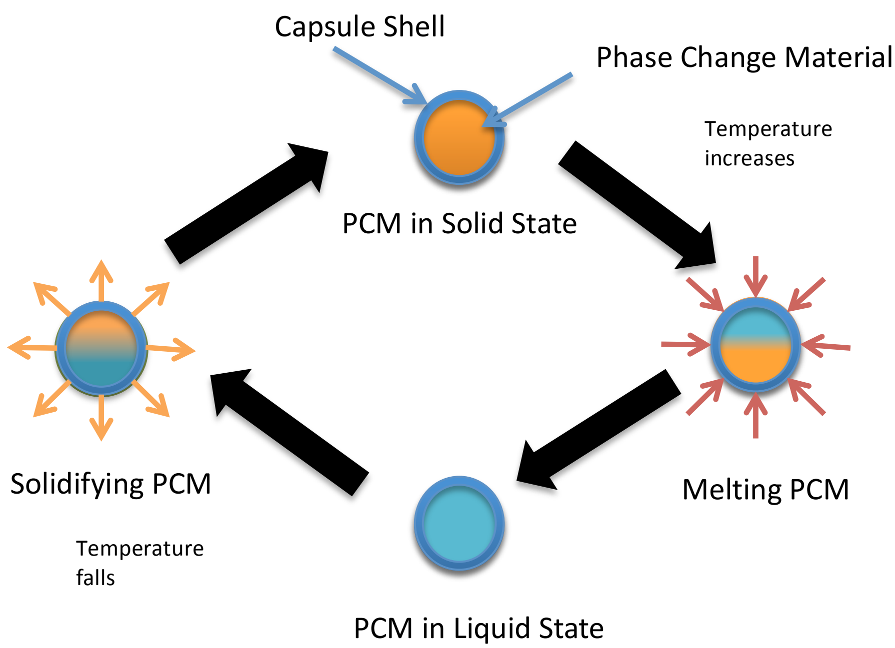

Figure 2-3 Phase Change Process

For a phase change material to be optimally efficient, its Stefans number should be lower than 1 and its performance depends on the mass fraction and the latent heat of fusion as can be seen below[4]:

$$St = \frac{C_p\left(Q_w \frac{r}{k}\right)}{c\lambda} \tag{2}$$

Where:

St = Stefan number

C_p = Specific heat

Q_w = Heat flux across the pipe wall

r = Radius of the pipe

k = Thermal conductivity

c = Volumetric mass fraction

λ = Latent heat of fusion of the PCM

For the data centers, phase change materials should be selected basing on the melting temperatures recommended by the ASHRAE guidelines. Another hurdel in implimenting the PCM is the clogging issue, due to particle to surface and particle to particle interaction within a flow.

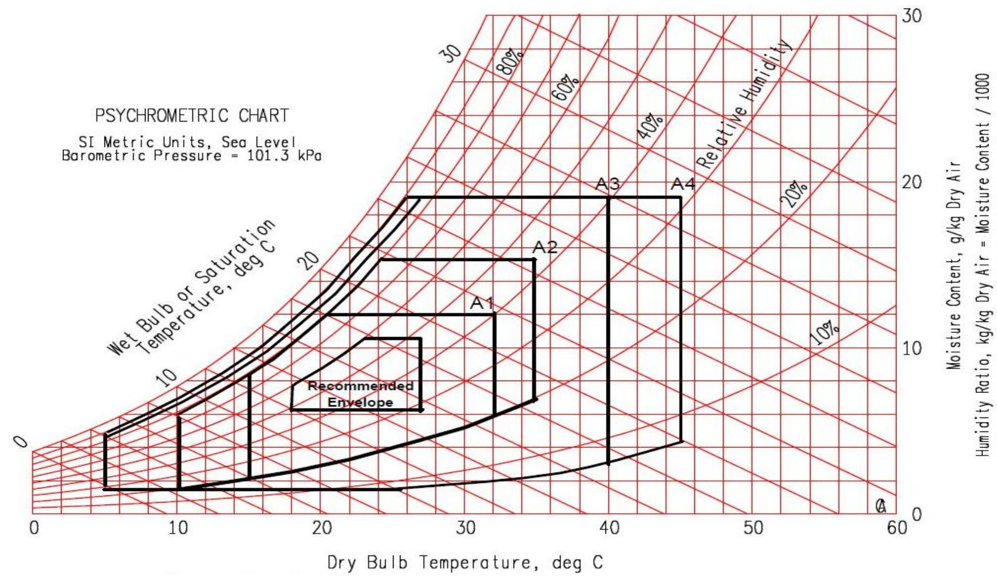

Figure 2-2 Psychometric Chart [4]

To overcome this, emulsifier should be mixed in along with mPCMs. The emulsifier helps in preventing the sticking of particles with eachother and also helps in minimizing the pressure drop. The advantages of PCM's are many but their effectiveness depends on several conditions which makes it even that more complex to model and analyse. Numerical method provides a way to account for a wide range of conditions and operating modes for the PCMs.

Table 2-1 Phase change materials

Core	Shell Material	Size distribution	Melting temperature	Phase change enthalpy
Tetradecane [9]	Polyethyl methacrylate (PEMA)	5-30 µm	5.68°C	80.62 KJ/kg
Paraffin [10]	Urea-formaldehyde	5-20 µm	~54°C	157.5 KJ/kg
Paraffin Wax [11]	Polymethyl methacrylate (PMMA)	0.15-0.33 µm	20°C-50°C	145-240 KJ/kg
Docosane [12]	Polymethyl methacrylate (PMMA)	0.14-0.466 µm	41°C	54.6 KJ/kg
n-Octadecane [13]	Melamine formaldehyde	2.2 µm	40.6°C	144 KJ/kg
1-Bromohexadecane [14]	Amino plastic	8.2 µm	14.3°C	13 KJ/kg
Micronal DS (BASF)		1-20 µm	~29°C	102.008 KJ/kg
DPNT06-0182 (Ciba Specialty Chemicals) [15]		10-100 µm	~35°C	96.968 KJ/kg

Table 2-2 Commercially available Phase change materials

Manufacturer	Product	PCM	Melting temperature	Latent heat
BASF	DS 5000	Paraffin	26°C	45 KJ/kg
	DS 5007	Paraffin	23°C	41 KJ/kg
	DS 5030	Paraffin	21°C	37 KJ/kg
	DS 5001	Paraffin	26°C	110 KJ/kg
	DS 5008	Paraffin	23°C	100 KJ/kg
Microtek Laboratories	MPCM-30D	n-Decane	-30°C	140-150 KJ/kg
	MPCM-10D	n-Dodecane	-9.5°C	150-160 KJ/kg
	MPCM 6D	n-Tetradecane	6°C	157-167 KJ/kg
	MPCM 18D	n-Hexadecane	18°C	163-173 KJ/kg
	MPCM 28D	n-Octadecane	28°C	180-195 KJ/kg
	MPCM 37D	n-Eicosane	37°C	190-200 KJ/kg
Capzo	Thermusol HD35SE	Salt hydrate	30°C-40°C	200 KJ/kg
	Thermusol HD60SE	Salt hydrate	50°C-60°C	160 KJ/kg

Chapter 3

Heat transfer Model

3.1 Analytical Model

A simple latent heat storage unit consisting of a tube carrying heat transfer fluid, which in this case is the phase change slurry, was designed to be studied using steady state analysis for laminar flow. For this mathematical model, the tube carrying the heat transfer fluid is considered to be unformaly heated by constant heat flux on wall to simulate the flow of hot air around the pipe. To include the effects of a latent heat source material a user defined volume heat source function is considered.

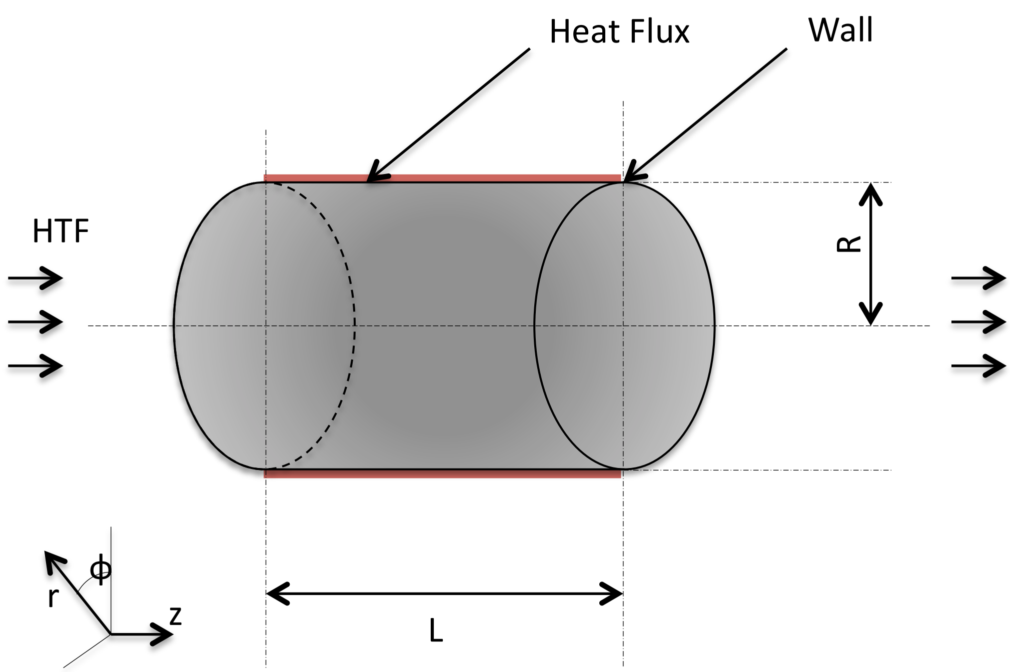

Figure 3-1 Model of the duct

3.1.1 Mathematical Formulation

The model is a pipe of length L and radius R. A standard differential heat transfer equation was reduced to the form as seen below for the case of laminar flow

$$2u_m\left[1-\left(\frac{r}{R}\right)^2\right]\frac{\partial T}{\partial z}=\frac{1}{r}\frac{\partial}{\partial r}\left[\frac{k}{\rho c_p}r\frac{\partial T}{\partial r}\right]+\frac{\nu}{\rho c_p} \tag{3}$$

Where,

u_m	= Mean fluid velocity in the pipe
k	= Thermal conductivity
c_p	= Specific heat capacity
ρ	= Fluid density
ν	= Uniform volume heat source
T	= Temperature
z	= Axial distance
r	= Radial distance

The latent heat effect of the phase change material or in other words the phase change process will be controlled by the term ν. As the flow progress downstream from the entrance the flow region becomes uniform and fully developed. The axial fluid temperature change becomes equal to mean axial fluid temperature.

i.e,

$$\frac{\partial T}{\partial z}=\left.\frac{\partial T}{\partial z}\right|_{mean} \tag{4}$$

On further simplifying eq(3) we get

$$Q_R 2\pi R\,dz-\nu\pi {R_m}^2\,dz=\pi R^2 u_m\rho c_p\left.\frac{\partial T}{\partial z}\right|_{mean} \tag{5}$$

Hence the change in temperature across the flow region is given by

$$\frac{\partial T}{\partial z} = \left.\frac{\partial T}{\partial z}\right|_{mean} = \frac{2Q_R R - \nu R_m^2}{R^2 u_m \rho c_p} \tag{6}$$

Substituting eq(6) into eq(3) we get

$$2\left(1 - \left(\frac{r}{R}\right)^2 \left(\frac{2Q_R R - \nu {R_m}^2}{R^2}\right)\right) = \frac{1}{r}\frac{\partial}{\partial r}\left(kr\frac{\partial T}{\partial r}\right) + \nu \tag{7}$$

Integrating eq(7);

$$kr\frac{\partial T}{\partial r} = \int \left[2\left(1 - \left(\frac{r}{R}\right)^2 \left(\frac{2Q_R R - \nu {R_m}^2}{R^2}\right)\right) - \nu\right] r\, dr \tag{8}$$

Upon integrating there results

$$kr\frac{\partial T}{\partial r} = \int \left[2\left(1 - \left(\frac{r}{R}\right)^2 \left(\frac{2Q_R R - \nu {R_m}^2}{R^2}\right)\right) - \nu\right] r\, dr \tag{9}$$

$$k\frac{\partial T}{\partial r} = \frac{1}{r}\left[\left(\frac{4Q_r}{R}\frac{r^2}{2} - \frac{4Q_R}{R^4} R\frac{r^4}{4}\right)\left[-\frac{2\nu {R_m}^2}{R^2}\frac{\nu}{2} + 2\nu\left(\frac{R_m}{R}\right)^2 \frac{1}{R^2}\frac{r^4}{4} - \frac{\nu r^2}{2}\right]\Bigg|_0^{R_m}\right] \tag{10}$$

$$= \frac{1}{r}\left[\frac{2Q_R}{R}\left(r^2 - \frac{r^4}{2R^2}\right) - \frac{2\nu {R_m}^4}{R^2} + \frac{\nu {R_m}^6}{2R^4} - \frac{\nu {R_m}^2}{2}\right] \tag{11}$$

$$= \frac{1}{r}\left[\frac{2Q_R}{R}\left(r^2 - \frac{r^4}{2R^2}\right) - \frac{\nu}{2}{R_m}^2\left(2\left(\frac{R_m}{R}\right)^2 - \left(\frac{R_m}{R}\right)^4 + 1\right)\right] \tag{12}$$

$$= \frac{2Q_R}{R}\left(r - \frac{r^3}{2R^2}\right) - \frac{\nu}{2r} {R_m}^2 \left(2\left(\frac{R_m}{R}\right)^2 - \left(\frac{R_m}{R}\right)^4 + 1\right) \qquad (13)$$

The boundary condition that at wall the uniform heat flux will be positive, negative or zero. Upon integrating this with the eq(12) we get

$$\int_0^r k\frac{\partial T}{\partial r} dr = \int_0^r \left[\frac{2Q_R}{R}\left(r - \frac{r^3}{2R^2}\right) - \frac{\nu}{2r} {R_m}^2 \left(2\left(\frac{R_m}{R}\right)^2 - \left(\frac{R_m}{R}\right)^4 + 1\right)\right] dr \qquad (14)$$

$$k(T - T_o) = \left[\frac{2Q_R}{R}\left(\frac{r^2}{2} - \frac{r^4}{8R^2}\right) - \frac{\nu}{2} {R_m}^2 \left(2\left(\frac{R_m}{R}\right)^2 - \left(\frac{R_m}{R}\right)^4 + 1\right) \ln r\right]_0^r \qquad (11)$$

$$= \frac{2Q_R}{R}\left(\frac{r^2}{2} - \frac{r^4}{8R^2}\right) - \frac{\nu}{2} {R_m}^2 \left(2\left(\frac{R_m}{R}\right)^2 - \left(\frac{R_m}{R}\right)^4 + 1\right) (\ln R_m - 1) \qquad (12)$$

Rearranging eq(15) we get

$$Q_r = -k\frac{\partial T}{\partial r} = -\left[\frac{2Q_R}{R}\left(\frac{r^2}{2} - \frac{r^4}{8R^2}\right) - \frac{\nu}{2} {R_m}^2 \left(1 + 2\left(\frac{R_m}{R}\right)^2 - \left(\frac{R_m}{R}\right)^4\right)\right] \qquad (13)$$

The eq(16) can be used to determine the radial temperatrue distribution in flowing fluids that possess internal source of heat generation. To validate it, the results obtained experimentally by Goel and Roy (1994) served for comparison.

3.2 Computational Model

Computational Fluid Dynamics (CFD) softwares use different algorithms to solve and analyze fluid flow models. The calculations required simulating the interaction of fluids with internal heat source defined by boundary conditions, and initial conditions are done by the ANSYS Fluent V15.0. By default FLUENT uses enthalpy-porosity method, developed from the Stefan Problem which does not explicitly track the solid-liquid interface. Contrary to the enthalpy-porosity method, the method which was used to analyze the solidfication/melting process calculates the energy stored/released directly from the temperature in the form of a user defined function (UDF) thereby eliminating the dependency of the enthalpy function to obey the lever rule and provides a more flexible approach.

A crucial aspect of the model is to differentiate between melting and freezing, so that the solver uses the appropriate heat source function. Melting is an endothermic process, i.e. absorbing heat as the temperature of the material increases, and freezing is an exothermic process, releasing heat as the temperature decreases. As a result, melting will be mimicked through a heat sink, while freezing as a heat source, with the corresponding change in temperature, incorporated in the source term S_E .The energy absobed/released by the PCM is given by the heat source function, which can be derieved from the energy equation(17) given below[18];

$$\frac{\partial}{\partial t}\rho H = -\frac{d}{dx}\rho u C_p T + \frac{d}{dx}\left[\lambda \frac{dT}{dx}\right] + S_e \tag{14}$$

3.2.1 Meshed Geometry

The case considered is a laminar; two-dimensional melting of a phase-change material (PCM) flowing in a circular duct of dimensions 10mm long and 1mm wide. The assumption of two-dimensionality follows from the fact that the unit is sufficiently long in the third dimension. For the ease of simulation a axisymmetric model is used.

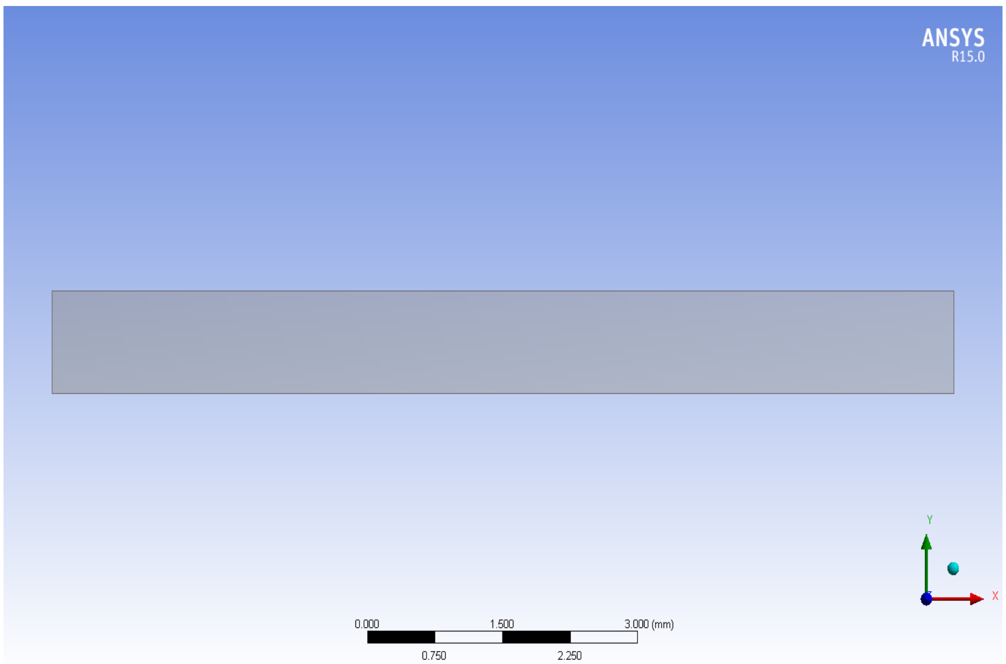

Figure 3-2 Model geometry

A coarser mesh is generated with mapped face meshing to get more precise meshing near the wall of the duct. This mesh contains cells having quadrilateral faces. Care is taken to use structured hexahedral cells as much as possible..

The mesh details are as follows:

- Relevance centre: coarse meshing
- Smoothing: medium

- Size: 5.0092e-003mm to 1.0018mm
- Nodes: 561
- Elements: 500

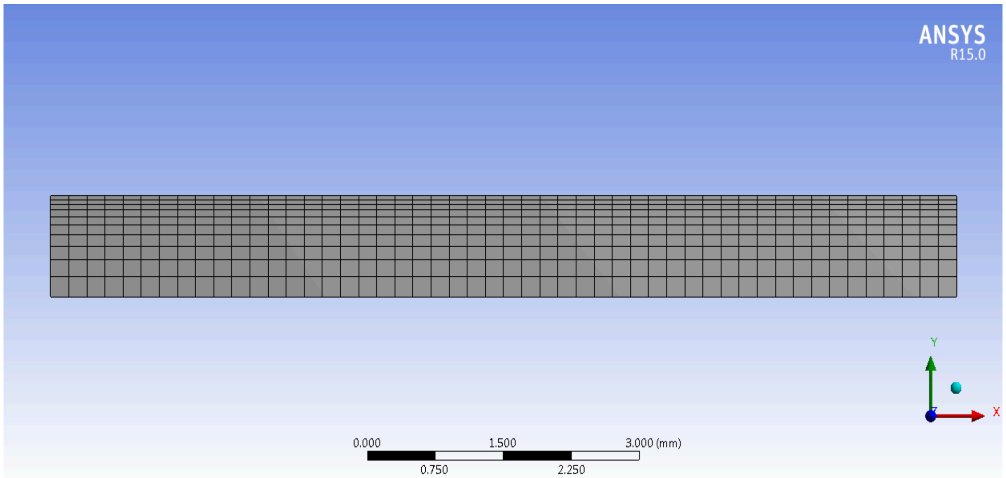

Figure 3-3 Meshed geometry

3.2.2 Fluent Setup

3.2.2.1 Material Setup

The PCM Micronal Ds 5000 is paraffin wax based material. It is commercially available as slurry. This slurry contains 6-8% emulsifier RT20. Since the effect emulsifier on the thermal properties of the PCM slurry is not that significant, it is not cosidered in the simulation.

Table 3-1 Material Properties

Property	Reference values
Density of Water (kg/m^3)	999.97
Density of Paraffin Wax (kg/m^3)	900
Density of PMMA (kg/m^3)	1180
Viscosity of Water (kg/ms)	9.070E-04
Latent Heat of Paraffin wax (J/kg)	44000
Solidus temperature (K)	298
Liquidus temperature (K)	296
Thermal conductivity of water (W/mK)	0.606
Thermal conductivity of Paraffin wax (W/mK)	0.21
Thermal conductivity of PMMA(W/mK)	0.21
Specific Heat capacity of water (J/kgK)	4180
Specific Heat capacity of Paraffin wax (J/kgK)	2000
Specific Heat capacity of PMMA (J/kgK)	1466

The material properties of the PCM are given as bulk supension properties which can be calculated from the equations below basing on properties in above table 3-1. For this experiment Micronal DS 5000 is chosen. It contains in the core of the microcapsule a latent heat storage material made from a special wax mixture.

3.2.2.1.1 Phase change particle properties

Table 3-2 Nomenclature

d	Diameter
ρ	Density
h	Height of cylinder
C	Specific Heat
k	Thermal Conductivity
c	Volumetric Concentration
μ	Viscocity
Suffixes	
c	Core (Paraffin wax)
p	PCM particle
w	Wall (PMMA)
f	Fluid (Water)
b	Bulk or suspension

3.2.2.1.2 Case Setup

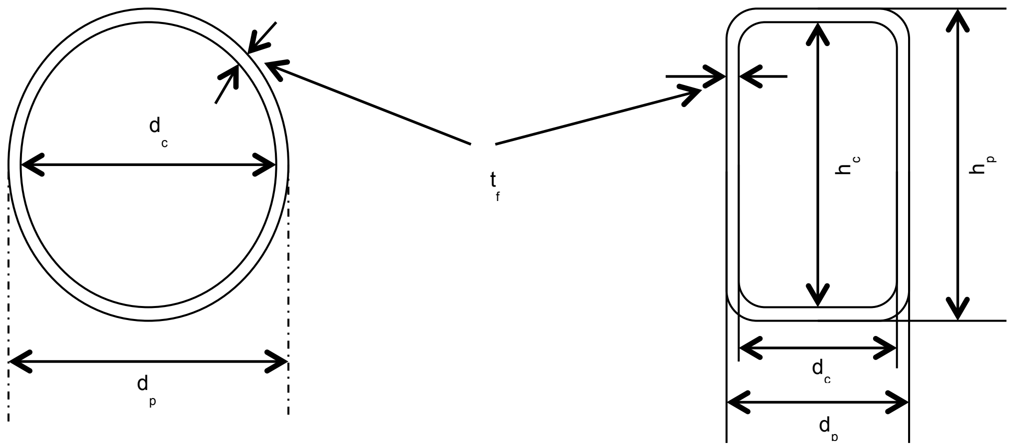

Figure 3-4 Case study of different shapes of particle

For the analysis, the study was done for slurry containing spherical shaped particles and the other having oval or rod-shaped particles.

Where;

d_p - Diameter of the particle

d_c - Diameter of the core

t_f - Flim thickness (5 microns)

h_c - Height of the core cylinder

h_p - Height of the particle

The material properties were calculated for four separate cases.

- Case 1: d_c of the spherical shaped particle is varied from 2.5 microns to 10.5 microns keeping the volumetric concentration of the slurry the same.
- Case 2: h_c of the rod shaped particle is varied from 11 microns to 31 microns keeping the volumetric concentration of the slurry the same.

- Case 3 and Case 4: Volumetric concentration of the slurry is varied from 10-50% with Spherical and rod-shaped PCM Particle

Density of spherical and rod-shaped particles;

$$\rho_{p_{spherical}} = \frac{900 d_c{}^3 + 1180\left(d_p{}^3 - d_c{}^3\right)}{d_p{}^3} \quad 15)$$

$$\rho_{p_{rod-shaped}} = \frac{900 h_c d_c{}^3 + 1180\left(h_p d_p{}^3 - h_c d_c{}^3\right)}{h_p d_p{}^3} \quad 20)$$

Specific heat capacity;

$$C_p = \frac{(7C_c + 3C_w)\rho_c \rho_w}{(3\rho_c + 7\rho_w)\rho_p} \quad 21)$$

Thermal Conductivity of the particles;

$$k_{p_{spherical}} = \frac{k_c k_w d_c}{k_w d_p + k_c\left(d_p - d_c\right)} \quad 22)$$

$$k_{p_{rod-shaped}} = \frac{h_p k_c k_w d_c}{h_c d_p \left[k_w + k_c d_c \ln\left({}^{d_p}/_{d_c}\right)\right]} \quad 16)$$

3.2.2.1.3 Bulk suspension properties

Bulk density

$$\rho_b = \rho_f - \left(c\left(\rho_f - \rho_p\right)\right) \quad 17)$$

Volumetric heat capacity;

$$C_b = \frac{c\rho_p c_p + (1-c)\rho_f C_f}{\rho_b}$$ 18)

Bulk thermal conductivity;

$$k_b = k_f\left(\frac{2k_f + k_p + 2c(k_p - k_f)}{2k_f + k_p - 2c(k_p - k_f)}\right)$$ 19)

Slurry viscosity;

$$\mu_b = \mu_f(1 - c - 1.16c^2)^{-2.5}$$ 20)

3.2.2.2 Boundary Conditions

Boundary conditions are used according to the need of the model. To simulate the flow of air over the duct a uniform heat flux is applied to the wall. To elimnate all the anomalies due to the flow pattern the flow region is analysed.

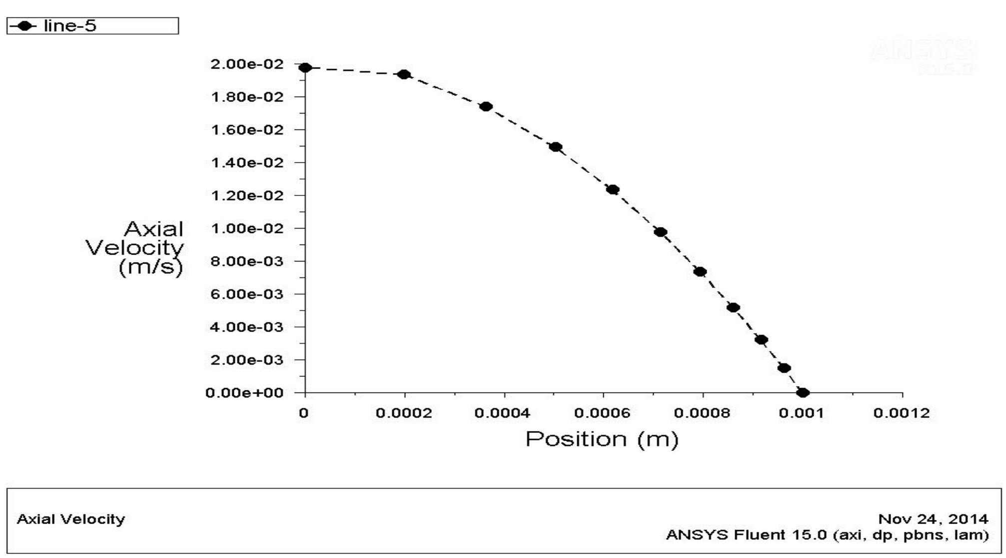

Figure 3-5 Velocity profile without UDF

It was found that the flow becomes fully developed at 0.2mm from the center which can be seen in the figure 3-5. To make the flow completely streamlined a UDF for velocity inlet is integrated into the flow making the entire flow in the duct fully developed.

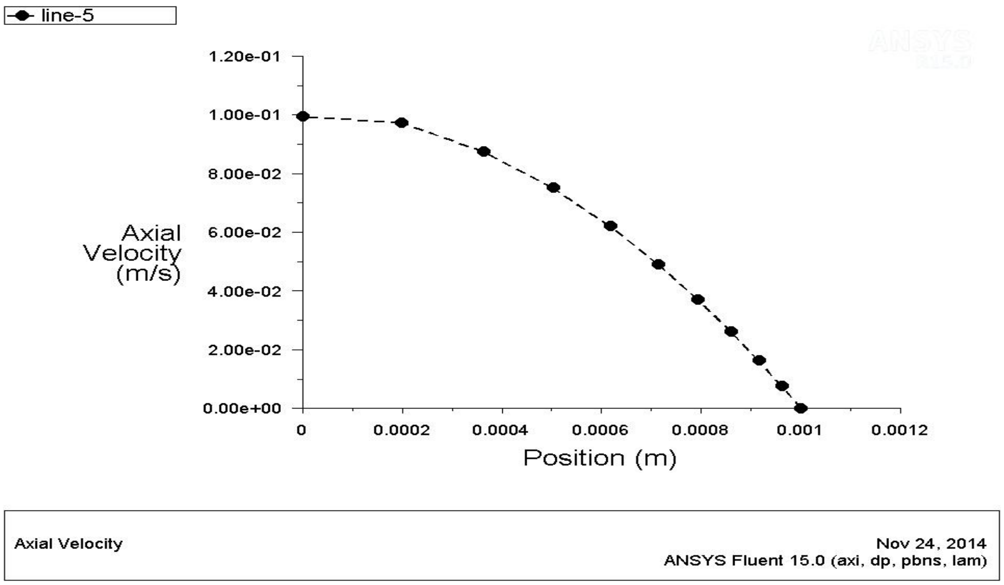

Figure 3-6 Velocity profile with UDF

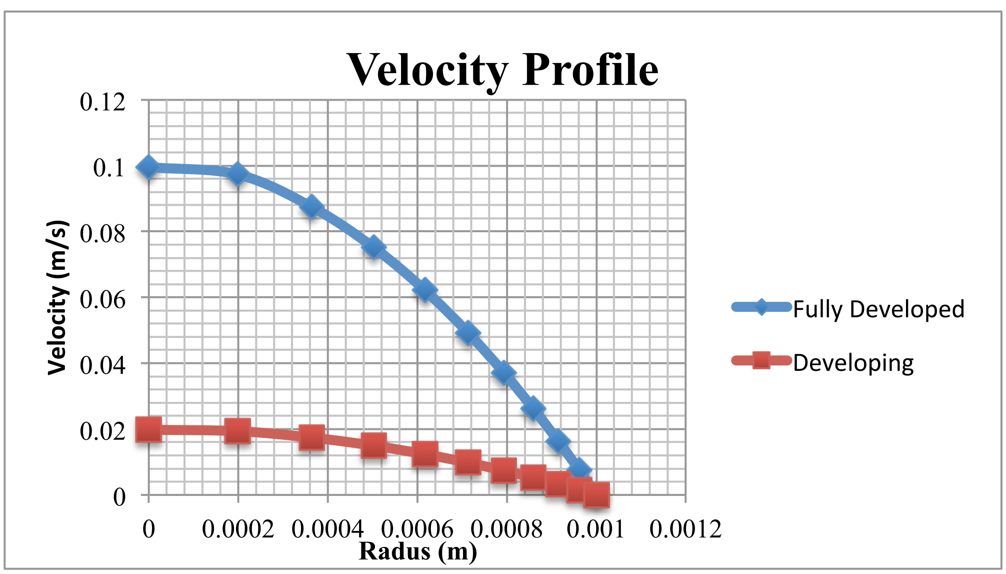

Figure 3-7 Velocity profile with and without UDF

The figure 3-7 shows the change in velocity profile before and after the integration of the user defined velocity profile. The user defined function makes the flow stream fully developed.

3.2.2.3 Reference Values

The velocity inlet is selected from the drop down list of “compute from”. The values are:

- Length = 1 m
- Temperature = 310 K
- Ratio of specific heats = 1.4

Chapter 4

Results and Discussion

Forced flow volume heat source analysis was done on a single channel heat exchanger carrying mPCM at heat transfer fluid. The temperature plots for various cases were obtained.

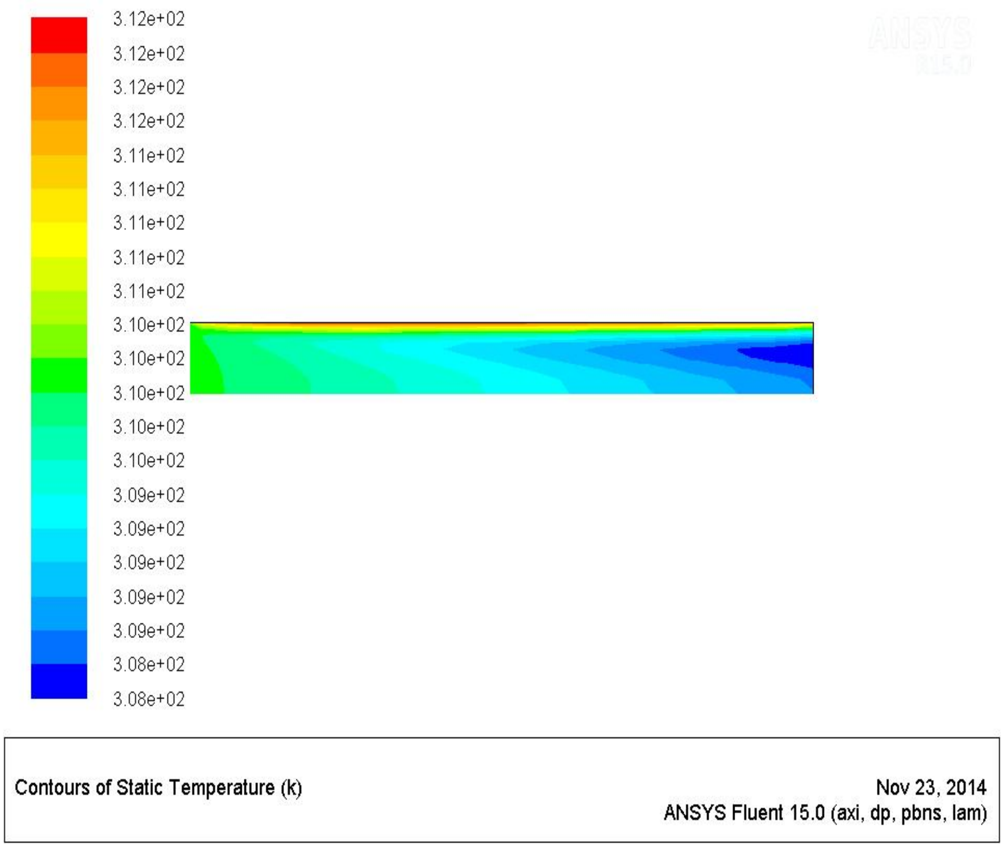

Figure 4-1 Temperature profile

Figure 4-1 shows the temperature plot along the duct. The working of the PCM can be seen downstream of the flow.

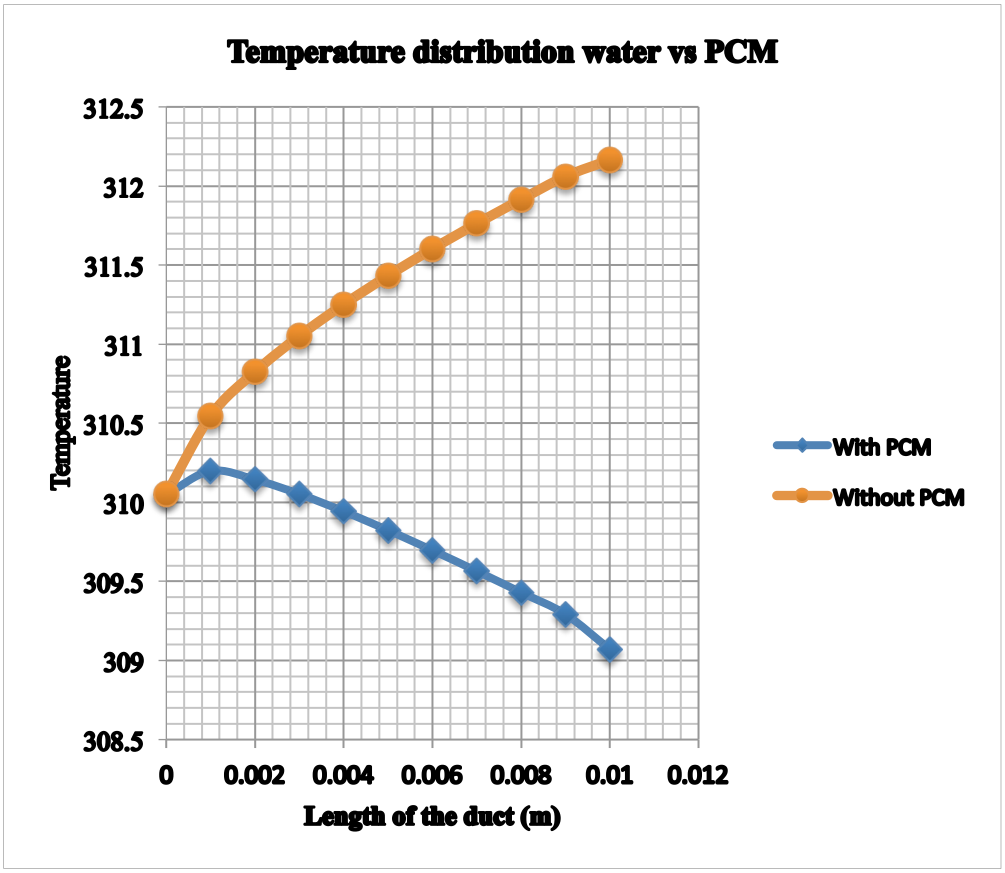

Figure 4-2 Temperature distribution water vs. PCM

In figure 4-2 we can see the difference in in temperatures between water and PCM slurry through the duct. The difference in heat flux between the water and PCM slurry gives the potential of PCM to absorb heat.

4.1 Change in Particle size/diameter

Effects of change in size of the particle were analyzed keeping the volumetric concentration of the surry at 42%

4.1.1 Case 1: Spherical Shaped Particle

The diameter of particle is varied from 2 microns to 10 microns.

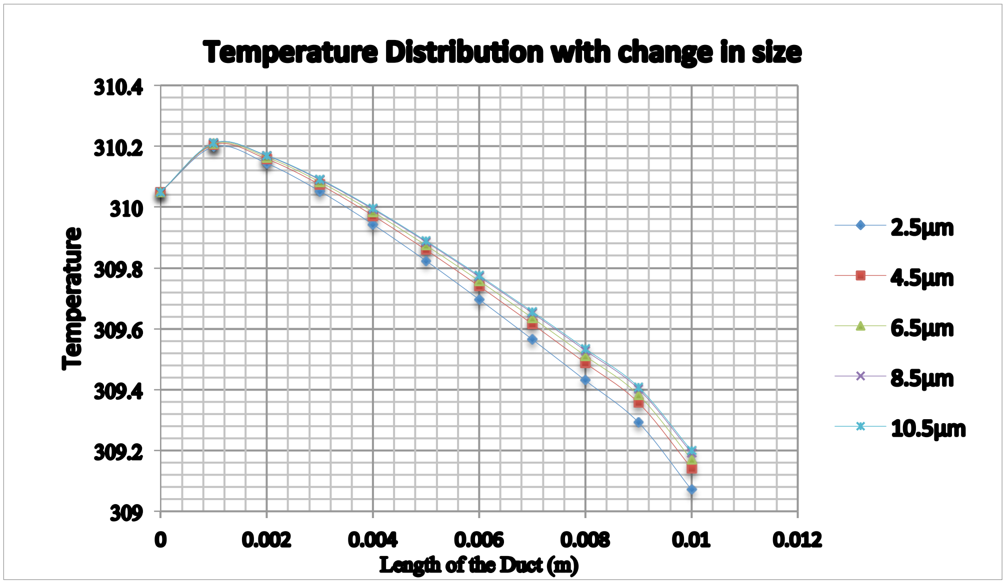

Figure 4-3 Temperature distribution with change in size of a spherical particle

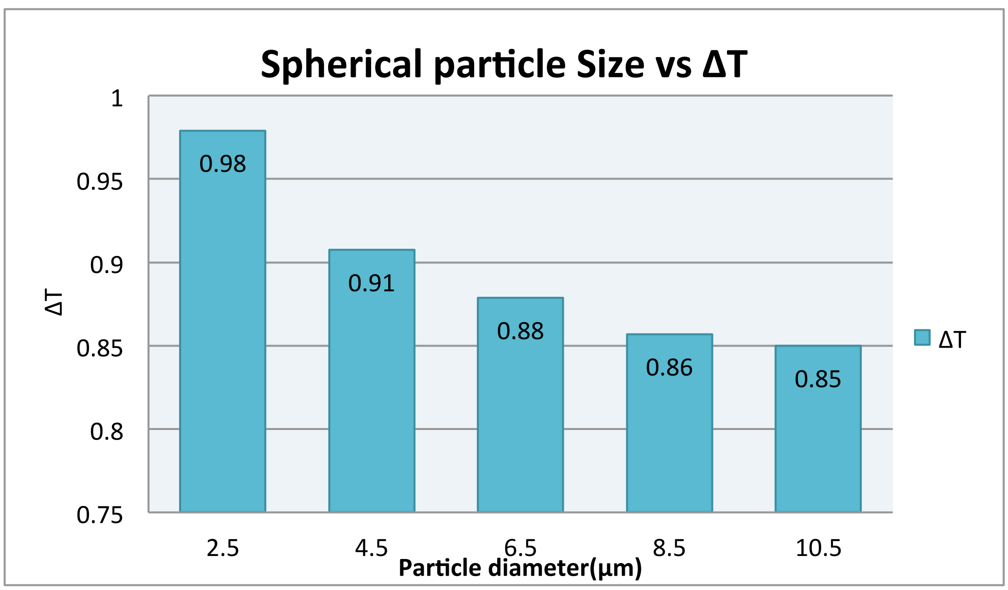

Figure 4-4 Heat Flux because of change in size of a spherical particle

*4.1.2 Case 2:	Rod-Shaped Particle*

The length of particle is varied from 10 microns to 30 microns.

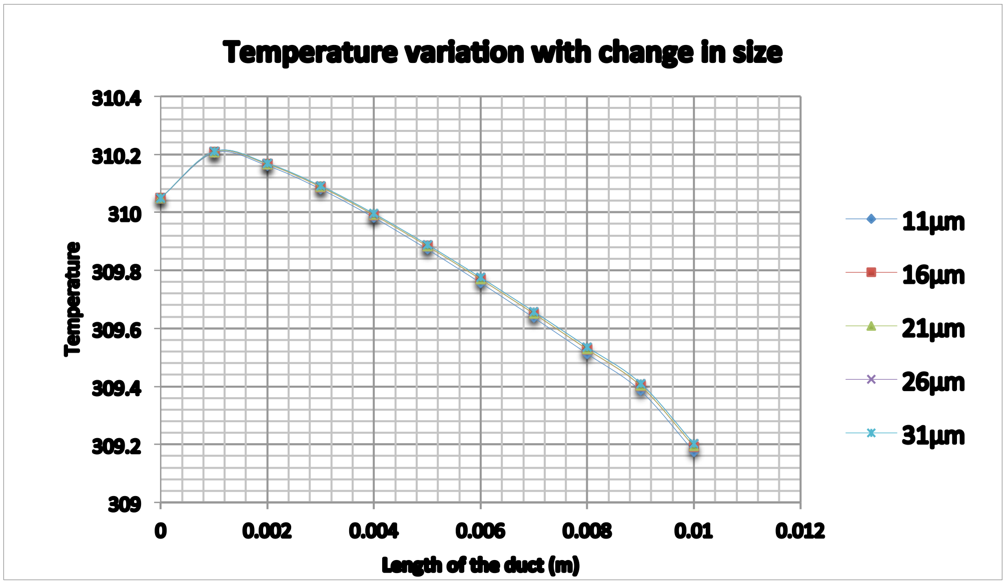

Figure 4-5 Temperature distribution with change in size of a rod shaped particle

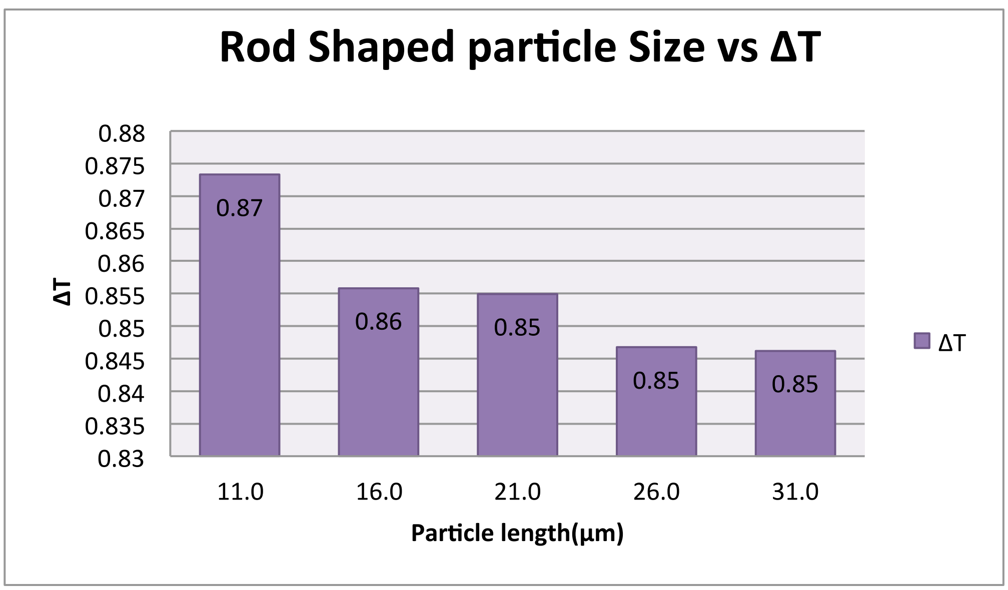

Figure 4-6 Heat Flux because of change in size of a rod-shaped particle

4.2 Change in concentration of the slurry mixture

Effects of change in volume of fraction of the slurry were analyzed for concentrations rangein from 10% to 50%

4.2.1 Case 3: Spherical-Shaped Particle

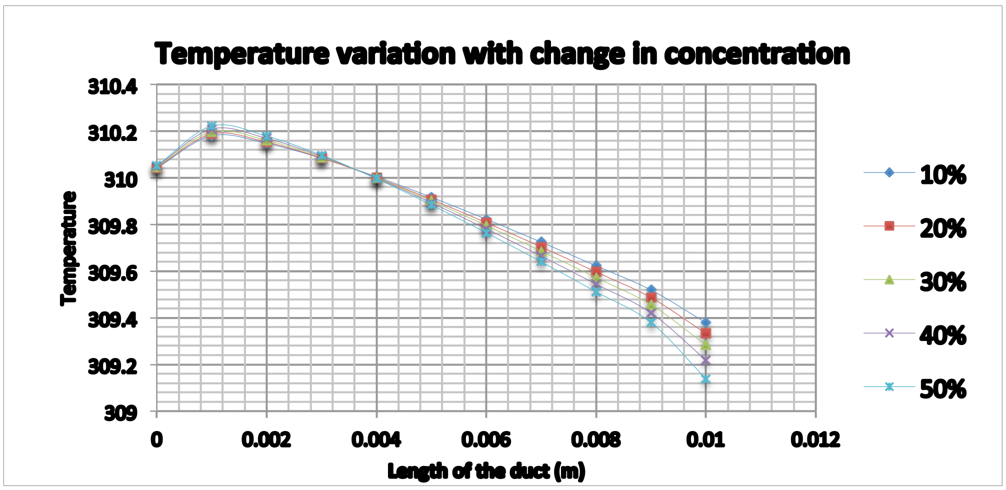

Figure 4-7 Temperature distribution with change in volumetric concentration of the spherical shaped PCM in slurry

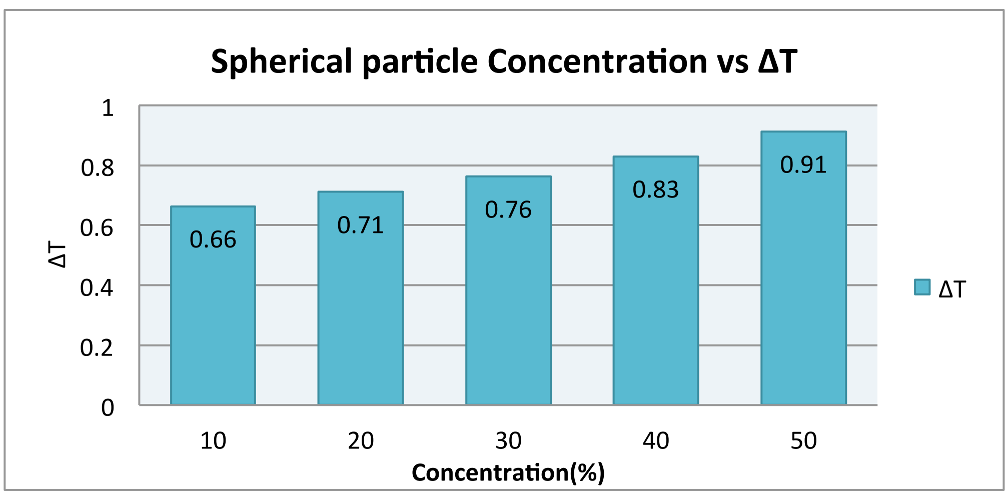

Figure 4-8 Change in Heat Flux with change in volumetric concentration of the spherical shaped PCM in slurry

4.2.2 Case 4: Rod-Shaped Particle

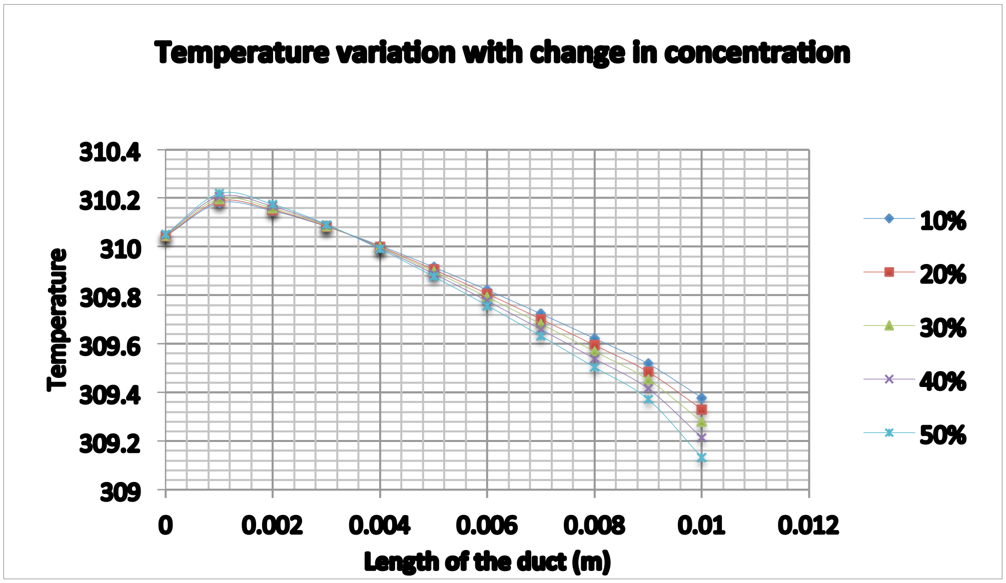

Figure 4-9 Temperature distribution with change in volumetric concentration of the spherical shaped PCM in slurry

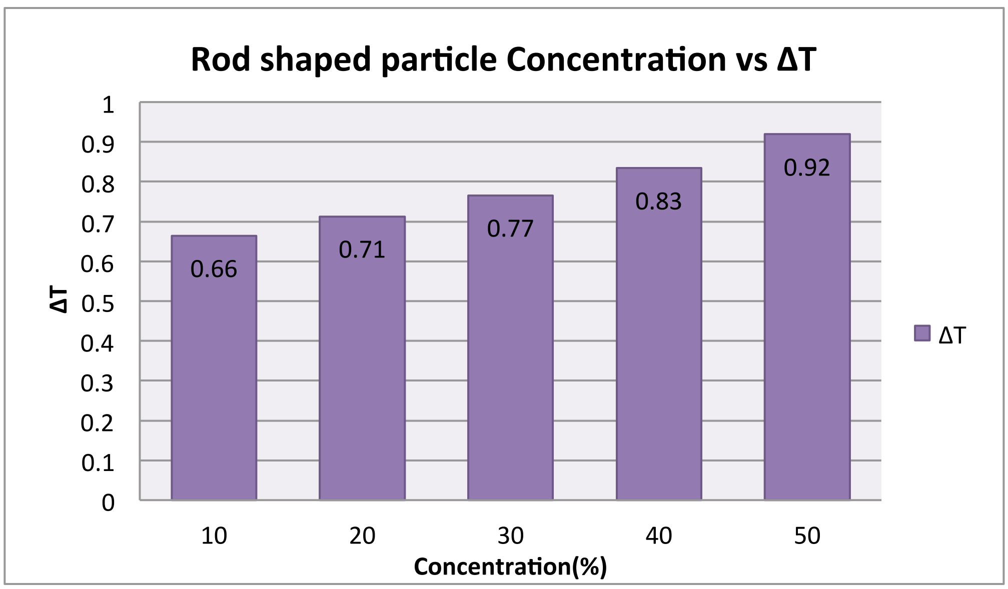

Figure 4-10 Change in Heat Flux with change in volumetric concentration of the Rod shaped PCM in slurry

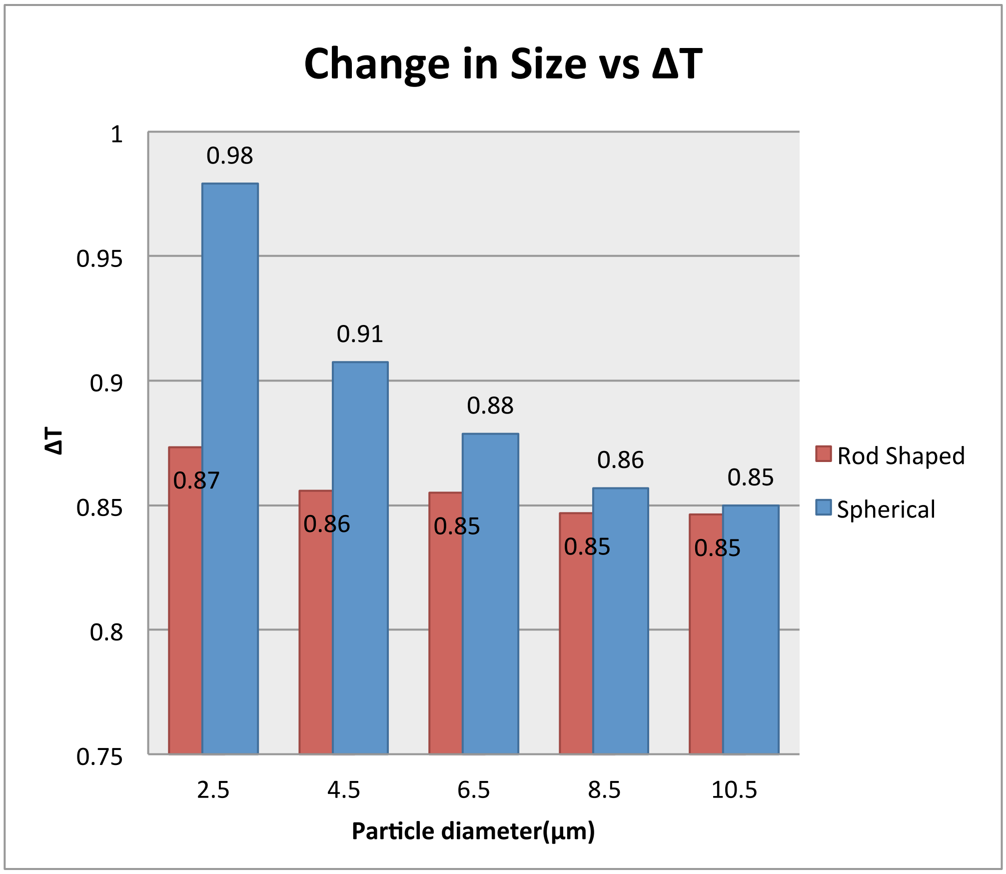

Figure 4-11 Comparison of Heat flux variation with the change in size of the particle

Temperature difference across the length of the circular duct from inlet to outlet is a measure of the potential for heat extraction from ambient. Temperature trends illustrated in Figures X-Y demonstrate that the modeling techniques employed in this project are consistent with theoretical understanding of flow and heat transfer in circular duct flow. Temperature difference across the length of the duct decreased with particle diameter. This may be due to the decreased surface area to volume ratio in particles of larger size.

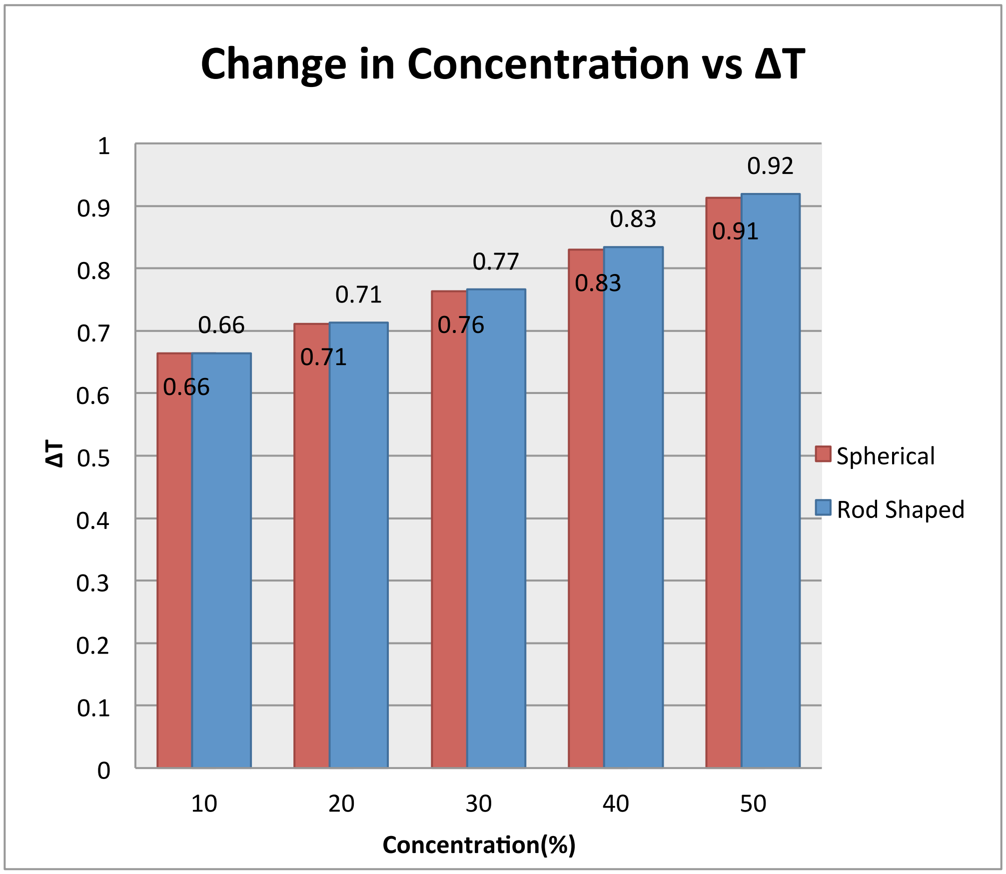

Figure 4-12 Comparison of heat flux variation with the change in volumetric concentration of the slurry

Temperature difference across the length of the duct increased with PCM concentration. Spherical particles have great potential for heat extraction when compared to rod shaped particles.

Chapter 5

Conclusion

The use of Phase change materials is good for energy storage in latent heat storage system. In this study heat extraction from ambient through a circular duct was simulated using constant heat flux through the lateral wall and parabolic velocity input at inlet and heat absorption through melting defined by a sink regional and boundary conditions. A user defined function to input parabolic inlet velocity profile was developed to simulate fully developed flow. This enabled comparison of heat transfer through a circular duct of same length over developing and fully developed flow situations. A user defined function to input source or sink terms into the energy equation was developed to study the impact of shape, size and concentration properties of phase change material slurries on heat transfer in circular duct flow. Parametric analysis of heat transfer and fluid flow through circular ducts demonstrated that for the same amount of heat influx, temperature difference across the length of the duct is significantly greater for PCM slurries when compared to water. Temperature difference across the length of the duct increased with PCM concentration and decreased when size of the particle increased. Simulation techniques elaborated in this project demonstrate the potential for future research in heat extraction using PCM slurries.

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Biographical Information

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