

Centre for Sustainable Energy Technologies

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DEVELOPMENT OF A NOVEL ENERGY EFFICIENT PHASE CHANGE EMULSION FOR AIR CONDITIONING SYSTEMS

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ABSTRACT

Buildings represent more than 40% of final global energy consumption, among which 50%-60% of energy consumption is attributed to Heating, Ventilation and Air Conditioning (HVAC) systems. The application of phase change material emulsions (PCMEs) in air conditioning systems is considered to be a potential way of saving energy because with their relatively higher energy storage capacity, they are able to reduce flow rate whilst delivering the same amount of cooling energy. PCMEs can also simultaneously act as cold energy storage to shift peak-load to off-peak time and improve coefficient of performance of systems. However, one of the main barriers affecting the application of PCME is the difficulty in maintaining stability in the emulsions without experiencing any temperature stratification during phase change process.

To this end, an innovative energy efficient phase change emulsion has been developed and evaluated. The emulsion (PCE-10) which consists of an organic PCM (RT10) and water has a phase change temperature range of 4-12°C with heat capacity of twice as much as that of water thus making it a good candidate for cooling applications. Particular attention was also paid to the selection of the surfactant blends of Tween60 and Brij52 since they are capable of minimizing the effect of sub-cooling as well as ensuring stability of the emulsion.

For the purpose of testing the performance of developed PCE-10 in fin-and-tube heat exchangers, series of theoretical and experimental studies have been carried out to understand the rheological behaviour and heat transfer characteristics of the developed PCE-10 in a fin-and-tube heat exchanger. Both experimental and theoretical results were fairly close and showed that the PCE-10 did enhance the overall heat transfer rate of the heat exchanger.

In order to evaluate the potential of the integrated system, whole building energy simulation was carried out with a building simulation code TRNSYS. It was found out that the required volumetric flow rate of PCE-10 was 50% less than that of water which is equivalent to 7% reduction in total energy consumption when providing the same amount of cooling power.

Despite its potential in cooling systems, the viscosity of the developed sample was found to be much higher than water which could contribute to high pressure drop in a pumping system. Its thermal conductivity was also found to be about 30% lower than the value for water which could influence heat transfer process. There is therefore the need to enhance these thermophysical properties in any future investigations.

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NOMENCLATURE

Fluid Properties

Variable	Description	Units
Δh_e	Total heat capacity	kJ/kg
$\Delta h_{f,e}$	Latent heat capacity of the emulsion	kJ/kg
$\Delta h_{\rm w}$	Sensible heat capacity of water	kJ/kg
Δh_{PCM}	Sensible heat capacity of paraffin	kJ/kg
$\Delta h_{f,PCM}$	Latent heat capacity of PCM	kJ/kg
$V_{\rm w}$	Volume of water	m ³
Φ	Volume fraction	Vol%
g	Gravity acceleration	m/s^2
δ_c	Viscosity of continue phase	Pas
δ_{PCM}	Viscosity of PCM	Pas
V	Separation speed	Kg/ m ² s
AR	Aspect Ratio	-
G	Gibb's potential	J
ρ	Density	Kg/m ³
Т	Temperature	°C
Ср	Specific heat capacity	kJ/Kg K
m	Mass flow rate	kg/s
k	Thermal conductivity	W/mK
Х	Mass fraction	Wt%
V	Velocity	m/s
Р	Pressure	Pa
Р	Turbulent Prandtl number	-
Y	Dissipation	J/kgs
α	Volume fraction	Vol%
h(T)	Enthalpy at temperature T	J

η_{eff}	Viscosity	Pas
F	Body force	Ν
E	Sensible enthalpy	J
γ	Shear rate	s^{-1}
ω	Mass concentration of surfactant	%
h	Heat transfer coefficient	W/m^2K

Building

Variable	Description	Units
ω	Humidity ratio	%
cap _{misoture}	Moisture capacitance	kg
cap	Building thermal capacitance	Wh/m ² K
Q_{cooling}	Cooling load	W

Chiller

Variable	Description	Units
COP _{nom}	Chiller nominal COP	-
COP _{rated}	Chiller rated COP	-
Q _{load}	Chiller load	W
Р	Power	J
FFLP	Fraction of full load power	%
COP _{ratio}	Ratio of chiller nominal COP to rated COP	-
Capacity	Chiller capacity	W
PLR	Partial load ratio	-
Qrejected	Energy rejected to cooling medium fluid	W
Q _{met}	Load met by chiller	W

Heat exchanger performance

Variable	Description	Units
U	Overall heat transfer coefficient	W/m ² K
А	Heat transfer area	m^2
F	LMTD correction factor	-
LMTD	Logarithmic Mean Temperature Difference	-
Q _{total}	Total heat transferred	W
3	Effectiveness	-
С	Heat capacity rate	J/gK
t	Time	S
Q _{max}	Maximum possible heat transfer in the heat exchanger	W
Q _{FCU}	Cooling capacity of fan coil unit	W
COP	Coefficient of Performance	-
a	Constant	-

Pump or fan performance

Variable	Description	Units
ψ	Ratio of flow rate to designed flow rate	%
Δp	Pressure drop	Pa
η	Efficiency	%
η_{motor}	Motor efficiency	%
Qfluid	Heat transfer to fluid	W
$\mathbf{f}_{motorloss}$	Fraction of fan motor inefficiencies	%
\mathbf{P}_{shaft}	Shaft power	J
W	Work	J

Subscripts

Variable	Description			
W	Water			
8	Surfactant			
vent	Ventilation air			
chw	Chilled water			
set	Set point			
0	Outlet			
h	Hot side fluid			
design	Design condition			
max	Maximum value			
р	Phase p			
m	Mixture m			
PCM	РСМ			
intial	Initial state			
cw	Cooling water			
i	Inlet			
c	Cold side fluid			
air	Air stream			
sat	Saturated air condition			
min	Minimum value			
fluid	Fluid flow			
fan	Fan			

CHAPTER 1

INTRODUCTION

1.0 Research background

The global demand for energy has been steadily rising as a result of continuing population growth and economic developments. For example, in 2013, the world consumed 12700 million tons of oil equivalents (Mtoe) of energy, a 2.3% increase from 2012 [1]. According to the recent energy outlook from BP [2], world energy use will grow by 41% by the end of 2035, with 95% of that increase projected to take place in the rapidly-growing emerging economies led by China and India.

Energy is principally consumed by three end use sectors, i.e. building, industry and transportation with buildings representing over 40% of final global energy consumption [3]. In considering a developed country like the UK, over 40% of total energy consumption is building related which in turn produces about half of the nation's CO2 emissions [4]. In the United States of America (U.S.A), the building sector also accounts for about 40% of primary energy consumption [5].

Meanwhile, developing countries are also contributing to the high level of energy consumption. For instance, China has become the world's top energy consumer since 2009 and by 2040 China's energy use is expected to be double that of the U.S.A [6]. China alone was responsible for 22.4% of global energy consumption in 2013 [1] with the building sector contributing about 30% of the total consumption [7].

Heating, Ventilation and Air Conditioning (HVAC) system is one of the largest energy end users both in buildings, which is responsible for 50% of residential buildings' energy consumption, and in commercial buildings, such as office buildings, retail stores, warehouses, schools, hotels, etc, the figure can be as high as 60% [8]. Currently, there is an increasing demand for cooling which is expected to increase even further in the years to come due to climate changes. For cooling applications, chilled-water systems are used widely in large-scale air-conditioning projects around the world. According to a recent report [9], global market scale of chilled-water system in 2013 was US\$ 8.5 billion, accounting for 9.3% of the world market value. Chilled-water system accounted for 59% of the market share in Europe and also dominated in Chinese HVAC sector by occupying 38% of the market share in 2013. Chilled water circulating pumps is a major energy consumer which is responsible for about 15-30% of system's energy consumption [8].

Past investigations have identified phase change slurries (PCSs) as potential working fluids which could be used to reduce energy consumption in HVAC systems [10, 11]. PCSs basically contain water and phase change materials (PCMs) such as paraffin. PCSs use the latent heat capacity of paraffin as well as sensible heat capacity of water and that of the PCM to store thermal energy. PCSs do not lose the fluidity during phase transition due to the carrier fluid. Hence, PCSs can be directly used both in storage systems and in pumped systems like cold supply network. Integrated PCS/air conditioning systems could take advantage of their high heat capacities to reduce flow rate and thus save pumping power whilst delivering the same amount of cooling effect. PCSs can also simultaneously act as cold storage materials to shift peak-load to off-peak time and improve the COP of systems. The commonly produced type of PCSs is phase change emulsion (PCME) which has attracted interests from a number of researchers over the past decade [12]. PCME is a colloidal system in which

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paraffin is directly dispersed in water by a surfactant. Without additional coating or supporting materials, PCME is cheaper to produce and would have lower viscosity as well as the thermal resistance compared with microencapsulated phase change slurry (MPCS).

However, most of the commercially available PCMEs do experience high levels of sub-cooling temperatures which tend to affect the performance of refrigeration equipment. For instance several studies on sub-cooling in PCME have been published including Huang et al. [13] who reported sub-cooling of 4.3°C and 14.7°C in tetradecane/water and hexadecane/water emulsions respectively. Royon and Guiffant [14] also observed a 5.6°C of sub-cooling in aliphatic hydrocarbons-water emulsion.

Another barrier affecting the application of PCMEs is the difficulty in maintaining stability in emulsions without experiencing temperature stratification during a long-term storage period and when subjected to mechanical-thermal loadings. Typically, air conditioning systems operate for a period of six months each year [15]. Therefore the PCMEs should be able to remain stable for at least six months. Moreover, in the actual air conditioning system the working fluid is typically exposed to strong mechanical stresses while undergoing freeze-thaw cycles, which may lead to coalescence that are otherwise stable under static conditions or under pressure variations of lower magnitude [16]. However, most of the currently available PCMEs fail to meet these requirements. For example, tetradecane and water emulsion made by Xu et al. [17] was stable for only 40-50s. Schalbart et al. [18] tested tetradecane and water emulsion and achieved a short period of storage

Chapter 1

stability at room temperature. The PCME developed by Huang [<u>19</u>] experienced a short storage period of instability after just one month and 100 melting/freezing thermal cycling. Vilasau et al. [<u>20</u>] also tested the freeze-thaw stability of a paraffin emulsion but achieved stability for only 5 cycles.

Above all, information on flow behaviour and heat transfer characteristics of PCME in heat exchangers are crucial for its application in air conditioning systems but currently, they cannot be readily deduced from commercial or published data. This is due to the fact that most of published data cover straight pipes and not coiled pipes. Even though some technical data on various types of heat exchangers (e.g. air-emulsion direct-contact heat exchanger, double-coiled heat exchanger) are available, there is limited information regarding fin-and-tube heat exchangers which are the types commonly used in air conditioning systems.

1.1 Aim and objectives

The main aim of the thesis was to develop a novel phase change material emulsion (PCME) capable of reducing energy consumption in air conditioning systems. In this regard the specific objectives were to:

- 1) Review relevant literature on PCMEs in order to establish their physical/chemical limitations regarding their applications.
- Investigate different PCMEs which possess suitable properties for cooling applications including large energy storage capacity and negligible subcooling.

- Assess the heat transfer and pressure drop characteristic of PCME in finand-tube heat exchangers through three dimensional computational fluid dynamics analysis.
- Experimentally investigate the flow and heat transfer performance of PCME in heat exchangers in a dedicated test rig.
- 5) Investigate the energy saving potential of a PCME/air conditioning system using transient simulation program

1.2 Structure of thesis

This thesis is composed of seven chapters. Chapter 2 presents a comprehensive review of phase change emulsions and their applications in air conditioning systems. It also looks at the preparation method, thermophysical properties and the application of PCME in air conditioning systems.

In Chapter 3, the development procedure of the novel PCME is explored in details. Its thermophysical properties such as heat of fusion, viscosity and sub-cooling temperature are presented. The chemical stability during both storage and discharge periods also covered.

Chapter 4 focuses on theoretical studies of thermal performance of PCME. This includes heat transfer and flow behaviour in fin-and-tube heat exchangers.

In Chapter 5, the flow behaviour and heat transfer characteristic of PCME in heat exchangers are experimentally investigated and the results will be used to validate the simulation results from Chapter 4. In Chapter 6, the performance of integrated PCME/air conditioning system will be evaluated using simulation software TRNSYS.

Chapter 7 concludes the research work by describing the outcomes of the study, and provides recommendations for further work in this field.

CHAPTER 2

REVIEW OF PHASE CHANGE EMULSIONS (PCMES) AND THEIR APPLICATIONS IN HVAC SYSTEMS

2.0 General

Phase change material emulsions (PCMEs) are multifunctional fluids consisting of Phase Change Materials (PCMs) and carrier fluids. PCMEs could be potential candidates as heat transfer media in heating, ventilation and air conditioning (HVAC) systems. This is mainly because PCME could take advantage of its high heat capacity to reduce flow rate and thus saving pumping power whilst delivering the same amount of cooling effect. PCME can also simultaneously act as cold storage to shift peak-load to off-peak time and improve the COP of systems. However, the optimum design of the integrated system requires a good understanding of flow behaviour and heat transfer characteristics of PCMEs. This chapter therefore covers a comprehensive review of the thermo-physical properties and potential applications of PCMEs as alternative heat transfer fluids in air conditioning systems

2.1 Classification of PCME

PCMEs are mixtures of two immiscible liquids and based on the sizes of droplets, they may be grouped into three main categories as follows: a micro-emulsion type, which is characterized by a thermodynamically stable behaviour with droplet diameters ranging from 10 to 100nm, mini-emulsion and macro-emulsion systems, which are both thermo-dynamically unstable with droplet sizes ranging from 100 to 1000nm and over 1µm respectively [21]. The research of PCME has been focused on macro-emulsions, but currently nano-emulsions have gained considerable attention because of their better long term stability and lower viscosity.

Depending on the type of dispersed phase and carrier liquid, oil-in-water (O/W) direct emulsion or water-in-oil (W/O) inverse emulsion can be formed but more complex systems such as oil-in-oil (O/O) or multiple emulsions of different kinds (W/O/W, O/W/O, W/O/O) can also be obtained as shown in Fig. 2-1 [22]. W/O emulsions have higher viscosities but lower conductivities than O/W emulsions [23, 24]. Since PCMEs are to be circulated within air conditioning systems, lower viscosity is of vital importance to ensure a low pump power. Therefore, O/W emulsions are more favourable for application in air conditioning systems. The type of emulsion also depends on the method of preparation, properties of the stabilizing additive, and other various factors. When oil and water are emulsified with stabilizing additive, it is difficult to tell whether O/W or W/O emulsion is formed. In general, it is difficult to identify a particular type of emulsion by physical appearance especially when two types of emulsions are quite similar. The following five basic tests are therefore normally used in determining the type of emulsion [24, 25]. There is no order of priority in these tests but at least two tests are required to achieve a reasonable level of accuracy and conclusion.

- Dilution test: The dilution test is based on the solubility of the continuous phase. O/W emulsions can be diluted with water and W/O emulsions with oil. If the emulsion is O/W type and it is diluted with water, it will remain stable, but if it is diluted with oil, the emulsion will break as oil and water are not miscible with each other.
- 2) Dye-solubility: In the dye-solubility test, either a water-soluble or oil-soluble dye can be used and then observe the colour pattern of emulsion under a microscope. Oil-soluble dye will dissolve and dye the dispersed phase in an O/W emulsion, and the continuous phase in a W/O emulsion.
- Filter paper: The filter paper test involves impregnating filter paper with Carbonyl Chloride (CoCl2). O/W emulsion will then change the colour of the filter paper from blue to pink.

- 4) Fluorescence: The fluorescence test is based on the idea that some oils are fluorescent under UV light. When using UV light to test the emulsions, O/W type will show a potty pattern, on the other hand, the main body of W/O emulsion will be fluorescent.
- 5) Conductivity: The conductivity test needs a light bulb to be attached to two electrodes. If the electrode is immersed into an O/W emulsion, the bulb glows as the O/W emulsions have a very high specific conductance.



Figure 2-1: Structure of oil-in-water, water-in-oil and water-in-oil-in-water emulsions [22]

2.2 Components of PCME

2.2.1 Paraffin

Paraffin wax consists of a mixture of mostly straight chain n-alkanes CH3-(CH2) $_n$ -CH3. The crystallization of (CH₃)- chain releases a large amount of latent heat. Both the melting point and latent heat of fusion increase with chain length. Paraffins are safe, reliable and non-corrosive. They are chemically inert and stable, and show little volume changes on melting [26]. Various types of paraffin are available for HVAC applications. For instance, when a paraffin is used for comfort cooling application, it should melt within the temperature range of 0 and 20°C [27]. There are three types of paraffin that have melting points fit this range: tetradecane CH₃-(CH₂)₁₂-CH₃ (T_m=5.9°C, h_f=227kJ/kg), pentadecane CH₃-

 $(CH_2)_{13}$ -CH₃ (T_m=10°C, h_f=206kJ/kg) and hexadecane CH₃-(CH₂)₁₄-CH₃ (T_m=18°C, h_f=236kJ/kg). Due to cost considerations, commercial products with polymers blends such as RT6, RT10 and RT18 from Rubitherm Technologies GmbH, which have similar properties as pure paraffins, are preferred. Despite of the favourable characteristics of paraffin, they have some undesirable properties such as low thermal conductivity and flammability. All these undesirable effects can be minimized by modifying the wax and the storage unit [26].

Higher concentration of paraffin tends to increase the heat capacity of the emulsion. But, increasing concentration level has little or no effect on the size of dispersed droplets. For instance, as shown in Fig. 2-2, Huang [19] tested different percentages of RT10 emulsion (15, 30, 50 and 75wt%) and found that their latent heat capacities increased proportionally. However all the samples covered a similar sizes range of 1–10 mm and had a mean diameter d_{50} of 3–5 mm, which suggested that the paraffin fraction had no significant effect on the droplets size distribution.



Figure 2-2: Relationships between heat of fusion and droplet distribution with different concentration of emulsion [19]

2.2.2 Surfactants

Surfactants are indispensable components used for stabilizing oil and water based emulsions. Surfactants are usually organic molecules consist of two parts: a water-attracting (hydrophilic) portion and a water-resistant (hydrophobic) portion. As shown in Fig. 2-3 [19], they accumulate at the interface of oil and water, with the hydrophilic group pointing towards water, and the hydrophobic group towards oil. The two main functions of surfactants during emulsification process are:

- 1) To reduce interfacial tension between oil and water molecules. Low interfacial tension requires, less energy in producing emulsions [28].
- 2) To form a protective layer around oil droplets in order to prevent coalescence (i.e. the process through which two or more droplets melt to form a bigger droplet and ultimately results in complete breakdown of emulsions). The protective layer also prevents the deposition of paraffin particles in cooling systems and thus reduces the risk of obstruction.



Figure 2-3: Illustration of PCME and surfactant [19]

Surfactants can be classified into different groups as shown in Fig. 2-4: ionic, nonionic, amphoteric surfactant and polymer surfactant [29-31].

1) There are two types of ionic surfactants: anionic surfactants and cationic

surfactant. Anionic surfactants possess a negative charge on their hydrophilic end, while cationic surfactants possess a positive charge. Ionic surfactants have the disadvantage of reacting with other ions thus causing precipitation or foaming in solutions.

- 2) Unlike ionic surfactant, nonionic surfactants do not have any positive or negative charge on hydrophilic end and are the most effective surfactants for stabilizing O/W and W/O emulsions [30, 32]. They do not produce ions in aqueous solution and therefore are compatible with other types of surfactants for complex mixtures. Nonionic surfactants can be used with high salinity or hard water as they are much less sensitive to electrolytes than ionic surfactants. They can also be used in strong acid solutions and have relatively low toxicity to plants.
- Amphoteric surfactants contain both positive and negative charge elements. They are generally quite expensive and thus only used in specialized cosmetics.
- 4) Polymer surfactant is a new class of surface active substance introduced in past two decades. They are formed from the association of one or more macromolecular structures exhibiting water-attracting (hydrophilic) and oil-attracting (lipophilic) characters.

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Figure 2-4: Surfactant classification based on [29-31]

The selection of appropriate surfactants is critical to their application in PCMEs. The Hydrophile-Lipophile Balance (HLB) system is the most frequently used method for selection of suitable surfactant [33]. The principle of this method is to assign a number to ingredients need to be emulsified, and then to select appropriate surfactant or blend of surfactants with the same number. The HLB number of a surfactant is related to its balance between the hydrophilic and lipophilic parts of the surfactant, which is either determined experimentally [33] or calculated from their chemical formulae [34]. A surfactant with a low HLB number is referred to as lipophile while the type with a high number is classified as hydrophilic. The HLB number is important because it tells what types of surfactants are needed for a specific product type. For example, preparing an oil-in-water emulsion requires surfactants exhibiting HLB values between 8 and 18 [35].

The right amount of surfactants in emulsions is also very important to the properties of final products. In the case when surfactant dosage is too low, droplets are likely to coalesce with their neighbours. Conversely, excessive amount of surfactant can affect the quality and stability of emulsion as it will introduce bubbles which are not easy to remove. Li et al. [36] investigated the effects of surfactant/oil ratio on droplet size of paraffin . As summarised in Tab. 2-1, the increase in the surfactant/oil ratio led to a reduction in droplet size as well as increase in the stability level of the emulsion. Similar conclusion was drawn by Wang [37] who estimated an optimal surfactant/oil ratio of 0.1-0.3 for a stable emulsion.

Surfactant blends and	Drople	et size un	nder diff	erent sur	factant	blends/			
omulsify tomporaturo	liquid paraffin ratio (μm)								
emuisity temperature	1:2	1:3	1:4	1:5	1:6	1:7			
System I @60°C	0.29		0.34	0.67	0.81	0.97			
System II @60°C		0.34	0.53	0.78	0.92	1.21			
System II @45°C			0.41	0.45	0.81	2.69			
System III @60°C			0.33	0.71	0.92	1.15			

 Table 2-1: Effect of surfactant/oil ratio on emulsion droplet size distribution by Li

 [26]

System I: Tween80 (58.8%) +Span80 (41.2%)

System II: Tween80(58.8%)+Span80(41.2%):liquid paraffin=3.8:1

System III: Tween80(26%)+Span80(34%):liquid paraffin=3:2

2.3 Preparation of emulsions

2.3.1 High-energy methods

Preparation of emulsions using high-energy stirring technique involves a mechanical process whereby large droplets are broken into small ones. High energy methods produce coarse droplets around 5–100µm, and further reductions
in the droplet size would require a substantially greater amount of energy input. The most common device is the rotor-stator system which applies shear stress to induce deformation of pre-emulsion droplets and breaking into smaller and uniform sizes, whose principle is shown in Fig. 2-5 [21]. The final diameter of the droplets depends heavily on the applied energy. The viscosity ratio of the dispersed to continuous phase also has limited effect on droplet size.



Figure 2-5: Scheme of principle of emulsification with a rotor-stator device [21]

The dispersion speed which is related to energy input, largely affects the quality of produced emulsion. If the dispersing speed is not high enough, i.e. energy input is insufficient, it would fail to mix the paraffin and the surfactant fully and result in an uneven distribution of droplet size. If the speed is too high, it would have little impact on droplet size, thus wasting energy. High speed dispersion is also more likely to introduce large amount of bubbles which will affect stability of the emulsion [38]. As shown in Tab. 2-2 [39], good levels of storage stabilities were achieved for emulsions when the dispersion speed reached 800rpm, but the fluidity started to deteriorate when the speed exceeded 1000rpm. In addition,

increasing dispersion speed over 800rpm had little effect on the particle size distributions of the emulsions as shown in Fig. 2-6. Therefore, a speed range of 800-1000rpm was considered to be adequate for the complete process [<u>39</u>].

Dispersing speed (rpm)	Stability in storage	Fluidity
200	Worse	Diluted fluid
500	Worse	Semi-fluid
800	Good	Semi-fluid
1000	Good	Viscous emulsion
1500	Good	Pasty emulsion

Table 2-2: Effect of dispersing speed on emulsion stability by [39]



Figure 2-6: Different dispersing speed vs. droplet size distribution (1#,2# and 3# represents dispersing speeds of 800, 1000 and 1500rpm) [39]

It is recommended to reduce the dispersing speed after an intense mixing to allow surfactant to be fully absorbed onto the paraffin droplets [<u>37</u>]. In industrial applications, low speed stirring around 200rpm is applied to pre-mix the paraffin and water in order to avoid slopping out before increasing the speed to 500-1000rpm to fully emulsify the paraffin. Dispersion for a longer period of time than

30 minutes is not effective and is considered to be a waste of energy as 30 minutes would be enough to completely emulsify a product and longer emulsification time has little influence on the droplet size [37].

2.3.2 Low-energy method

PCME can also be prepared using low energy methods called spontaneous emulsification. Spontaneous emulsification makes use of the chemical energy stored in molecules to complete the transition between O/W emulsions and W/O emulsions without external energy input [28].

Phase inversion method is the most often considered approach of spontaneous emulsification method. Emulsion prepared by phase inversion tends to be fine and well-dispersed [40]. Phase inversion makes use of changing the spontaneous curvature (molecular geometry) or relative solubility of nonionic surfactants in water and oil. Due to the hydrophobic effect, surfactant molecules tend to spontaneously associate with each other in water to form a monolayer which allows the most efficient packaging of the molecules. In an emulsion, the surfactant monolayer is presented in an oil-water interface. The type of emulsion that a particular surfactant tends to favour depends on surfactants' spontaneous curvature change of surfactant during phase inversion and the resulting type of emulsion. Phase inversion can be achieved either by changing the temperature (phase inversion temperature, PIT) [41, 42] or by changing the volume fractions of water/oil (emulsion inversion point, EIP) [43-47].

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Figure 2-7: Phase inversion (a) surfactant morphology, (b) type of assembly, (c) emulsion type [28]

The phase inversion temperature (PIT) method (vertical arrow in Fig. 2-8) was first described by Shinoda and Saito [41, 48] as an alternative to high shear emulsification methods. The essential processes of PIT method are shown in Fig. 2-9, in which oil, water and nonionic surfactants are pre-mixed together at room temperature [49]. At low temperatures, the surfactant tends to be more soluble in water and the surfactant monolayer exhibits positive curvature which packs oil droplet in middle. When this emulsion is heated gradually, the surfactant gradually becomes lipophilic (soluble in oil) and at a particular PIT, the solubility of the surfactant in the oil and water phases is approximately the same, and surfactant monolayer then exhibits zero curvature. At temperatures higher than PIT, the surfactant gets completely dissolved in the oil phase and the original O/W macro-emulsions becomes the W/O emulsion. Mechanical power is applied at a constant temperature above PIT temperature to break large drops. When this system is

cooled below the PIT, surfactant molecules move from the oil phase into water phase rapidly which leads to the spontaneous formation of small oil droplets.



Figure 2-8: Schematic illustration of phase inversion for preparation of O/W emulsions [50]



Figure 2-9: Schematic diagram of the formation of emulsions by the PIT method $[\underline{49}]$

The change from one type of an emulsion to another is through a catastrophic phase inversion (CPI) also shown in Fig. 2-8. In this case a W/O emulsion with a high oil content is first formed with some specified types of surfactants before the water is gradually added at a continuous stirring rate (Fig. 2-10). Above critical water content, the emulsion reaches the phase inversion point where it changes into an O/W system [49]. The size of the droplets depends on various process variables, such as the dispersing speed, the surfactant concentration and the rate of water addition, etc. The surfactant used in EIP method is limited to small molecule surfactants to stabilize both W/O and O/W emulsions.



Figure 2-10: Schematic diagram of the formation of emulsions by the EIP method [49]

Adding water to surfactant and oil mixture followed by inversion is indeed a widely used method. As both of the lipophilic and hydrophilic surfactants are used, this method is sometimes modified as in the mixing film synthesis [45, 46]. As mentioned before, the inversion can take place in a catastrophic or transitional way. The catastrophic inversion is due to a change in WOR, and the transitional one is because of a change in formulation, which can be done by varying the temperature, or changing the weight composition of the surfactants in the emulsion as in the mixing film synthesis. In this method, the lipophilic surfactant (e.g. Span60) is pre-mixed with the oil (called mixture A). Therefore, the PIT temperature is very low. The hydrophilic surfactant (e.g. Tween60) is mixed with water forming mixture B. Mixture B is then added to mixture A at a specified temperature. This process increases both the concentration of hydrophilic surfactant and the PIT. With this method, it is possible to trigger a transitional inversion at constant temperature [18].

Schalbart et al. [<u>18</u>] used three different low energy methods (EIP, mixing film and PIT) to prepare tetradecane emulsions and found the PIT to be the most effective approach. During the preparation, all the components which were premixed at room temperature were heated to 72°C, and then were cooled down to room temperature by adding 54g cool water. In this way, the emulsion crossed the inversion line twice which achieved a narrow droplet size distribution in the range of 200-250nm. They maintained stability with low viscosity values (2-4 times of water) for more than six months at room storage temperature.

2.4 Main characteristics of PCME

2.4.1 Thermophysical properties: Heat capacity

Total heat capacity of PCME consists of not only the latent heat of the paraffin,

but also the sensible heat capacity of the water and that of the paraffin. In the temperature range from T_1 to T_2 , total heat capacity of PCME Δh_e is the sum of latent heat capacity of the emulsion $\Delta h_{f,e}$, the sensible heat capacity of water Δh_w and that of paraffin Δh_{PCM} as follows [27]:

$$\Delta h_{e} = \Delta h_{f,e} + \Delta h_{w} + \Delta h_{PCM} = X_{PCM} \Delta h_{f,PCM} + X_{W} C_{p,W} (T_{2} - T_{1})$$

$$+ X_{PCM} C_{p,PCM} (T_{2} - T_{1})$$
(2-1)

Where, X_w and X_{PCM} are the weight fractions of water and paraffin, $\Delta h_{f,p}$ is the heat of fusion of paraffin in the temperature range of T_1 to T_2 . $C_{p,w}$ and $C_{p,PCM}$ are the specific heat capacity of water and the average specific heat capacity of the paraffin, respectively.

Total heat capacity of the emulsion can be estimated experimentally. Experiment to determine the heat capacity of the emulsion under certain similar operational conditions in practical applications has been carried out in a test rig (Fig. 2-11) by Huang [27]. The rig consisted of a storage tank, a thermostat, a plate heat exchanger and a heating pump. A flow meter and thermocouples were used to determine the flow rates and the temperatures of the primary heat transfer fluid and the sample. All of the elements and pipes were well insulated to reduce the heat loss. The values from the test rig did agree well with those estimated according to Eq. (2-1).



Figure 2-11: The test rig for studying the heat capacity of the emulsion [27]

2.4.2 Thermophysical properties: Thermal conductivity

Concluding from past investigations [11, 28, 43, 51], PCMEs exhibit substantially higher thermal conductivities than PCMs [52, 53]. The size, shape and components of dispersed phase all have effects on conductivity. Emulsions with small particles have higher thermal conductivity than ones with large particles. Emulsions with spherical shape particles exhibit slightly higher thermal conductivity compared with the ones having irregular particles. In addition, high temperature could enhance the conductivities by increasing the molecular activeness. Although there are some empirical correlations such as the Eurken's model [10] and laws of mixtures [43] available for calculating thermal conductivities of PCME, they are not considered to be sophisticated and robust enough for high level accurate results.

Even though PCMEs have higher conductivities than PCMs, they are still lower than that of water. Therefore, various researchers have tried to enhance the conductivities of PCMEs with different types of materials. For instance, Zou [54] looked at the possibility of increasing the thermal conductivity of paraffin emulsions with 1wt% Al nanoparticles. The results showed about 29.4% enhancement in thermal conductivity which was almost equal to the conductivity of water. Ho [55] prepared several different nanoparticle-in-paraffin emulsions by emulsifying 5wt% and 10wt% of alumina (Al₂O₃) particles in n-octadecane emulsion and achieved corresponding thermal conductivity enhancements of 2% and 6% respectively at 30°C. However, the alumina also contributed to an increase in the dynamic viscosity and was therefore considered to be unsuitable for application in air conditioning systems. Another main disadvantage in using metallic materials is that they displace a considerable amount of PCM and increase the weight of the dispersed phase thus causing sedimentation [10, 55]. As a result some investigations involving the use of lightweight materials such as carbon fibres and nanofibres have been carried out [56-58]. For instance Elgafy and Lafdi [56] employed carbon nanofibres to enhance thermal performance of PCME. As shown in Fig. 2-12, the thermal conductivity at room temperature did increase with increasing mass fraction of carbon nanofibres (CNFs) and the cooling rate during the solidification process also increased significantly. However, the problem of settlement of nanoparticles after freeze-thaw cycles was observed by Fleischer et al. [58] and was attributed to the change in density which occurred during the period of liquid-solid transition.



Figure 2-12: The thermal conductivity at room temperature vs. CNFs content ratio [56]

2.4.3 Sub-cooling

Sub-cooling phenomenon is one of the issues which limit the applications of PCME. It lowers the crystallization temperature of a liquid or a gas, thus latent heat is released at a lower temperature or in a wider temperature range [59]. A large sub-cooling is not acceptable because it would enlarge the operating temperature range of a PCS system and thus deteriorates the system performance and reduces energy efficiency.

The level of sub-cooling for PCMEs highly depends on the PCM used. However, PCMs behave differently from microscopic geometry to macroscopic geometry. Even though the used paraffin may have negligible sub-cooling, the PCME still crystallizes below freezing temperature [<u>11</u>, <u>16</u>, <u>19</u>]. For instance, the tetradecane and hexadecane emulsions prepared by Huang et al. [<u>13</u>] showed temperatures of 4.3°C and14.7°C sub-cooling respectively, while the PCM tetradecane and

hexadecane only have 0.5°C and 1.9°C degree of sub-cooling.

Nucleation theory

Gunther et al. [60] tried to interpret the cause of sub-cooling with nucleation theory which describes the formation of small particles of the new phase in the middle of the mother phase in early stage of phase transitions. When the temperature of bulk liquid is reduced below melting point, the formation of solid phase would be spontaneous and release energy. When a particle of radius r is formed, a phase interface between the bulk liquid and solid agglomerate would be created. The creation of the interface would need energy input to fight against the effect of surface tension force in order to prevent it from reducing the size of the liquid surface. However for very small solid particles, the energy released during phase transition is not enough to overcome the effect of surface tension force. It blocks the nucleation process and prevents spontaneous solidification from happening.

The thermodynamic potential that describes this effect is the Gibb's potential G. The difference in G between the old and the new phases, Δ G, is a function of cluster radius, and its maximum defines the critical radius r_{crit} as sketched in Fig. 2-13 [<u>61</u>]. Δ G(r), called the nucleation barrier, is a potential barrier that needs to be overcome to initialize macroscopic nucleation. This means, only for clusters larger than the critical radius, the radius grows spontaneously and solidification proceeds [<u>60</u>]. The crystallization process can be regarded as a two-step process [<u>62</u>]. The first one corresponds to the apparition within the liquid phase of a "supercritical" aggregate. It is the process to develop a cluster larger than the critical radius. The second phase corresponds to crystalline growth, initiated by this supercritical cluster and leading to the total crystallization of the system.



Figure 2-13: Maximum of $\triangle G$ defines the critical radius beyond which radius growth is possible [61]

The nucleation in bulk PCM is known as heterogeneous nucleation. In bulk PCM, crystallization will occur more rapidly as one nucleation event per PCM module is sufficient to assure the functionality of PCM. However, when the PCM is finely divided into emulsion droplets, the volume of the single droplets is very small; freezing process in emulsions is predominantly a homogeneous nucleation which leads to the lower crystallization temperature of PCME. At some point, the droplet size is smaller than critical radius, the nucleation is blocked. One of the solutions is by using nucleating agents.

Nucleating agent

Substances that have a particularly small phase tension, so-called nucleating agents, can be added to the liquid to enhance nucleation. Nucleating agents in

principle are impurity or surface in contact with the liquid. They offer the necessary seeds inside of the paraffin droplets to start the nucleation and thus are acting as a nucleating catalyst. The nucleating agent and its fraction have important effects both on the melting and nucleation temperature.

In the case of PCME prepared by Huang et al. [13], a nucleation agent was added before emulsifying. Paraffin wax with a higher freezing peak point than PCM was selected as the nucleating agent which, helped to reduce the degree of sub-cooling by 6.9°C.

Effect of droplet size

The smaller the droplets are, the higher degree of sub-cooling the emulsions probably will have [<u>16</u>, <u>63</u>, <u>64</u>]. To analyse the relationship of droplet size on nucleation process and sub-cooling, Huang [<u>13</u>] prepared different phase change emulsions with different droplet sizes. Droplet sizes and their distributions played vital roles in both the melting and nucleation temperature. Both temperatures decreased with the reduction in droplet size. The influence of the droplet diameter on the melting temperature was found smaller than 2°C, but the influence on the nucleation temperature was as high as 15°C for the emulsion with the smallest droplets.

The effect of surfactant

The degree of sub-cooling depends not only on the droplet size but also the type of surfactant. Huang et al. [13] prepared hexadecane emulsions with three different surfactants (SDS, Tween40 and surfactant mixture) and found that

surfactants had no significant impact on the melting temperatures. However, the nucleation and sub-cooling temperatures of emulsions did change with different types of surfactants. The hexadecane emulsion with surfactant mixture had two levels of nucleation temperature of 11.6/6°C, and corresponding sub-cooling degrees of 5.7/11.3°C respectively. The nucleation temperature of sample prepared with SDS surfactant achieved the lowest nucleation temperature of about 2.6°C with sub-cooling temperature of 15°C. For the sample with Tween40, nucleation temperature increased to 6°C and resulted in a smaller sub-cooling temperature of 11°C.

There is extensive evidence to show that adsorption layers of surfactants could trigger nucleation. Surfactant molecules at the interface, acting as a template, increase the ordering of oil molecules to form nuclei of a critical size to initiate crystallization [64, 65]. Experiment by Golemanov et al. [16] revealed that emulsions with surfactants with longer alkyl tails (i.e. with 16 or 18 Carbon atoms) had a lower degree of sub-cooling, and that addition of oil-soluble co-surfactant could further suppress the sub-cooling phenomenon. This finding indicated that the dense adsorption layers of long-chain surfactant molecules could act as templates for oil crystallization.

2.4.4 Stability

Stability is always the priority of PCME studies. For practical uses, PCME should be stable for long-term storage and even under shear rate. Emulsions may degrade via a number of different mechanisms as in Fig. 2-14 [11]:

1) Creaming: the process of forming a concentrated layer at the superior part

of sample as a result of gravity

- Flocculation: the process through which small oil particles stick together and form clumps
- Coalescence: the process through which two or more droplets melt and form a bigger droplet
- Ostwald ripening: the process of forming a bigger droplet due to solubility difference
- 5) Phase inversion: the process through which continuous phase become disperse phase while disperse phase becomes part of continuous phase

In the majority of cases, the predominant degradation mechanism is coalescence. As a result of thermodynamic instability, emulsions will finally tend to increase in droplet diameter. Even though creaming and flocculation do not normally change the droplet size distribution, yet they are the signals to coalescence since coalescence requires the droplets to be close proximity. Ostwald ripening, on the other hand, does not require the droplets to be close, since the ripening occurs by transporting of dissolved matter through the dispersion medium [<u>66</u>].



Figure 2-14: Instability in PCME [11, 66]

Solutions to Instability

In an ideal suspension, instability should never occur. Unfortunately, majority of PCMEs tend to separate, especially under thermal-mechanical cycles. For practical applications, it is acceptable that the emulsion changes the properties in a tolerable range [19]. The PCMEs can be considered as stable when neither distinct growth in droplet sizes nor phase separation occurred during the storage period, and in practical applications, it is tolerable when emulsions remained in dispersion and kept the heat capacity under the designed operational conditions [67]. Schalbart et al. [18] tested tetradecane-in-water emulsions and achieved good storage stabilities against creaming at room temperature for six months. But

Ostwald ripening destabilisation was still observed resulting in a slight increase in the droplet size. Huang et al. [67] evaluated the stability of their PCMEs under different conditions. The droplet size distribution changed slightly after 28-month storage at room temperature. However, slight coalescence was observed in the properties after 100 thermal-mechanical cycles.

The standard procedure for testing the stability of paraffin dispersions is to subject them to shear rate. Fig. 2-15 is a schematic drawing of the test apparatus designed by Vilasau [20] to test the stability of emulsion under shear stress. This stability test rig consists of a centrifugal pump connected to a column with a free fall. The stability under freeze-thaw cycles could also be achieved by circulating the dispersion through a model heat-exchanger system. If the dispersion was unstable, a clear separation could be witnessed after a number of cycles and this number would be used as a measure of dispersion stability. However, the circulation method requires special equipment, it is also time-consuming, especially for the more stable systems, and no precise quantification of dispersion stability is possible. Moreover, in the actual air-conditioning systems, the working fluid is typically exposed to strong mechanical stresses, which may lead to coalescence in emulsions that are otherwise stable under static conditions or under pressure variations of lower magnitude [16].



Figure 2-15: Stability test facility of emulsion under shear designed by Vilasau [20]

Zou et al. [11, 46] suggested two principles to overcome the instability.

- Seek an optimal concentration of PCM. Higher concentration will easily cause agglomeration. However, reducing concentration will deteriorate the latent heat capacity of PCME. So, a proper concentration should be studied.
- Reduce the diameter of droplet. The speed of separation can be described using Stockes equation [<u>37</u>]:

$$V = (\Delta \rho \cdot \mathbf{d}^2 \cdot g) / 18 \,\delta_c \tag{2-2}$$

Where V, $\Delta \rho$, g and δ_c represents separation speed, density difference between two phases, gravity acceleration and viscosity of continue phase respectively.

According to the Eq. (2-2), a stable emulsion which means a low separation speed, needs to have a high viscosity, small density difference between two phases

and small droplet diameter. However, for an emulsion, it is difficult to reduce density difference. So, the easiest way is to reduce droplet size which reduces the influence of gravity. Droplet size can be control by both formulation and composition of the surfactant-oil-water system as well as the stirring conditions prevailing.

Factors affecting stability

The stability of PCME is closely related to surfactant. Surfactant forms an absorption layer around oil droplet. If the layer is not strong enough, droplets will cream. Therefore, surfactant selection is the top priority to produce small and even paraffin droplets.

Golemanov et al. [16] tested the effect of surfactant on emulsion stability. They found that solid-liquid phase transition of the dispersed paraffin particles reduced the dispersion stability significantly. Irregularly shaped crystals could significantly destabilize the system during the freezing-thaw cycles. It was pointed out that nonionic surfactants were more effective at stabilizing emulsions than others. The surfactant mixtures comprising hydrophilic and hydrophobic components were found efficient in stabilizing the dispersions, which was attributed to the formation of denser adsorption layers around dispersed phase. HLB number, used to choose surfactants for conventional emulsions, may not be effective for emulsion with particles which undergo freeze-thaw cycles. The chain length of the surfactants was also very important for such systems.

Vilasau [20] analysed the stability of PCME with mixed ionic/nonionic surfactant

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system. Emulsions were prepared as a function of three parameters: 1) ionic surfactant fraction (=ionic surfactant/total surfactant), keeping the total surfactant concentration a constant. 2) The total surfactant concentration, keeping constant ionic fraction, and 3) the homogenization pressure. They found that the most stable emulsions were those with high ionic surfactant ratio. Moreover, stability increased at high surfactant concentration and high pressure which produce emulsions with small sizes.

In the same study, the stability was also assessed against electrolyte concentration, since paraffin emulsions were generally diluted using water containing high concentration of electrolyte for industrial applications. In this study, emulsions were diluted in water (1:1000) and two electrolytes: Sodium Chloride (AnCl) and Calcium Chloride Chloride (CaCl₂) were added. Samples were mixed and then kept at 25°C for 24hrs. Afterwards, critical coagulation concentration (CCC) was used as a measuring parameter of stability. Generally, a higher CCC was preferred. The results indicated that diluted emulsions presented low stability against electrolyte (CCC_{NaCl} and CCC_{CaCl2} were 17 and 4mM, respectively). It was probably caused by the charges presented on the particles. With the Ca^{2+} , stability was much lower than emulsion with NaCl. Fig. 2-16 [20] indicates the relationship between CCC_{CaCl2} and ionic surfactant weight fraction. The emulsions prepared without ionic surfactant ($\Phi I=0$) appear flocculated after emulsification, considering their CCC as zero. Addition of ionic surfactant led to an increase in CCC_{CaCl2} . CCC peaked at ionic surfactant weight fraction of 0.56. It suggested the coexistence of nonionic and ionic surfactant was better at stabilizing emulsions, and nonionic surfactant was required to maximize CCC.



Figure 2-16: Determination of the critical coagulation concentration of $CaCl_2$ (CCC_{CaCl_2}) as a function of the ionic surfactant weight fraction [20]

2.4.5 Rheological behaviour

In emulsion studies, two factors must be considered: droplet size distribution and viscosity. It is of vital importance to control the droplet size distribution of emulsions as viscosity tends to increase as the droplet sizes decrease [63]. Dispersions with fine droplets normally are more stable than those with large droplets. However, smaller droplets lead to a larger sub-cooling degree [37]. Therefore, a compromise on droplet sizes should be reached to ensure an emulsion with low viscosity, small sub-cooling degree and good stability.

Viscosity is an important parameter of emulsions in terms of stability. If viscosity of an emulsion remains unchanged after pumping then it is considered to be kinetically stable. Within air conditioning systems, emulsions would be continually pumped while undergoing heating and cooling cycles. It is therefore important that the viscosity is stable and low enough to ensure low pump energy consumption.

Theoretically, the Einstein equation can be used to describe the viscosity of emulsions [68]:

$$\delta = \delta_{PCM} \left(1 + 2.5\phi \right) \tag{2-3}$$

Where, δ and δ_{PCM} are the viscosities of emulsion and dispersed phase.

For O/W type of PCME,

$$\phi = V_{PCM} / (V_{PCM} + V_W) \tag{2-4}$$

Among which, V_P+V_w is a constant. With the water content reduces, viscosity of emulsion increases.

Huang et al. [19] measured viscosities of the different concentrated levels of RT10 emulsions. They realized that the fraction of the dispersed phase was one of the important parameters that affect the rheological behaviour of an emulsion. They noticed that the samples with 55–75wt% paraffin were much more viscous than ones with 15-50wt% paraffin. A PCME is required to have a low viscosity for applications in air conditioning systems. Thus, the paraffin weight fraction over 50wt% was not recommended. PCMEs containing 30–50wt% paraffin would be favourable alternative to water in energy storage applications because it had an energy storage density two times higher than that of water but still kept a low viscosity.

In 2010, Huang [<u>67</u>] further studied the relationship between temperature and viscosity by testing three types of PCMEs, named as Cryosol 6,10 and 20. As shown in Fig. 2-17, the viscosities were different for these three types of PCMEs

and thus demonstrated the impact of the dispersed material on emulsion viscosity. The studied emulsions were shear-thinning fluids. The viscosities dropped with shear rate increasing when the shear rate was lower than 30s⁻¹ but then maintained at the same level after that. Temperature ramp tests were also carried out aiming at determining the viscosity during the phase transition. The samples were cooled from 30°C to 0°C at a rate of 2°C/min and were measured at a constant shear rate of 50s⁻¹. The result as shown in Fig. 2-18 indicated that temperature had a significant impact on the emulsions due to the phase transition of the paraffin. Sudden jumps in viscosities of emulsions were witnessed at corresponding peak freezing point. The most likely cause was that at the freezing point, PCM got frozen and solid PCM particles deformed little under shear stress and thus caused higher viscosities.



Figure 2-17: Viscosity of Cryosol 6, 10 and 20 versus shear rate at 25°C [67]



Figure 2-18: Viscosity of Cryosol 6, 10 and 20 versus temperature at 50s⁻¹ [67]

Zhao et al. [69] studied the rheological behaviour of 49.8wt% and 16.3wt% tetradecane emulsions with rotational viscometer. The tested emulsions showed basic characteristics of non-Newtonian fluid and formed the solid-liquid dispersion after solidification. High concentrated emulsions had higher viscosities. Fig. 2-19 shows the viscosity changes against temperature. Change in viscosity appears to be linear with temperatures from phase change point (6°C) to 50°C. The variation with temperature was smaller in high concentrated emulsion, which the author speculates was caused by the water content difference. The viscosity of 49.8wt% PCME was 5-10 times of the viscosity of 16.25wt% emulsion after solidifying. Before solidifying, viscosity of 16.25wt% PCME almost kept constant, while for the 49.8wt% one, the viscosity increased dramatically.



Figure 2-19: Viscosity against temperature phase change emulsions [69]

Dai et al. [23] set up a series of experiments to explore the factors affecting surfactant viscosity. water content, and dispersing speed. The e.g. experiment exhibited that the viscosity reduced with increasing water content, which agrees well with other researchers. The viscosities of emulsions increased remarkably from 0.55 to 2.94mm²/s when the concentration of surfactant was increased from 2.6% to 4.7% and was consistent with the formation of a stronger surfactant monolayer. This is due to the fact that stronger surfactant monolayer requires more energy to move the PCM molecules [70]. The viscosity also doubled when the dispersing speed increased from 400 to 1300rpm.

2.5 Heat transfer of the PCME

In comparison with conventional single phase heat transfer fluid containing no PCM, substantial heat transfer enhancements do occur in phase change emulsions. Investigation by Huang et al. [67] did reveal that the presence of micro-sized

PCM particles in slurries can enlarge the surface-to-volume rate, and therefore increase their heat transfer rates as compared with bulk PCMs [67]. In addition, as PCMEs remain fluidity during phase change process, the convection heat transfer, instead of conduction heat transfer, is the domination process during solidification [43]. The following section focuses on review of heat transfer phenomenon including experimental and numerical studies on both laminar and turbulent flow.

2.5.1 Theoretical studies

In spite of numbers of experimental studies with phase change material emulsions, theoretical models for heat transfer are limited. Chen and Chen [71] used a Dirac δ -function based model to investigate the heat transfer process over a plate at a constant temperature.

Charuyakorn [72] developed a comprehensive model for laminar heat transfer in a circular duct and between parallel plates. The model included the enhancement of thermal conductivity due to particle/fluid interactions. Heat gain due to phase change process was also involved in the energy equation as a source term. However, the experimental result of Geol [73] agreed little with their numerical prediction.

Zhang [74] modified the Charuyakorn's [72] model. This model was validated with the result of Geol [73]. This analysis pointed out the biggest reason causing the difference between Geol and Charuyakorn was the effect of sub-cooling. However, the authors did not provide any correlation or similar criteria for future design.

Alisetti et al. [75] introduced an "effective heat capacity model" for heat transfer in PCM slurries. In the effective heat capacity method, the phase change effects are directly incorporated into the energy equation by assuming the specific heat capacity of the phase change material to be a function of the temperature. The model overcomes the limitations of previous models [71, 72, 74] which are using complicated source terms or special analytical techniques. This model is easier to apply in standard computer fluid dynamic (CFD) packages without complicated terms or special analytical techniques [76]. Roy et al. [77] used effective heat capacity model to predict the forced heat transfer process for fully developed laminar flows. The model was verified by comparing its numerical predictions with previous theoretical results as well as well as experimental data. The numerical study showed that the effect of sub-cooling would be quite significant at very low heat fluxes or when the inlet temperature was much lower than the melting point.

Roy and Avanic [78] adopted the "effective heat capacity model" to simulate the turbulent heat transfer of PCME in a circular tube with constant wall heat flux. The Stefan number was found to be primary parameter influencing heat transfer. For low to moderate Stefan numbers, wall temperature reduced considerably over distance. The locations where phase change effects were dominated could be estimated from numerical results. Further experimental studies with DSC tests were suggested to characterize PCMs to accurately calibrate the numerical models.

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2.5.2 Experimental studies

Roy and Avanic [79] presented an experimental study of laminar forced convection heat transfer in a circular duct with a phase change material emulsion (n-octadecane/water emulsion). The bulk Stefan number considered was up to 3.0 and the concentration of phase change material up to 30vol%. The experimental results were used to further verify previous theoretical models [72, 74] and to evaluate any differences in the heat transfer process between an emulsion type suspension and microencapsulated phase change material suspension as in Geol [73]. The result suggested the heat transfer characteristics for PCMEs were similar to those of microencapsulated phase change slurries. It meant the capsules of MPCM had little effect on heat transfer. The results also showed that the overall heat transfer characteristics did not change drastically for different concentrations. From an applications point of view, the results confirmed that the sub-cooling of the fluid and its melting temperature range were important issues and must be considered in future experiments.

Zhao [69] carried out an investigation on non-Newtonian heat transfer performance for a new O/W type emulsion in a circular tube. From Fig. 2-20, two different concentrations of PCMEs (13 and 40wt%) had the similar heat transfer characteristics. At the same flow velocity, Reynolds number for 40wt% was lower than that for 13wt% emulsion, but the Nusselt numbers for two emulsions were at the same level. After phase change, the emulsions became solid-liquid two phase flows. Nusselt numbers for both emulsions were slightly increased compared to the results when PCMs were in pure liquid phase. It could be concluded that convective heat transfer performances were enhanced by phase change, but they were not affected by PCM concentration.



Figure 2-20: Convective heat transfer characteristic of Phase Change Emulsion [69]

Zhao et al. [80] further tested the laminar flow heat transfer in coiled tubes and gave the dimensionless equation for convection heat transfer. Fig. 2-21 demonstrates the heat transfer characteristic for different concentrated emulsions. The line in the diagram represents the heat transfer characteristic of water. The result showed that the Nusselt numbers of different concentrated emulsions were higher than that of water and increased with the increasing concentration. For same concentration, the Nusselt number was higher when solid phase presented. It was probably due to the conductivity of solid PCM was higher compared to liquid phase and the presence and perturbation action of solid PCM promoted the heat transfer inside the tube.



Figure 2-21: Heat transfer characteristic of PCME in coiled tubes [80]

Choi et al. [81] investigated the heat transfer process of a hexadecane emulsion in turbulent range through a circular tube. Pressure drop significantly decreased at the point where PCM particles in emulsion melted. Both the local convective heat transfer coefficient and the effective thermal capacity of the PCME varied significantly along the test section. This made it difficult to apply the log-mean-temperature-difference (LMTD) method for the analysis of heat transfer in the PCM slurry. The investigators therefore proposed a three-region melting model. Using this three-region melting model, the bulk mean temperature profile was determined along the heating test section as shown in Fig. 2-22. In Region I, the average emulsion temperature was below melting temperature, the bulk mean temperature, increased linearly along flow direction. In the second region (Region II), phase change happened and the bulk mean temperature was constant at the

melting temperature. In the final third region (Region III), PCM was completely melted and the mean temperature increased linearly again. Local convective heat transfer coefficient was found increasing in Region I, and reducing in Region II, and increasing again in Region III. Based on the model, the authors provided an explanation of the mechanism of the convective heat transfer enhancement due to the PCM particles. Along the flow direction, the emulsion temperature increased. As a result, the viscosity dropped and Reynolds number increased, leading to a slight increment in the heat transfer coefficient. They also pointed out that the layer of melted PCM close to wall became thicker when the heat flux was very high. This phenomenon prevented the radial movement of solid PCM from the centre to the wall and resulted in a lower convection coefficient.



Figure 2-22: Measured local heat transfer coefficient and bulk mean temperature

[<u>81</u>]

Choi and Cho [82] studied the influence of the aspect ratio (AR, height/width ratio) of a rectangular channel on the cooling characteristics of C₂₂H₄₆ emulsion flow with a linear array of discrete square heat sources with high heat flux. Other factors such as concentration, heat flux and Reynolds number were also investigated. The result is shown in Fig. 2-23. At a high heat flux of 40W/cm2, the influence of the aspect ratio of the rectangular channel on the local heat transfer coefficients, for both water and the 5wt% paraffin slurry, was greater than those at a low heat flux. Local heat transfer coefficient for 5wt% PCME was larger than that for water. The thermal enhancement was the result of both the particle migration and subsequent collision against the wall in turbulent flows and the latent heat of the PCM. The local heat transfer coefficients for a rectangular channel with an aspect ratio of 0.10 were larger than those for the rectangular channels with aspect ratios of 0.20 and 0.50. The reason was that the average flow velocity for a rectangular channel with an aspect ratio of 0.10 was the largest among three channels at the same Reynolds number because of the smallest hydraulic diameter. The differences of the local heat transfer coefficients between the PCME and water increased as the aspect ratio reduced.



Figure 2-23: Local heat transfer coefficients with respect to row number @ Re=15000 [82]

Inaba et al. [83, 84] dealt with the natural convection heat transfer characteristics of PCME in rectangular enclosures. The PCM mass concentration of the emulsion varied from 30wt% to 5wt%, and the experiments have been done three times T_1 <313K when PCM was in solid phase, 313K< T_2 <323K when PCM was in the phase changing region and T_3 >323K when PCM was in liquid phase. The result showed that the PCM mass concentration had little impact on the Nusselt number when PCM was in solid phase. Fig. 2-24 shows the Nusselt number against Rayleigh number at various mass concentration and enclosures [83]. In the phase transition range, the Nusselt number increased with increasing PCM concentration at low Rayleigh numbers, but at high Rayleigh numbers (>10⁵) the Nusselt

number decreased with PCM concentration increasing. The effect of aspect ratio on heat transfer was greater for the PCM in the solid region than the effect for PCM in liquid region because the viscosity of the PCM slurry reduced when PCM particles were melted. There was little difference in natural heat transfer characteristics of the PCME at concentration slower than 10wt%. Fig. 2-25 shows the convection heat transfer coefficient versus temperature difference between the heating and cooling plates TH-TC (TC=313K) for different PCM concentrations in the enclosure of AR=21: 8. As indicated in Fig. 2-25, the overall heat transfer coefficient declined with the increasing PCM concentration, because the viscosity of PCME increased along with PCM concentration increasing. The heat transfer coefficient peaked at $T_H-T_C=6K$ (or $T_H=319K$) which corresponded to the temperature with the highest apparent specific heat of PCM. It was mainly due to the involvement of the phase transition process of PCM. The heat transfer coefficient then dropped dramatically from temperature difference from 6K to 10K, since the latent heat transferred reduced with heating plate temperature $T_{\rm H}$ increasing over 320K.



Figure 2-24: Nusselt number against Rayleigh number at various mass concentration and enclosures [83]



Figure 2-25: Heat transfer coefficient vs. temperature difference between two horizontal boundaries for PCME with various mass concentrations for AR=21.8 and PCM in phase changing region [83]

2.6 Applications in HVAC system

Currently, most PCMEs are in the development stages and therefore there are no adequate publications regarding their application at either commercial or industrial level available. It is however widely agreed that PCMEs have great potential as cold storage material and heat transfer fluid for heating and cooling systems.

Huang et al. [<u>67</u>] outlined some potential applications based on different phase transition temperatures and operating systems. Conventional cold distribution networks for an air conditioning system are working with cold water within a temperature range of $6-12^{\circ}$ C. It is possible to use a PCME with a melting
temperature between 0 and 6°C, such as a tetradecane/water emulsion, in storages for cold supply networks. The alternative approach is to directly apply PCME, such as pentadecane/water emulsion with melting temperature between 7 and 10°C, in the cooling pipe systems. This would enable the volumetric flow rate to be reduced and ultimately achieve lower energy consumption for the circulating pumps. However, applying viscous PCME in pipelines is likely to cause higher pressure drop than water, which should be considered in practical applications. The idea of utilising PCME instead of water for cooling of ceilings and walls in capillary tubing system was also considered for increasing thermal mass of building components [<u>67</u>]. However, the PCME should have a relatively low viscosity to prevent blockage of the tube.

Pollerberg and Dötsch [85] investigated the possibility of using PCME as a heat transfer medium. The study compared three scenarios for a cooling supply network (based on tetradecane emulsion, ice slurry and water) by considering the cost of cold generation and distribution. They found cold water had the highest distribution cost, followed by tetradecane emulsion and ice slurry. As the evaporation temperature had to be low in order to solidify tetradecane, the COP of the chillers deteriorated and consequently increased the generating cost. In general, the tetradecane emulsion was able to compensate the additional generating cost through reduction in distribution cost, and made it the overall cheapest option under the three scenarios.

Besides the economic aspects, there were technical aspects which were considered by Pollerberg and Dötsch [85]. First of all, the material used in the distribution

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network was based on the type of paraffin. For instance, paraffin is a solvent for rubber, so the seals of EPDM need to be replaced with fluororubber. The second important point is the water endangering potential of paraffin and additives was taken into consideration since they are considered as threats to the safety of potable water. Furthermore, the recycling was suggested, as PCMEs are not considered as hazardous substances.

In a study carried out for California Energy Commission [86], PCME was applied to a domestic air conditioning system as shown in Fig. 2-26. In this study, PCME was used as a cold storage material for peak shifting. The PCM had a phase change range from 10 to 15°C. The system consisted of a blower, heating coil, two heat exchangers for PCM emulsion, an air conditioning unit, a PCME slurry storage tank, pumps and mixer. One of the heat exchangers was used for heating the emulsion (upstream of the AC unit) and the other was for cooling the emulsion (downstream of the AC unit). During off-peak time, the slurry would be pumped through the heat exchanger, and the cool air from air the conditioner unit would remove heat from the emulsion. At peak time, solidified PCME would be passed through the heat exchanger to cool warm air prior to it entering the air cooling coil. Generally speaking, the bench-scale test was successful and the slurry did not clog the pipes. Test results showed that 40% of peak load can be absorbed by PCME system. In addition, the system was able to operate at an efficiency level 10-12 times greater than the existing air conditioning systems during the hottest period of the day.



Figure 2-26: Schematic of air conditioning systems [86]

PCME with melting point over 70°C can be a good candidate as a new category of heat storage material and latent heat transfer fluid in the area of waste heat recovery, boiler and heating system. Ishikawajima Harima Heavy Industries Co, Ltd. [87, 88] proposed to use PCME to recover waste heat from a heat and steam generator of a lithium bromide (LiBr) absorption air conditioning system. The selected phase change temperature range was between 84-91°C which according to their report helped to reduce the spatial and temporal mismatches between waste heat emission and cooling demand. In comparison with water based system, the proposed system could save up to 82% of the energy used for pumping the fluid.

Zou [88] described an electrical boiler which uses PCMEs as shown in Fig. 2-27. It consists of a heat exchanger and a storage tank. Hot water from the boiler heats up the PCME inside the storage tank which in turn heats up secondary working fluid, normally water, to provide heating for users.



Figure 2-27: PCME integrated with electrical boiler [88]

Zou [88] investigated the use of PCME for space heating. The heat transfer fluid for space heating works at 70-95°C. 30wt% tetratetracontane emulsion whose melting point is 85-87°C was therefore selected as heat transfer fluid in this study. The latent heat of fusion of the PCME was taken as 60kJ/kg and the equivalent specific heat around 6.6kJ/kgK, which was 1.5 times of that of water. The results showed that to be able to deliver the same amount of heat, the volume flow rate of PMCE would be equivalent to 0.67 times of that of hot water, which could save 45% of the energy consumed by the pump. Detailed comparisons are listed in Tab. 2-3.

	Hot water	Phase change emulsion
Equivalent specific heat	4 2	6.6
(kg/kgK)		0.0
Volumetric flow rate (%)	100	67
Flow velocity (m/s)	3	2
Viscosity (mPas)	0.4	4.1
Friction loss per unit length (%)	100	83
Pump power consumption (%)	100	55

Table 2-3: Parameter comparisons by [88]

2.7 Concluding remarks

This review focused on the characteristics, challenges and advancement in the development and application of PCM emulsions in HVAC systems. The review has outlined the composition, preparation methods and analysis of properties of various PCM emulsions (PCMEs), which are crucial to the development of PCMEs as heat transfer fluids. After analysis of the works compiled herein, it is concluded appropriate PCME for application in air conditioning systems should have the following desirable properties:

- Phase transformation temperature should be compatible with the designed operating temperature
- High heat capacity and heat transfer rate
- Little sub-cooling and small temperature swing
- Low viscosity
- It should be stable over a long term storage period and be able to withstand reversible freezing/thaw cycles when subjected to thermal mechanical loads in pumping systems
- Non-corrosive to construction materials
- Non-toxic, non-flammable and non-explosive
- No or low hazard to water
- Low cost
- Large -scale availability

It was noticeable from the review that the heat transfer performances of PCMEs were higher than pure water due to their better thermo-physical properties. However, the mechanisms of heat transfer enhancements are not well established due to lack of adequate experimental studies. Several numerical models have also been developed but not fully validated. In HVAC systems, most cooling coil and heating coils are fin-and-tube heat exchangers in which PCMEs are expected to behave differently from straight pipes. The influences of the factors such as frost formation and overall heat transfer of PCMEs on the performance of heat exchangers are also unclear.

Peak load shifting, energy cost saving and pump power reduction have also been reported from limited sources of short term experimental studies. Long term experimental studies are therefore needed to evaluate their overall performances.

Regarding the applications of PCMEs, there are few practical examples at industrial level but yet to become commercially viable. In principle, applications of PCMEs in HVAC systems for either as cooling medium or cold storage material seem feasible. There are, however, some vital problems which need to be overcome before a competitive edge over water can be achieved. These scientific barriers are summarised as follows:

- The issue of higher viscosities in PCMEs needs to be improved in order to minimise pumping power.
- Thermal stability of PCMEs needs to be enhanced in order to prolong period of usage.
- Level of sub-cooling needs to be reduced in order to enhance system performance.
- Sub-cooling, stability and viscosity are related to each other and how to reach a balance among these three factors require more investigations.

CHAPTER 3

DEVELOPMENT OF NOVEL PHASE CHANGE EMULSION

3.0 General

Most of the commercially available phase change emulsions do experience sever sub-cooling problem. For instance several studies on sub-cooling in PCEM have been published including Huang et al. [13] who reported sub-cooling of 4.3°C and 14.7°C in tetradecane/water and hexadecane/water emulsions respectively. Royon and Guiffant [14] also observed a 5.6°C of sub-cooling in aliphatic hydrocarbons-water emulsion.

Another main barrier affecting the application of PCMEs is the thermal instability that they experience after a period of usage and storage. Typically, air conditioning systems operate for a period of six months each year [15]. Therefore the PCMEs should be able to remain stable for at least six months and also not experience any temperature stratification when the air conditioning systems are in operation. Literature review shows that most of the currently available PCMEs fail to meet these requirements. For example, tetradecane and water emulsions made by Xu et al. [17] was stable for only 40-50s. Schalbart et al. [18] tested tetradecane and water emulsion and achieved a short period of storage stability at room temperature. The PCME developed by Huang [19] experienced instability just after a short storage period of one month and 100 melting/freezing thermal cycling. Vilasau et al. [20] also tested the freeze-thaw stability of a paraffin emulsion but achieved stability for only 5 cycles.

This chapter is therefore intended to develop a novel phase change material emulsion that is capable of overcoming these scientific barriers.

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3.1 Selection criteria for PCMEs

As summarized from Chapter 2 appropriate PCME for application in air conditioning systems should have the following desirable properties [<u>16</u>, <u>19</u>, <u>43</u>,

<u>51, 85, 86</u>]:

- Phase transformation temperature should be compatible with the designed operating temperature
- High heat capacity and heat transfer rate
- Little sub-cooling and small temperature swing
- Low viscosity
- It should be stable over a long term storage period and be able to withstand reversible freezing/thaw cycles when subjected to thermal mechanical loads in pumping systems
- Non-corrosive to construction materials
- Non-toxic, non-flammable and non-explosive
- No or low hazard to water
- Low cost
- Large -scale availability

3.2 Selection of emulsion component

3.2.1 Paraffin

According to Huang et al. [19], selection of PCMEs for air conditioning application should be based on properties such as high heat capacity, low subcooling temperature, long term storage stability and low viscosity. Paraffin materials such as tetradecane CH₃-(CH₂)₁₂-CH₃ (T_m=5.8°C, Δh_f =227kJ/kg) and pentadecane CH₃-(CH₂)₁₃-CH₃ (T_m= 9.9°C, Δh_f =206 kJ/kg) are typical pure organic PCMs which may be combined with water to form PCMEs for cooling applications within a phase transition temperature range of 7-12°C. However, due to the relatively high cost of pure PCM, blended paraffin such as RT6 and RT10 (see Tab. 3-1 and Fig. 3-1), are the types which are commercially available for practical application. It should be noted that since the RT series are mixtures of different paraffin their melting/freezing processes do not occur at specific temperatures but rather over a temperature range.

For this development, RT10 was selected as the core material because its melting/freezing temperature falls within the working temperature of air conditioning systems. It also has higher heat of fusion and lower sub-cooling temperature than RT6.

RT6	RT10
0.77	0.77
4-7	4-10
Peak at 6	Peak at 10
6-2	10-4
0	0
175 (±7.5%)	150 (±7.5%)
(-2-13°C)	(2-17°C)
2	2
0.2	0.2
18	22
30	45
Chemical inert	Chemical inert
Water hazard	Water hazard class
class (WGK)1	(WGK)1
	RT6 0.77 4-7 Peak at 6 6-2 0 175 (±7.5%) (-2-13°C) 2 0.2 18 30 Chemical inert Water hazard class (WGK)1

 Table 3-1: Thermophysical properties of RT6 and RT10 from Rubitherm

 Technologies GmbH [89]







Figure 3-1: Enthalpy distribution of RT6 and RT10 sample [89]

3.2.2 Selection of surfactant

Since paraffin and water are immiscible, the emulsion has to be stabilised with surfactant to prevent segregation of different components. The Hydrophile-Lipophile Balance (HLB) method was therefore adopted to determine the appropriate amount of surfactants needed to stabilize the RT10/water emulsion. The principle of this method is to match the specific HLB requirement (RHLB) of the oil phase against the HLB value of the surfactant to obtain an optimum emulsification. HLB values can range from 0 to 20 depending upon the composition of the surfactant. As the HLB value increases, the surfactant becomes more hydrophilic and less lipophilic. In a situation where two or more surfactants are mixed together, the resulting HLB may also be calculated by using the following formula [33, 90]:

$$HLB = \omega_{S,1} \times HLB_{S,1} + \omega_{S,2} \times HLB_{S,2} + \dots$$
(3-1)

Where, $\omega_{s,1}$ and $\omega_{s,2}$ represent the weight fraction of surfactant 1 and surfactant 2.

Determination of HLB requirement

For the required HLB of oil phase, a series of emulsions will be made. Each emulsion will have the same amount of oil phase, water and surfactants but at a different HLB value. Any pair of non-ionic surfactants can be used to determine the appropriate values of HLB but one of them should have a low HLB number (<6) and the other a high HLB number (>14). In this study, two types of surfactants Brij 52 ($C_{56}H_{114}O_{21}$, HLB=5) and Tween 20 ($C_{11}H_{22}O_4$, HLB=16.7) were selected to produce five different emulsions and Eq. (3-1) was used to obtain their corresponding HLB values. The blended samples were then kept in test tubes and observed over a period of time in order to evaluate their chemical stability levels. As summarised in Tab. 3-2, sample No. 3 achieved the longest period of chemical stability. Based on this result, the HLB requirement of the system is closest to 10.9.

Chapter 3

Surfactants				Chemical
Samala Na 🗕			Calculated UI P	separation
Sample No.	Brii 52	Tween 20		period
	Dig 52	Tween 20		(min)
1	100%	0%	5.0	≤5
2	60%	40%	9.7	≤90
3	50%	50%	10.9	>90
4	40%	60%	12.0	≤30
5	0%	100%	16.7	≤5

Table 3-2: Required HLB values

Optimised blend of surfactants

Even though sample No. 3 appears to be the best candidate, it may not be the best combination as their chemical structures can affect the stability of the emulsions. This is due to the fact that surfactants with longer hydrocarbon tails are more effective in stabilizing dispersions than those with shorter carbon chain lengths [16]. For these reasons different types of non-ionic surfactant and an ionic surfactant (HTAB) with mixture HLB values around 10.9 were blended and tested for stability after 48-hour storage period. This was assessed through their separation rates by dividing the sum of the volume of water separated in the test tubes plus that of paraffin ($V_s = V_w + V_p$) by the total sample volume ($V = V_w + V_p + V_e$). This is explained in Fig. 3-2 and expressed mathematically in Eq. (3-2) as:

$$V\% = \frac{V_S}{V} \times 100\% = \frac{V_w + V_p}{V_w + V_p + V_e} \times 100\%$$
(3-2)



Figure 3-2: Emulsion separation

Analysis of Tab 3-3 shows that the blend of Tween 60 ($C_{35}H_{68}O_{10}$, HLB=14.9) and Brij 52 surfactants achieved the highest level of stability without any sign of separation after the monitored period and was attributed to their combined long hydrocarbon tails of 18 and 16 carbon atoms respectively in the alkyl chain. This is further supported by previous research [16] which indicated that long hydrocarbon tail can also act as nucleation centre for activating crystallization and for reducing sub-cooling. To this end, 60% Tween 60 and 40% Brij 52 surfactants were selected for the optimum surfactant blend.

Chapter 3

Sample	RT10	Water	Blended su	rfactant	Calculated	Separation
No.	Concentration	Concentration	Blends of	Wt%	HLB	rate
			surfactant	ratio		
1	25%	72%	Brij35/	50.50	11.0	60%
1	2370	/2/0	Brij 52	50.50	11.0	0070
2	250/	720/	HTAB/		10.0	(00/
2	25% /2%	Brij 52	55:45	10.9	00%	
2	250/	720/	Tween60/	60.40	10.0	0
3	23%0	12%	Brij52	00:40	10.9	0
4	250/	720/	Tween20/	50.50	10.0	200/
4	4 25% /2%	Brij52	50:50	10.9	20%	
F	250/	720/	Brij58/	50.50	10.5	C00/
5	25%	/2%	Brij52	50:50	10.5	30%

Table 3-3: Optimised surfactant blends

3.3 Determination of component concentration

3.3.1 Determination of Paraffin Concentration

Thermal properties play an important role in applications of PCME systems. It is expected that the novel PCME should have a high heat storage capacity. In the temperature range from T_1 to T_2 , total heat storage capacity of PCME Δh_e is the sum of latent heat capacity of the emulsion $\Delta h_{f,e}$, the sensible heat capacity of water Δh_w and that of paraffin Δh_{PCM} as follows [27]:

$$\Delta h_{e} = \Delta h_{f,e} + \Delta h_{W} + \Delta h_{PCM} = X_{PCM} \Delta h_{f,PCM} + X_{W} C_{p,W} (T_{2} - T_{1})$$

$$+ X_{PCM} C_{p,PCM} (T_{2} - T_{1})$$
(3-3)

Where, X_w and X_{PCM} are the weight fractions of paraffin and water, $\Delta h_{f,p}$ is the heat of fusion of paraffin in the temperature range of T_1 to T_2 . $C_{p,w}$ and $C_{p,PCM}$ are the specific heat capacity of water and the average specific heat capacity of the paraffin, respectively.

PCMEs are expected to have at least heat capacity of twice as high as that of water [67]. In the temperature range of 7–12°C, water has a sensible heat capacity of 20.9 kJ/kg , and therefore the PCME should have at least a heat capacity of 40 kJ/kg .The heat capacity of PCME. In the temperature range of 7–12°C against PCM concentration has been plotted in Fig. 3-3. As indicated in Fig. 3-3, the heat capacity of PCME is proportional to concentration of paraffin [19]. In the temperature range of 7–12°C, 25wt% PCME has a total heat capacity of 40 kJ/kg, which is almost 2 times as high as that of water. For paraffin concentration lower than 25wt%, the total heat capacity of the sample would fall short of the above criteria hence the limit for this case.



Figure 3-3: Heat capacity and viscosity against PCM concentration

Appropriate PCME for application in air conditioning systems should also have low viscosity to ensure an acceptable level of pressure drop in pumps. Zhao [91] proposed the following equation for predicting the kinematic viscosity of a two liquid mixture.

$$\gamma = \gamma_{\rm w} e^{k\varphi} \tag{3-4}$$

Where, γ and γ_w are the viscosities of emulsion and water respectively, φ is the volume fraction of oil phase and k is a constant which equals to 7 when $\varphi \leq 0.74$.

The viscosity of PCME against PCM concentration has also been plotted in Fig. 3-3. For PCM concentrations over 35wt%, the viscosity of emulsions increases dramatically and will be 15 times that of water. A PCME is required to have a low viscosity for applications in pump systems. Thus, the paraffin weight fraction should not be more than 35wt% for this case.

Based on the theoretical analysis, different samples of emulsions consisting of 25wt%, 30wt% and 35wt% concentration levels of paraffin were prepared for evaluation as summarised in Tab. 3-4. Analysis of the results shows that the sample containing 25wt% of paraffin is the most appropriate type with the acceptable level of heat of fusion and highest thermal conductivity. It also has the lowest viscosity value and therefore suitable for reducing energy consumption in pumping systems.

Table 3-4: Physical properties of 25-35wt% of K110/ water emulsion					
RT10	Heat of	Heat	Heat storage		
Concentration	fusion	storage	ratio	Viscosity	Conductivity
Concentration		capacity	(PCME-to-	(mPas)	(W/m K)
(wt%)	(kJ/kg)	(KJ/kg)	water)		
25	30.8	42.0	1.99	21.0	0.40
30	31.2	42.2	2.01	30.0	0.34
35	31.5	42.6	2.03	36.0	0.33
0%	-	21.0	-	1.3	0.60

3.3.2 **Determination of Surfactant Concentration**

Regarding the concentration of surfactant, Wang et al. [37] suggested that 10-25% of dispersed phase of surfactant would be effective in ensuring the stability of PCME under either storage or mechanical-thermal cycling condition. However, other investigations revealed that, concentration level should not be higher than 5wt% of total emulsion mixture [92, 93].

In order to establish the appropriate level of concentration, 25% RT10 and water emulsions with different mass fractions of surfactant blend (5-20% of oil phase) were prepared and tested for storage stability for a period of 72 hours. As displayed in Fig. 3-4 and summarised in Tab. 3-5, sample No. 4 achieved the best stability result with no sign of chemical separation whereas samples No. 1-3 suffered from rapid chemical separation of various components before the end of testing period. Sample No. 5 (20% Surfactant/oil phase ratio) produced the worst result with large amount of bubbles during the emulsification process and was therefore considered unsuitable for testing.



Figure 3-4: Tested samples

Table 3-5: Evaluation test results for surfactants						
			Surfactar	nt blends	Chemical	
No	RT10	Water	Tween 60/I	Brij 52(3:2)	separation	Separation
110.	Concentration	Concentration	Concentration	Surfactant/oil	period	rate at 72hr
			of surfactant	ratio	(min)	
1	25%	74.75%	1.25%	5%	<6	60%
2	25%	73%	2%	8%	≤20	45%
3	25%	72.5%	2.5%	10%	≤48	50%
4	25%	71.25%	3.75%	15%	>72	0
5	25%	70%	5%	20%	Not applicable	Not applicable

3.4 Preparation of emulsion

Phase change emulsion was prepared by the so-called "mixing film synthesis". All the constituents were initially weighed by a precision weighing balance with an electronic measuring scale (Fig. 3-5). All the constituents were initially weighted by a precision weighing scale as given in Tab. 3-6. Then, the lipophilic surfactant Brij 52 was dissolved in RT10 (Mixture A) while hydrophilic surfactant Tween 60 was mixed with deionized water (Mixture B). The mixtures were heated separately to about 50- 55°C. The mixture B was added to mixture A and they

were pre-emulsified using a hot plate magnetic stirrer (IKA C-MAG HS IKAMAG as shown in Fig. 3-6) at 500rpm at 50°C for 15 minutes. Final emulsification was carried out by passing the coarse emulsion through a rotor-stator machine (ULTRA-TURRAX T18 as shown in Fig. 3-7) at an emulsifying speed of 7200rpm for another 45 minutes. The developed sample was named as PCE-10 and described in Fig. 3-8 as a white milky Oil-in-Water emulsion.

Components	Wt%	
RT10	25.0	
Tween 60	2.25	
Brij 52	1.5	
Water	71.25	

Table 3-6: Composition of Phase Change Emulsion



Figure 3-5: Measuring scale



Figure 3-6: IKA C-MAG HS IKAMAG hot plate magnetic stirrers



Figure 3-7: Rotor-stator machine



Figure 3-8: Developed PCME sample PCE-10

3.5 Characterisation of developed emulsion (PCE-10)

3.5.1 Density

Bulk density, ρ , is a characteristic of a material and is given by the mass, M, of a powder occupying a known bulk volume, V, as expressed by

$$\rho = \frac{M}{V} \tag{3-5}$$

The mass of emulsion was weighted using an electronic measuring scale and the volume of emulsion was given by cylinder. The density of PCE-10 is:

$$\rho = \frac{6.57g}{7cm3} = 0.94g/cm3 \tag{3-6}$$

3.5.2 pH value

Corrosion in air conditioning systems may cause serious damage to cooling tower, condensers, piping, pumps and other equipment. Corrosion products may reduce heat transfer and clog spray nozzles. Refrigerant needs to be treated to reduce corrosiveness.

Generally, metal corrosion will be accelerated under acidic conditions, i.e. at pH values below 7. Thus, the pH value of PCME should therefore be adjusted to an alkaline level to reduce the risk of corrosion.

The pH value of PCE-10 was measured using PHB-3pH tester (Fig. 3-9) and the tester gave the value of 7.9. It suggested the PCE-10 is slightly alkaline and should not be corrosive to air conditioning systems.



Figure 3-9: pH testing

3.5.3 Particle sizing

Particle sizing and their distribution profiles are important parameters for specifying emulsions. Normally, dispersions with smaller particle sizes represent increased stability than those with coarse droplets. On the other hand, viscosity generally increases significantly with smaller particle sizes for a given mass or

volume concentration in the dispersed phase. This is confirmed by Schramm et al. [63] who reported that the size and characteristics of particles do affect the stability, viscosity and thermal properties of emulsions. In this regard, a laser particle size analyser, as shown in Fig. 3-10, was used to establish the size of the PCE-10 and as a reference for the stability test. As presented in Fig. 3-11, the mean particle size was obtained as 3.14µm.



Figure 3-10: Laser particle size analyser Bettersize 2000



Figure 3-11: Laser Particle sizing of PCE-10

3.5.4 Thermal conductivity

The thermal conductivity of the sample was determined with a KD2 Pro Thermal Properties Analyser (Fig. 3-12) which is able to test liquid samples with accuracy of \pm 5%. After repeated tests the average thermal conductivity of the PCE-10 was obtained as 0.4W/(mK) at a temperature of 25°C as summarised in Tab. 3-7.



Figure 3-12: KD2 Pro Thermal Properties Analyser

Test No.	Temperature	Thermal Conductivity (W/mK)
1	24.94	0.409
2	25.00	0.406
3	25.22	0.400
Average	25.11	0.403

Table 3-7: Thermal Conductivity Results

3.5.5 Phase transition temperature and heat capacity

Melting temperature and melting enthalpy are the two key thermal properties of phase change materials. They are crucially related to the phase change process and latent heat storage capacity.

DSC monitors heat effects associated with phase transitions as a function of temperature. In a DSC the difference in heat flow between the sample and reference at the same temperature, is recorded as a function of temperature.

$$\frac{dH}{dt} = \left(\frac{dH}{dt}\right)_{sample} - \left(\frac{dH}{dt}\right)_{reference}$$
(3-7)

The use of the extrapolated peak onset temperature T_e , end temperature T_c and the peak temperature T_p from DSC measurements is recommended when reporting the melting and freezing peak characteristics. T_e is the intersection between the tangent to the maximum rising slope of the peak and extrapolated sample baseline. Here, the phase change temperature range is defined as the temperature difference between T_e and T_c of a DSC curve. The integral under the DSC peak, above the base line, gives the heat of fusion for the process:

$$\int \left(\frac{dH}{dt}\right)_p dt = \Delta h_{f,p} \tag{3-8}$$

The following procedures were adopted for the test:

Step 1 3-5mg of PCM sample was loaded into one of the aluminium pans whilst the other one was kept as a reference pan

Step 2 The testing temperature was set between -5 $^{\circ}$ C and 15 $^{\circ}$ C at a heating rate of 1 $^{\circ}$ C/minute and then held for 5 minutes in order to achieve a stable curve.

The phase change temperature and the heat of fusion tests were conducted on 5mg sample of the RT10 with a Differential Scanning Calorimetry (DSC) apparatus (type EXSTAR SII DSC6220 as shown in Fig. 3-13) at a scanning rate of 1°C/min. After repeated tests the average value for the heat of fusion was obtained as 30.5 kJ/kg within a temperature range of 4 °C to 11.94°C as presented in Fig. 3-14 and the results from three tests are summarised in Tab. 3-8. Now by using Eq. (3-3), the heat storage capacity was calculated as 60.38kJ/kg which is 1.8 times of that of water (i.e. 33.4kJ/K) within the same temperature range.



Figure 3-13: Picture of DSC (EXSTAR SII DSC6220)



Figure 3-14: DSC curves of PCE-10

	Latent heat	Melting temperature	
Test No.	st No.	range	
	(KJ/Kg)	(°C)	
1	30.7	4.09-11.94	
2	30.6	4.11-11.92	
3	30.2	4.08-11.92	
Average	30.5	4.09-11.92	

Table 3-8: DSC results for PCE-10

3.5.6 Specific heat capacity

The specific heat of the PCE-10 could also be measured with the DSC equipment and calculated using the supporting MUSE software based on Eq. (3-9). The measurement condition for the sample, the reference (a standard sample with a known specific heat capacity e.g. sapphire) and the empty pan were the same.

$$C_{p,s} = \frac{u}{U} \cdot \frac{M_r}{M_s} \cdot C_{p,r}$$
(3-9)

Where,

Cp,s is the specific heat of sample Cp,r is the specific heat of reference Ms is the mass of sample Mr is the mass of reference U is the DSC difference of sample and empty pan u is the DSC difference of reference and empty pan

Fig. 3-15 shows the thermal profiles of PCE-10 during the repeated tests. The specific heat capacity curves did change sharply within the range of the phase change process with little differences between the tests. The curves however stabilised at about 3.924kJ/kgK after the phase change process was completed.



Figure 3-15: Specific heat capacity profiles of PCE-10

3.5.7 Sub-cooling temperature

The sub-cooling test was conducted with the T-history method which is normally used in measuring the thermal properties of bulk materials [94]. As demonstrated in Fig. 3-16, one of the test tubes was filled with PCM and the other with a reference material, usually water, due to its well-known thermo-physical properties. The tubes were then preheated above the melting temperature of the PCM and then simultaneously exposed to the ambient temperature to obtain the temperature history (T versus t) curves as presented in Fig. 3-17. Analysis of the curves showed a small sub-cooling degree of 0.2° C in the developed sample.



Figure 3-16: Schematic setup for T-history method



Figure 3-17: T-history curve of PCE-10

3.5.8 Rheological behaviour

The viscosity η is defined as the ratio of the shear stress τ to the shear rate.

$$\eta = \frac{\tau}{\gamma} \tag{3-10}$$

For Newtonian fluid, the viscosity η is a constant. For a non-Newtonian fluid, the viscosity is a function of shear rate γ . The non-Newtonian behaviour can be described using Ostwald Equation:

$$\eta = K \times \gamma^{n-1} \tag{3-11}$$

Where, K and n are coefficients which can be determined with a rheometer.

The coefficient n is the flow behaviour index of a fluid which characterizes its

degree of non-Newtonian behaviour. The greater the divergence of n from unity in either direction, the more non-Newtonian is the fluid. For Newtonian fluids, n=1 and for pseudo-plastic fluids n<1. The coefficient K, flow consistency index, defines the fluid's consistency. The larger the value of K, the thicker is the fluid.

Kinexus Lab+ Rotational Rheometer (Fig. 3-18) was used to determine the viscosity of PCE-10. The plate diameter is 50 mm, the cone diameter is 50 mm, and the cone angle is 1°. Fig. 3-19 illustrates the relationship between viscosity and shear rate for the developed PCME at 25°C and 10°C. The emulsion was a pseudo-plastic fluid and showed a shear thinning behaviour. The viscosity of the emulsion reduced rapidly with increasing shear rate but stabilised at around 13mPas after 400s⁻¹. However, the viscosity appeared quite high since it was about 13 times higher than that of water.



Figure 3-18: Kinexus lab+ Rotational Rheometer



Figure 3-19: Relationship between viscosity and shear rate for PCE-10 at 10 and $25^{\circ}C$

The coefficients K and n of PCE-10 were obtained by interpreting the curves in Fig. 3-19 with the Ostwald Equation Eq. (3-11). The obtained K and n are shown in Tab. 3-9.

Т (°С)	k	n
10	0.189	0.662
25	0.102	0.675

Table 3-9: Coefficients K and n of PCE-10

3.5.9 Stability test

Non-cycling mode

The emulsions were stored in containers made of glass at room temperature. The test was carried out at a room temperature through visual inspection and over periodic number of days. As presented in Fig. 3-20, separation started to occur

after 120 days of storage and by the end of 270 days there was a clear separation between the upper and lower layers. This phenomenon is called creaming and it is caused by the fact that paraffin has a lower density than water. Creaming is the movement of oil droplets under gravity or in a centrifuge to form a concentrated layer at the top of an oil-in-water emulsion. Particle size distribution of sample was tested at the end of the storage period. The result shows a value of 3.41µm (see Fig. 3-21) at the end of 270 days storage as compared with the original value of 3.14µm. This indicated the presence of oil droplets coalescence.



Figure 3-20: Phase separation of emulsion



Figure 3-21: Particle sizing of PCE-10 after 270-day storage

Cycling mode

The stability of the sample was examined after being subjected to 500 thermal and mechanical cycles. The test rig as illustrated in Fig. 3-22, consists of a storage tank, a heat exchanger and a circulating pump. The sample was initially cooled down from 25°C to 5°C and then reheated to 20°C. After 500 cycles, a clear layer of oil was observed (i.e. oiling-off phenomena) at the surface of the sample which was a sign of separation of the base material (RT10) from the emulsion as shown in Fig. 3-23.



Figure 3-22: Schematic diagram of the test rig



Figure 3-23: Oiling- off phenomena of PCE-10

The droplet size distribution, viscosity and thermal properties of the samples were then analysed and compared with the data before cycling. As shown in Fig. 3-24, there was a slight drop in viscosity from 13mPas to 10mPas at 500rpm. The droplet size also increased slightly from the original size of 3.14µm to 3.93µm (see Fig. 3-25). The latent heat of fusion was reduced from 30.5 to 29.9kJ/kg as illustrated in Fig. 3-26. This was attributed to the fact that when the emulsion was heated above the melting point, the crystallized droplets melted and fused together thus leading to coalescence and oiling-off at the surface of the emulsion. Although some changes were observed in the properties, the emulsion remained in dispersion and maintained the heat capacity after the test.



Figure 3-24: Rheological behaviour after and before thermal-mechanical cycles at 25°C


Figure 3-25: Laser Particle sizing after 500 thermal mechanical cycles



Figure 3-26: DSC curves of PCE-10 after 500 thermal mechanical cycles

3.6 Concluding remarks

In this paper, a novel paraffin-water emulsion (PCE-10) has been developed based on an organic paraffin material called RT10. In order to achieve good level of stability in the emulsion, the HLB method was used to obtain an optimum surfactant blend consisting of two selected surfactants (Tween 60 and Brij 52). The phase change emulsion PCE-10 meets the application requirements defined in section 3.1 and is an attractive candidate for cooling applications. Analysis of the thermophysical properties revealed the particle size of the PCE-10 sample to be 3µm with a storage heat capacity of almost twice as much as that of water and with a negligible level of sub-cooling. The PCE-10 also achieved a good level of storage stability for 9 months and was able to withstand over 500 cycles in a pumping system without any significant sign of degradation to particle size and heat storage capacity.

However the viscosity of the PCE-10 was found to be much higher than water which could contribute to high pressure drop in a pumping system. Its thermal conductivity was also found to be about 30% lower than the value for water and could influence heat transfer. Further enhancements of these thermophysical properties are therefore recommended.

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CHAPTER 4

NUMERICAL INVESTIGATIONS INTO THERMAL PERFORMANCE OF PCE-10

4.0 General

The optimal design of an integrated PCME air conditioning system requires a good understanding of flow behaviour and heat transfer characteristics of PCME in heat exchangers which cannot currently be readily deduced from manufacturer's data or published data. This is due to the fact that most of the published data cover straight pipes and not coiled pipes. Even though some technical data on various types of heat exchangers (e.g. air-emulsion direct-contact heat exchanger [95], double-coiled heat exchanger [80, 96]) are available, there is limited information regarding fin-and-tube heat exchanger which is the type commonly used in air conditioning systems. To this end computational fluid dynamics (CFD) investigation was carried out to analyse the flow behaviour and heat transfer characteristics of the developed PCE-10 sample in a fin-and-tube heat exchanger.

4.1 Heat transfer mechanism

Fig. 4-1 shows a sectionional view of pipe line, where the inner and outer surfaces are exposed to cold fluid and hot air respectively. Ignoring the wall thickness and wall temperature difference, the heat flux through the wall can be computed with Eq. (4-1).

$$q = \frac{\Delta T}{R} = \frac{T_{\infty} - T_i}{\frac{1}{h_i} + \frac{1}{h_o}}$$
(4-1)

Heat flux at the wall surface can also be computed using external heat transfer coefficient and external heat sink temperature T_{∞} as given in FLUENT [97] as:

$$q = h_o(T_{\infty} - T_w) = h_i(T_w - T_i)$$
(4-2)

Where,

 T_{∞} =External heat sink temperature

T_w=Wall temperature

 $T_i =$ Fluid Temperature

h_i = Internal heat transfer coefficient

 $h_o = External$ heat transfer coefficient

It can be seen from Eqs. (4-1) and (4-2), there are various parameters that affect the heat transfer rate across the pipe:

- Internal heat transfer coefficient, h_i=h_{fluid.}
- External heat transfer coefficient, $h_0=h_{air}$, which can be controlled by varying air flow velocity. Reducing h_{air} will reduce the heat flux.
- External free stream temperature T_{∞}
- Internal flow temperature T_i.



Figure 4-1: Heat transfer process of internal flow

Increasing fluid-side heat transfer coefficient h_{fluid} will increase the heat flux and reduces the wall temperature at the same time. However, h_{fluid} is not a property of a material, which makes it difficult to control. Considering the fluid inside the circular pipe at a uniform velocity v, heat transfer between wall and fluid is

achieved by convection. Convection is viewed as a mode of heat transfer in which there is combined effect of conduction and fluid motion between a solid surface and adjacent fluid flow. According to Newton's law of cooling, the convective heat transfer coefficient can be mathematically expressed as:

$$q = h_{\text{fluid}} \left(T_w - T \right) \tag{4-3}$$

For a viscous fluid flow inside a pipe, fluid layer coming into contact with the wall surface comes to a complete halt and adjacent layer slow down gradually due to viscosity effect. A thin velocity boundary layer is formed in near-wall region. Thickness of velocity boundary layer increases along the flow length until the entire pipe is filled up with boundary layer.

Similarly, when the fluid at a uniform temperature T enters a pipe with wall temperature T_w , the fluid particles in contact with the wall surface tries to achieve thermal equilibrium at the surface and exchanges energy. Heat is then transferred to the next fluid layer and thus results in temperature gradients in the fluid. These temperature gradients continues to exist till a distance of δ'_{fluid} is reached where the temperature effect is negligible. Thickness of boundary layer δ'_{fluid} increases along the flow until reaches the centre of the pipe [98].

Due to the no-slip condition of a stationary wall and the viscous damping by the fluid, there is a very thin layer of fluid at the wall in which the flow velocity can be considered as zero. The main heat transfer method in this layer is conduction, outside this layer, convection also happens. Energy exchange between wall and fluid has to pass through this stationary layer. Ignoring the effect of radiation, total amount of heat transferred between wall and fluid stream equals to the heat conduction through the thin layer [99]. Applying the Fourier Law to pipe wall, the heat flux can be expressed as:

$$q = -\mathbf{k} \frac{\partial T}{\partial \mathbf{n}}\Big|_{n=0} \tag{4-4}$$

Recalling the Newton's Law of cooling to the internal flow, fluid heat transfer coefficient can also be expressed as:

$$h_{\rm fluid} = \frac{-k \frac{\partial T}{\partial n}\Big|_{n=0}}{T_w - T}$$
(4-5)

As Eqs. (4-4) and (4-5) suggest, the amount of heat transfer is restricted by thermal conductivity of fluid, k, and the temperature gradient $\frac{\partial T}{\partial n}\Big|_{n=0}$ which is related to the fluid properties such as viscosity, flow conditions (laminar or turbulent) and flow velocity.

4.2 Mathematical modelling

4.2.1 Basic Assumptions

The following assumptions were considered:

- The PCE-10 was treated as an incompressible, steady state, homogeneous single-phase fluid with negligible effect of viscous heating.
- Fluid properties were assumed to be constant with respect to temperature, except heat capacity of material.
- Phase change process was assumed to take place within a specific

temperature range but volume change was neglected.

- Buoyancy effects were neglected as density was assumed to be constant.
- Inter-particle interaction and particle-wall interactions were neglected.

4.2.2 Governing Equation

Modelling conjugate heat transfer

The governing equations include continuity, momentum and energy equations from which all the theory and more complex models are derived. The continuity and momentum equations were used to calculate velocity vector. The energy equation was used to calculate temperature distribution and wall heat transfer coefficient. The basic equations for the conjugate heat transfer between fluid (PCE-10) and solid (finned tube) are summarized below [97]:

Conservation of mass:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{v}) = 0 \tag{4-6}$$

Conservation of momentum:

$$\frac{\partial}{\partial t}(\vec{\rho v}) + \nabla \cdot (\vec{\rho v v}) = -\nabla P + \nabla \cdot \left[\mu \nabla \vec{v} - \frac{2}{3}\mu \nabla \cdot \vec{v}\right] + \vec{\rho g} + \vec{F}$$
(4-7)

Conservation of energy for fluid:

$$\frac{\partial}{\partial t}(\rho E) + \nabla \cdot (\vec{v}(\rho E + P)) = \nabla \cdot (k_{eff} \nabla T) + S_E$$
(4-8)

Conservation of energy for solid:

$$0 = \nabla \cdot (k_{eff} \nabla T) \tag{4-9}$$

Where, P, μ_m and ρg are static pressure, molecular viscosity and gravitational

body force, respectively. F defines external body force and also contains other model-dependent source terms such as porous-media and user-defined sources. E equals to the sensible enthalpy for phase p. S_E includes any other volumetric heat sources.

Modelling turbulence flow

Turbulent flows are characterized by fluctuating velocity fields. These fluctuations mix transport quantities such as momentum, energy, and species concentration, and cause them to fluctuate as well. Since these fluctuations can be of small scale and high frequency, they tend to be computationally too expensive to simulate directly in practical engineering calculations. Instead, the instantaneous governing equations can be time-averaged or otherwise manipulated to remove the small scales, resulting in a modified set of equations that are computationally less expensive to solve. However, the modified equations contain additional unknown variables, and therefore turbulence models are needed to determine these variables in terms of known quantities.

FLUENT provides the following choices of turbulence models which are widely used: K- epsilon model (K- ε model) and K-omega model (K- ω model). In the derivation of the K- ε model, the assumption is that the flow is fully turbulent, and the effects of molecular viscosity are negligible. The standard K- ε model is therefore valid only for fully turbulent flows. SST k- ω model was developed by Menter [100] to effectively blend the robust and accurate formulation of the k- ω model in the near-wall region with the free-stream independence of the k- ω model in the far field. To achieve this, the K- ε model is converted into a k- ω formulation. The SST k- ω model is similar to the standard k- ω model, but includes the following refinements:

- The standard k-ω model and the transformed K-ε model are both multiplied by a blending function and both models are added together. The blending function is designed to be one in the near-wall region, which activates the standard k-ω model, and zero away from the surface, which activates the transformed K-ε model.
- The SST model incorporates a damped cross-diffusion derivative term in the ω equation.
- Modified turbulent viscosity formulation to account for the transport effects of the principal turbulent shear stress.

These features make the SST model more accurate and reliable for a wider class of flows than the standard k- ω model. In industry, flows inside a coiled tube are mainly turbulent or transition flow, the shear-stress transport (SST) K-omega model is selected.

Transport equation for SST K-mega model:

$$\frac{\partial}{\partial t}(\rho k) + \nabla \cdot (\rho k \vec{v}) = \nabla \cdot [(\mu + \frac{\mu_t}{\sigma_k})\nabla k] + G_k - Y_k + S_k$$
(4-10)

$$\frac{\partial}{\partial t}(\rho\omega) + \nabla \cdot (\rho\omega v) = \nabla [(\mu + \frac{\mu_t}{\sigma_\omega})\nabla\omega] + G_\omega - Y_\omega + D_\omega + S_\omega$$
(4-11)

Where, σ_k and σ_{ω} are constant turbulent Prandtl numbers for k and ω , respectively. Y_k and Y_w represent the dissipation of k and ω due to turbulence. D_w represents the cross-diffusion term.

Modelling phase change emulsions

An effective heat capacity approach, first introduced by Alisetti et al. [75] was used since it can be more easily implemented in practice. In the effective specific

heat model, the phase change material is assumed to be melted over a constant temperature, so that the phase change effects are directly incorporated into energy equation by assuming specific heat capacity of phase change material to be a function of temperature, as Fig. 4-2 shows.

The following equation can be used to calculate the suspension specific heat capacity for all temperature:

$$C_{p,m,eff}(T) = \begin{cases} X_W C_{p,w} + X_{PCM} C_{p,PCM} & T_s < T_e \\ X_W C_{p,w} + X_{PCM} C_{p,PCM} + X_{PCM} \Delta h / \Delta T & T_e < T_s < T_c \\ X_W C_{p,w} + X_{PCM} C_{p,PCM} & T_c < T_s \end{cases}$$
(4-12)



Figure 4-2: Specific heat capacity and enthalpy of phase change material with temperature [77]

In order to use the above equation, it is necessary to define T_{MPL} , onset melting temperature and T_{MPH} , end phase change temperature. Outside the phase change range, i.e. for T< T_{MPL} and T> T_{MPH} , the specific heat of emulsion is the mass-

averaged specific heat of its constituents [72]. Since the specific heat capacity of PCM in its solid and liquid phases are usually similar and do not vary significantly with temperature, it can be assumed as a constant. As a result, the specific heat of PCME, was calculated using the above equation, can also be assumed to be a constant outside the phase change range.

In the phase change range, the specific heat of PCE can be obtained from suitable analytical tests (e.g. Differential scanning calorimetry DSC). The specific heat is related to latent heat of PCM h, through the following equation:

$$h = \int_{T_{MPL}}^{T_{MPH}} \int_{T_{MPL}} C_{p,PCM} dT$$
(4-13)

Previous studies with phase change slurries showed that the shape of specific heat capacity-temperature curve has little effect on heat transfer process [101]. As a result, it is possible to assume the specific heat capacity of PCM remains constant during the phase change process. Its effective value in the phase change range is therefore given by following equation:

$$C_{\mathrm{p},PCM} = \frac{h}{T_{MPH} - T_{MPL}} \tag{4-14}$$

4.3 Simulation data

In the present work, the phase change material emulsion PCE-10 was used. The emulsion contains 3.75wt% surfactant and 25wt% of RT10. Listed in Tab. 4-1 are the simulation data for the PCE-10 which was extracted from Tab. 3-8 and 3-9. The heat capacities were calculated by using Eq. (4-12) and Eq. (4-14).

Chapter 4

Properties	Measured	Unit	
Density	940	kg/m3	
	T<4°C	3800	
Specific Heat Capacity	$4^{\circ}C \leq T \leq 12^{\circ}C$	8718	J/kgK
	12°C <t< td=""><td>3800</td><td></td></t<>	3800	
Thermal conductivity	0.40	W/mK	
Consistency index k	10°C	0.1887	
	25°C	0.102	-
Flow behaviour index n	10°C	0.662	
	25°C	0.675	-

Table 4-1: Specification of PCE-10

4.4 Modelling procedure

-

Heat transfer and flow performance of PCME in coiled pipes were investigated numerically using Computational Fluid Dynamics (CFD) software ANSYS. Computational fluid dynamics (CFD) study of the system started with the construction and meshing of desired geometry in ANSYS workbench design module. Meshing was the discretization of the domain into small volumes where the equations were solved by the help of iterative methods. Simulation in FLUENT started with the describing of the boundary conditions for the domain and then led to modelling of the entire system. The steps which were followed for the entire modelling process are shown in Fig. 4-3.



Figure 4-3: Modelling and solution procedure

4.4.1 Geometry & Meshing

Fig. 4-4 shows the views of the fin-and-tube heat exchanger which was considered for the modelling exercise. It consists of horizontal tubes with coiled ends and thin vertical fins meant to increase the heat transfer area. The physical parameters are given in Tab. 4-2.



Figure 4-4: Physical model of finned-tube coil

Parameter		
Finned-tube length	250mm	
Number of circuits	3	
Distance between rows	22mm	
Number of tubes per circuit	9	
Distance between columns	25mm	
Tube material	Copper	
Tube inner diameter D	9.3mm	
Tube wall thickness	0.35mm	
Fin material	Aluminium	
Fin thickness	1mm	
Face area	250*250	

To analyse the problem with CFD, the geometry of the heat exchanger has been prepared in ANSYS workbench design module, as shown in Fig. 4-5.



Figure 4-5: 3D model (a) finned pipe (b) fluid flow

The geometry was meshed with automatic meshing method of defined global size, as shown in Fig. 4-6. To account for the high variable gradients which usually exist near the pipe wall due to the no-slip condition of the wall, inflation layers were created near the wall. The number of nodes and elements for the current case were 733087 and 2197511. The mean orthogonal qualities for the whole model and fluid domain were 0.679 and 0.948, respectively.



(b)

Figure 4-6: Grid Meshing (a) finned pipe (b) fluid flow

4.4.2 Boundary Condition

The simulation was subjected to the following boundary conditions which were set according to design conditions of real air conditioning systems [102, 103]. Velocity inlet boundary condition was applied at the inlet. 7°C fluid entered the heat exchanger at 0-1m/s. The turbulence intensity was fixed at 5% with a hydraulic diameter pipe of 9.3mm. The pressure at outlet was assumed to be 0.

There are two main types of thermal boundary conditions that can be applied at the wall. One is specifying a constant heat flux and the other is specifying a constant wall temperature. However, in heat exchangers, the incoming fluid exchanges heat with the pipe walls. This will result in change in temperature of the fluid as it moves down stream. This change in temperature of fluid will also result in change in temperature differential between the walls and fluid, which ultimately leads to the change in heat transfer rate. The problem is therefore neither isothermal nor iso-flux. In such cases, FLUENT provided a third option: convective heat transfer wall boundary, which is the best and most accurate approximation of real life example. External air heat transfer coefficients were determined using following empirical correlations [104].

$$j = 16.06 \operatorname{Re}_{D}^{-1.02(pf/D) - 0.256}(\frac{A_{a,total}}{A_{tube}}) \stackrel{-0.601}{\stackrel{-0.001}{N_{bank}}}(\frac{pf}{D})$$
(4-15)

$$h_{a} = \frac{j\rho(m_{air}/\rho A_{c})C_{p}}{\Pr^{2/3}}$$
(4-16)

In this case, three different air flow velocities of 0.8, 1 and 1.5 m/s were studied and the corresponding convective heat transfer coefficients were 38.7, 49.9 and 59.3W/m²K, respectively. Therefore, the convective heat transfer coefficients of 40, 50 and 60W/m²K were applied at the external wall surface of the pipe with free-stream air temperature of 300K. A no-slip and no-penetrating boundary conditions were also imposed on the wall surface of the pipe.

4.4.3 Solution Control

The pressure-velocity coupling was done using coupled scheme and the pressure discretization was achieved by the PRESTO procedure. Second order upwind scheme was employed to discretize the convection terms, diffusion terms and other quantities resulting from the governing equations.

4.5 Result and discussion

4.5.1 Pressure distribution

Fig. 4-7 shows the total pressure drop for pure water and PCE-10 versus the flow velocity at different temperatures. It can be seen that at the same velocity, the PCE-10s produced much bigger pressure drops i.e. 2-10 times higher than water due to high viscosities. Moreover, the pressure loss of the emulsion at 10°C (within the phase change range of 4-12°C) was higher than that at room temperature, because of the existence of solid paraffin.



Figure 4-7: Pressure drop of the emulsion versus flow velocity compared that of water at 10°C and 25°C.

4.5.2 Heat transfer characteristics

The temperature profiles at the exit of the heat exchanger for PCE-10 and water are given in Fig. 4-8. The average outlet temperature of PCE-10 was lower than water due to the differences in their heat capacities. The heat capacity of PCE-10 was twice as much as that of water hence the smaller temperature variation at the outlet.



Figure 4-8: Temperature contour plot for outlet at v=0.3m/s

Fig. 4-9 shows the temperature profiles for $h_{air}=60W/m^2K$ and $T_{\infty}=300K$. It can be seen that PCE-10 helped to reduce the wall temperature and therefore increased

the temperature gradient between wall and air. However, the difference between temperature gradients for PCE-10 and water gradually reduced as the flow rate increased.

As Eq. (4-2) suggested, at the constant external heat transfer coefficient, heat transfer rate was proportional to the temperature gradient between wall and air, it is thus reasonable to believe that PCE-10 would help increase the heat transfer rate significantly at low flow rates but gradually lost its effectiveness at heat transfer enhancement when flow velocity was higher.



Figure 4-9: Wall temperature profile v/s flow velocity at h=60W/m2K

Fig. 4-10 compares the heat transfer rates of the PCE-10 and water at different external heat transfer coefficients. It is obvious that total heat transfer rate increased with external heat transfer coefficient increasing. It can be seen that PCE-10 did help to increase the total heat transfer rate as predicted at a constant heat transfer coefficient. For example, at $h=60W/m^2K$, the total heat transferred by PCE-10 was about 1.1-1.3 times the value for water at the same flow rate. The same trends were witnessed in the other two cases.

However, the heat transfer enhancement did vary at different flow velocities, as the differential heat transfer ($Q_{(PCE-10)}-Q_{(Water)}$) did not increase linearly with flow rate. There was rather a significant increase in the heat transfer rates at lower flow rates with the maximum heat transfer enhancement occurring within the range of 0.5-0.6m/s. Similar results were also obtained for h= 50W/m²K and 40W/m²K where the maximum heat transfer enhancement occurred around 0.4-0.5m/s.



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Figure 4-10: Heat transfer rate v/s flow velocity.

Fig. 4-11 shows the relationship between pumping power against heat transfer rate. Pumping power is defined as

$$P = p \times V \tag{4-17}$$

Where, P= pump power, p= pressure drop, and V=volume flow rate

It can be seen from Fig. 4-11 (a) that the pump power consumption increased with the heat transfer rate increasing. At a constant air-side heat transfer coefficient, reduction was observed in PCE-10 pump power consumption required to transfer a same amount of heat in comparison with water. To transport the same amount of energy, a higher mass flow will be necessary with water.

However, as shown in Fig. 4-11 (b), when heat transfer rate decreased as the flow velocity reduced, the pump power consumptions of PCE-10 were actually higher than that of water. For heat transfer coefficient of 60W/m²K, PCE-10 only showed its advantage when transporting heat over 1.6kW and the minimal flow velocity is 0.6m/s. For external heat transfer coefficients of 50 and 40W/m²K, the heat transfer rates were 1.4kW and 1kW, the corresponding flow velocity is 0.5 and 0.3m/s respectively. Therefore, in the best interest of pump energy saving, the PCE-10 flow rate should not be lower than 0.3m/s.

Generally speaking, water worked better at high heat transfer rate while PCE-10 system was more effective at low heat transfer coefficient, as PCE-10 could still achieve the pump energy saving at low fluid flow velocity.



Figure 4-11: Pumping power v/s heat transfer rate at flow velocity range



Figure 4-12: Nusselt number v/s Reynolds number plot

The fluid-side convective heat transfer coefficient, Nusselt number and generalized Reynolds number for the PCE-10 were calculated from the following equations:

$$h = \frac{Q}{T_w - T_{fluid}} \tag{4-18}$$

$$Nu = \frac{h D_i}{K} \tag{4-19}$$

$$\operatorname{Re} = 8^{1-n} \left(\frac{3n+1}{4n}\right)^{-n} \frac{\rho u^{2-n} D_i^n}{\eta}$$
(4-20)

Where,

Q = surface flux (W/m^2),

 $T_{wall} = =$ average temperature (°C) of tube wall

 T_{fluid} = average temperature (°C) of fluid

D = inner diameter of pipe (m)

k= thermal conductivity of fluid

Fig. 4-12 represents the plot of Nusselt number versus Reynolds number (Re) for the PCE-10 in a velocity range of 0-1m/s which fell in the laminar regime with Re<2300. The results showed a remarkable improvement in the heat transfer performance. In comparison with water, the Nusselt number did increase by more than 10 times at the same Reynolds number condition. Therefore, the Reynolds number was established as a dominating parameter affecting the Nusselt number of the emulsion.

4.5.3 Factors affecting heat transfer characteristics

As shown in Eq. (4-4), the amount of heat transfer is restricted by thermal conductivity of fluid, k, and temperature gradient $\frac{\partial T}{\partial n}\Big|_{n=0}$. Temperature gradient is related to the fluid properties such as viscosity, flow conditions (laminar or turbulent) and flow velocity. Previous simulation results revealed that increase in flow velocities and external heat transfer coefficient did help to increase the temperature gradient and ultimately led to the increase of total heat transfer rate. However, for a given flow condition, temperature gradient of fluid is constant. Therefore by increasing thermal conductivity the heat transfer of the fluid should be enhanced.

To this end series of CFD simulations were carried out to explore the effectiveness of increasing thermal conductivity. The simulation was subjected to the following boundary conditions: v=0.5m/s, T=7°C, T_{∞} =27°C and h_0 =60W/m²K. The result of the total heat transfer rate was plotted in Fig. 4-13. As shown in the graph, heat transfer coefficient h_{fluid} and heat transfer rate increased proportionally to fluid thermal conductivity. For instance, a 10% increase in heat transfer rate could be



achieved if the conductivity was increased to 0.6W/mK.



4.6 Concluding remarks

CFD analysis for PCE-10/coiled tube system was carried out. The results of rheological performance and heat transfer parameters have been compared with water under similar operating conditions.

The flow behaviour was analysed within a range of 0.1-1m/s, in which the flow was considered to be laminar. The pressure drop was found to be 2-10 times much higher than that of water under identical conditions.

The heat transfer study was performed under convective heat transfer wall boundary. It was found out that the PCE-10 did help to increase the total heat transfer rate by a factor of 1.1-1.3 within flow velocity of 0.4-0.6m/s. However, the PCE-10 gradually lost its advantages as flow velocity increasing.

The relationship between pump power and heat transfer rate was also investigated. It was observed that for the same value of heat transfer rate, the pumping power was decreased in comparison with water. However, at lower fluid flow velocity, the pump power consumption of PCE-10 would still be higher than water. But generally speaking, water worked better at high heat transfer rate while PCE-10 system was more effective at low heat transfer coefficient

In the best interest of pump energy saving and optimum heat transfer efficiency, the optimal working range should be 0.3-0.6m/s.

The study also explored the factors could affect the heat transfer characteristic of

PCE-10, such as air side conditions, flow condition and fluid properties. Increase in thermal conductivity also helped to enhance the heat transfer effect. For instance a 10% increase in heat transfer rate could be achieved if the conductivity was increased to 0.6W/mK. Therefore thermal conductivity enhancement is recommended for further studies. However, in this part of the study, the simulation was only carried out in the laminar region. Further works are also recommended to numerically explore the heat transfer performance in the turbulent regime. In addition, heat transfer studies with different geometries would also be helpful to enhance the knowledge of heat transfer behaviour.

CHAPTER 5

EXPERIMENTAL THERMAL PERFORMANCE EVALUATION OF PCE-10

5.0 General

Theoretical studies in Chapter 4 indicate promising performance of phase change emulsion as a heat transfer medium. However, experimental investigations on phase change emulsion heat transfer characteristics in fin-and-tube heat exchangers appear to be limited for practical engineering so far. In an effort to understand the heat transfer effectiveness and fluid characteristics of phase change emulsions, an experimental setup consisting of a fin-and-tube heat exchanger was designed for that purpose. These results was used to validate the test results with the numerical predictions from Chapter 4.

5.1 Experimental setup

An experimental setup for the determination of the pressure drop and heat transfer performance of a commercially available fin-and-tube heat exchanger using PCE-10 as a coolant was built. The schematic view of the test rig is shown in Fig. 5-1. The PCE-10 was stored at 7°C in an insulated thermostatic bath equipped with cooling device. The fin-and-tube heat exchanger was manufactured by Xinhong Refrigeration Co., Ltd as detailed in Fig. 5-2 and Tab. 5-1. The coolant was distributed into the coil from the inlet manifold located on the top of the coil and returned to the storage tank through the bottom outlet.



Figure 5-1: Scheme of the test rig



Figure 5-2: Geometry of finned-tube coil

Symbols	Description	Dimension	Symbols	Description	Dimension
P _f	Twice the amplitude of fins corrugation	3mm	\mathbf{K}_{fin}	Conductivity of fin	200 W/mK
N _{bank}	Number of banks of tubes	3	Xf	half wavelength of in wave	1.5mm
N _{tubes/bank}	Number of tubes per bank	9	D	Interior diameter of tube	9.3mm
$\mathbf{N}_{\mathbf{fin}}$	Number of fins	83	r	Outer radius of tub	4.96mm
$\mathbf{N}_{\mathbf{fin}}$	Number of fins	83	$\mathbf{A_{f}}$	Total wetted area of fins	2.3m2
Pt	Transverse pitch	25mm	A _{a,total}	Total Airside area	2.8m2
Pı	Longitudinal bank-bank pitch	22mm	\mathbf{A}_{tube}	Total outer area of tubes	0.21m2
\mathbf{L}_{tube}	Length of one tube	0.25m	D _{coil}	Diameter of turning	25mm

Table 5-1: Fin-and-tube coil data

The volume flow rate of the PCE-10 was measured at the inlet of finned-tube coil with an accuracy of $\pm 0.1\%$. The temperatures of fluid and air flow, as well as the tube wall temperature were measured at the coil inlet and outlet by K-type thermocouples with accuracy of $\pm 0.07\%$. The pressure drop was measured by two pressure sensors with accuracy of $\pm 1\%$. The air velocity was measured at the inlet and outlet flow cross-sections with TESTO anemometer 410-1, whose accuracy is $\pm 2\%$. The full specifications are presented in Tab. 5-2.

Table 5-2: Rig specification

Data loggers and sensors:

Agilent Technologies-Data Logger Switch Unit 34970A Data Acquisition LZB-10 glass rotameter flow meter Accuracy: ±2.5% TESTO- Anemometer 410-1 Accuracy: ±(0.2 m/s +2 % of reading) SZ Joint Sensor Instruments (H.K.) Ltd.- Model 641S Pressure Sensor Accuracy: ±1%

Omega-K type thermocouple Accuracy: ±0.07%.

Cooling units:

Tianheng Instrument Factory-Low-Temperature Thermostat Bath THD-1010 Cooling medium temperature: 7°C Flow rate: 160L/h to 40L/h

Coil:

Xinhong Refrigeration Co., Ltd-Fin-and-Tube Coil (3-Row)

5.2 Experimental procedure

Different sets of experiments were conducted to determine the pressure drop and heat transfer performance of developed PCE-10 in fan-coils units. Initial experiments were carried out with water, and the results are served as control
groups.

The pressure drops of PCE-10 were measured under almost isothermal conditions (fan turned off) and inlet temperature at 10°C. The flow rate used here varied between 24l/h-144l/h (or 0.1-0.6m/s) at intervals of 8l/h.

The heat transfer experimental analysis was carried out under standard operation conditions as defined in Design manual for domestic heating cooling and air conditioning [102, 103]. The design manual specified that for chilled water fan coil unit systems, the design conditions for water and air side temperatures are 7 and 27 °C, respectively, and the optimal chilled water flow and air velocities are 0.3-0.5m/s and 1.5-2m/s [102, 103]. Therefore, the following operating conditions were adopted: flow rate of 40l/h-144l/h (0.2-0.6m/s), inlet air temperature of 27°C, air flow velocity of 0.8, 1 and 1.5 m/s.

5.3 Data deduction

To describe the flow inside the tube, the generalized Reynolds number needs to be introduced:

$$\operatorname{Re}^{*} = 8^{1-n} \left(\frac{3n+1}{4n}\right)^{-n} \frac{\rho u^{2-n} D_{i}^{n}}{\delta}$$
(5-1)

$$Nu = \frac{h D_i}{\lambda} \tag{5-2}$$

Where, n characterizes the degree of non-Newtonian behaviour of fluids. As stated earlier, the PCE-10 is a non-Newtonian fluid and so n equals to 0.51 in this case. For water which is a Newtonian fluid, n=1.

As the tube wall temperature is difficult to measure for finned surface, the fluid

side heat transfer coefficient is calculated through overall heat transfer coefficient and air-side heat transfer coefficient.

The heat transfer rate (Q) of heat exchanger was determined from the energy balances on the fluid side and on the air side. The air-side heat transfer rate and fluid-side heat transfer rate, which are calculated as:

$$Q_{air} = C_{p,air} \times m_{air} \times T_{air,diff} + m_{air} \times LH \times (g_{in} - g_{out})$$
(5-3)

$$Q_{fluid} = C_{p, fluid} \times m_{fluid} \times T_{fluid, diff}$$
(5-4)

Where,

LH =latent heat of vaporisation of water=2450kJ/kg.

 g_{in} - g_{out} =air moisture content difference (kg/kg).

The air temperature and relative humidity at the inlet and outlet cross-sections were determined by averaging the experimental measurements. The air mass flow rate was obtained by considering the outlet air density.

A detailed analysis of the experimental uncertainties was carried out. The uncertainty calculation procedure was based on the application of the general uncertainty propagation expression according to the ISO Guide [105] to the calculated magnitudes. The uncertainty analysis revealed that the mean typical uncertainties in the calculation of the heat transfer rate from the energy balance on the air side are higher than from the energy balance on the fluid side. The mean values of the mean typical uncertainties in the heat transfer rate from the energy balance on the fluid side and on the air side were 5.8% and 30.8%, respectively. Taking into account these results, the capacity of the fan-coil was calculated at the fluid side.

The relationship between the overall heat transfer coefficient (U) and the heat transfer rate (Q) can be demonstrated by the following equation:

$$Q_{fluid} = U \times A \times T_m = U \times (\pi D_i L) \times F \times LMTD$$
(5-5)

 T_m , also named as Logarithmic Mean Temperature Difference (LMTD) for a counter-current heat exchanger, was defined as:

$$LMTD = \frac{(T_{H,out} - T_{C,in}) - (T_{H,in} - T_{C,out})}{\ln[(T_{H,out} - T_{C,in})/(T_{H,in} - T_{C,out})]}$$
(5-6)

The LMTD correction factor F is required because the heat exchanger is crossflow rather than parallel-flow or counter-flow. The correction factor is obtained from the calculation of two ratios P and R associated with the inlet and outlet temperatures of the two fluids as shown in Fig. 5-3. The value of F is then read from the graph.



Figure 5-3: Correction factor for a single-pass cross-flow heat exchanger with one fluid mixed [106]

Several equations can be used to determine the U value, one of which is:

$$\frac{1}{U A_{average}} = \frac{1}{h_{fluid} \times A_{fluid}} + \frac{\ln(D_{ext} / D_{int})}{k \times 2\pi L} + \frac{1}{(\eta^* \times h_{air} \times A_{air})}$$
(5-7)

where, η^* is the surface efficiency defined by Schmidt [107].

$$\eta^* = 1 - \frac{A_f}{A_{a,total}} (1 - \eta_f) \tag{5-8}$$

$$\phi = (\frac{r_f}{r} - 1)[1 + 0.35\ln(\frac{r_f}{r})]$$
(5-9)

$$\eta_f = \frac{thn(mr\phi)}{mr\phi} \tag{5-10}$$

$$m = \sqrt{\frac{2h_a(\frac{C_s}{C_p})}{k_{fin}t}}$$
(5-11)

Where,

r = outer diameter of tube,

 r_f = the outer radius of fin.

 C_s/C_p = the correction for heat/mass transfer for a wetted surface. C_s/C_p is set to unity if the fins are dry

In order to determine the heat transfer coefficients between the fluids and the heat exchanger surface, convection correlations must be used. The air side heat transfer coefficient for wavy-louvered fins is calculated from following empirical overall correlations [104]:

$$j = 16.06 \operatorname{Re}_{D}^{-1.02(pf/D) - 0.256} \left(\frac{A_{a,total}}{A_{tube}}\right)^{-0.601} N_{bank}^{-0.069} \left(\frac{pf}{D}\right)^{0.84}$$
(5-12)

$$h_{a} = \frac{j\rho(m_{air}/\rho A_{c})C_{p}}{\Pr^{2/3}}$$
(5-13)

Recalling Eq. (5-7), the liquid side heat transfer coefficient can be determined by

$$h_{fluid} = \frac{1}{\left[\frac{1}{U A_{average}} - +\frac{1}{(\eta^* \times h_{air} \times A_{air})}\right] \times A_{fluid}}$$
(5-14)

5.4 Results and discussion

5.4.1 Pressure drop

Experimental measurements of the isothermal pressure drop were carried out with water and PCE-10 at 10°C. The fluid volume flow rate through the coil was varied between 24l/h and 144l/h at intervals of 8l/h to obtain the graphical results in Fig. 5-4.

In this experimental study, the coil was fully insulated to prevent air from flowing through the coil. Even though it could not completely eliminate the effect of free convection yet the air flow velocity across the heat exchanger was very lower. Therefore, the result of pressure drop was taken under condition that was similar to the CFD model.

Pressure drop for PCE-10 is found to be higher than that of water at same flow rate. Analysis of the results indicates that the pressure drop of PCE-10 did increase with the flow velocity. For instance, at 0.5m/s, the pressure drop of PCE-10 was 5 times of water. The extensively high pressure drop of PCE-10 was due to its high viscosity. Presence of a secondary phase dissipates kinetic energy, thus increasing the resistance to flow.



Figure 5-4: Pressure drop of PCME and water vs. flow velocity at v_{air}=0m/s.

Friction factor

There are several models that can be used to describe the friction factors of coil pipes. Wojtkowiad and Chen [108] tried to define the relationship of friction factor and Dean number. Dean number, D_n , is used to describe fluid flows in curved tubes, which measures the balance between the forces due to inertia and centripetal acceleration and the viscous force.

$$f = a + b \ln(D_n) + c D_n^{-1}$$
(5-15)

$$D_n = \operatorname{Re}(\frac{D}{D_{coil}})^{0.5}$$
(5-16)

Where,

D = inner diameter of tube

 D_{coil} = diameters of turning

L= length of straight pipe

$$a = 0.121433 - 0.00182312 \times (\frac{L}{D_{coil}})$$

$$b = -0.010311 + 0.0019359 \times (\frac{L}{D_{coil}})$$

$$c = 16.68855 - 0.16757 \times (\frac{L}{D_{coil}})$$

The above equation is valid in following ranges: $18 < D_n < 38002$, $0 < L/D_{coil} < 40$, 2R/d=6.62.

Or,

$$\ln(\frac{f_c \,\text{Re}}{64}) = a + b \left(\ln D_n\right)^2$$
(5-17)

Where, a=0.021796 and b=0.0413356. It stands only when 2R/d=6.67-27.85, Re=500-20000, and L/D=0.

However, the above two equations both failed to describe this test. Because the above two equations are only valid with specified spacer lengths and curvature coefficient. The dimension of U-shape coil in this research is fixed, i.e. 2R/d and l/d are constant, which are not in the ranges given with the equation.

In order to analytically calculate the pressure drop, the Equivalent Length Method was adopted [109]. In this method, the equivalent length of a straight pipe, corresponding to a specific fitting (e.g. bends and manifolds) is determined. The equivalent length of a fitting is the length of a straight pipe that produces the same pressure drop as the fittings. When the equivalent length is obtained, it is added to the total pipe length and the new length is then used to calculate the pressure loss with Darcy–Weisbach formula (Eq. (5-19)).

$$\Delta P = \Delta P_s + \Delta P_{fitting} = f \frac{L}{D} \frac{\rho v^2}{2} + f \frac{L_{equ}}{D} \frac{\rho v^2}{2}$$
(5-18)

Where,

L/D =length factor, which is the ratio of the length to diameter of the pipe;

 ρ = density of the fluid, (kg/m3);

v = mean velocity of the flow, (m/s),

f = friction actor.

The tests with water were used to determine an equivalent length of the coil bends and manifolds. Water flow at 0.1-0.5m/s mainly fell in the laminar region. The friction factor for laminar flow can be calculated by

$$f_L = \frac{64}{\text{Re}} \tag{5-19}$$

For this specified coiled pipe, the mean equivalent length corresponding to the bends and manifolds was 31m.

Assuming the equivalent length obtained for bends and manifolds is also applicable to phase change emulsion, then the friction factors could be determined from the experimental pressure drop measurements using Eq. (5-18). The experimental friction factors are plotted against Reynolds number in Fig. 5-5. The fitting formulas are derived by NLSF method:

$$f = 14.44 \,\mathrm{Re}^{-1.007}$$
 for $T = 10^{\circ}\mathrm{C}$ (5-20)



Figure 5-5: Friction factor of PCME vs. Reynolds number at 10°C.

5.4.2 Heat transfer

The operating conditions considered to carry out the fan-coil heat transfer tests are listed in Tab. 5-3. The inlet and outlet temperatures of air and fluid, as well as the fluid pressure drop across the coil were recorded. The experimental data shown in this section were recorded after all parameters were stable for at least 5 min.

	Analysis
PCME volume flow rate (l/hr)	144-40
Inlet air temperature(°C)	27-28
Inlet air velocity(m/s)	0.8, 1, 1.5
Inlet PCME temperature (°C)	7-8

Fig. 5-6 shows the experimental results of the fan coil heat transfer rate against the inlet flow velocity. Analysis of the results shows that PCE-10 did helped to improve the heat transfer rate by a factor of 1.1 at a constant heat transfer coefficient on average. For example, at $v_{air}=1.5$ m/s, the total heat transferred by PCE-10 was about 1.1-1.2 times the value for water at the same flow rate. The same trends were witnessed in the other two cases.

The relationships between pumping power against heat transfer rate are also plotted. It was noticed that the pressure drop for PCE-10 was bit higher than those measured at the air flow velocity of 0m/s, which was probably caused by the heat transfer process. A higher pressure drop increased the power input to the pump which made PCE-10 less competitive to water. It was also observed that for the same heat transfer rate, the pump power for water was actually lower than that for PCE-10. However, at the case of air flow velocity of 0.8m/s, for heat transfer rate higher than 1.1kW, the water pump power consumption was higher than that for PCE-10, which suggested that PCE-10 is more suitable at lower air side heat transfer coefficient (low air flow velocity).







(c) Air side flow velocity $V_{air} = 0.8 \text{m/s}$ Figure 5-6: Fluid side heat transfer rate v.s. fluid flow velocity

Fig. 5-7 demonstrates the variation of fluid-side heat transfer coefficient under different boundary conditions. The fluid side heat transfer coefficient increased with the flow velocity. It is noticed that even though the increasing air-side flow rate helped to increase the heat transfer rate, but the heat transfer coefficients did not vary significantly.



Figure 5-7: Fluid side heat transfer coefficients at different boundary conditions

For fully developed internal laminar flow, the Nusselt numbers are constant values which depend on the hydraulic diameter.

$$Nu = \frac{hD}{k} \tag{5-21}$$

However, the above equation is only valid for straight pipes. Therefore, for coiled pipes, it is of great necessity to derive another equation. For laminar phase change emulsion, the following dimensionless parameters are used to account for the effect of PCM particles under laminar flow conditions: average Reynolds number (Re), average Prandtl number (Pr) and Stefan number (Ste) [<u>110</u>]. A linear model was proposed based on the heat transfer coefficient data:

$$Nu = 1.26 \operatorname{Re}^{0.4} \operatorname{Pr}^{0.2} Ste^{-0.01} + 6.5$$
(5-22)

$$Ste = \frac{Cp_{eff} \Delta T}{L}$$
(5-23)

$$\Pr = \frac{Cp_{eff}\,\mu}{k} \tag{5-24}$$

Where,

L=latent heat

Cp_{,eff}= effective heat capacity of PCE

 ΔT =temperature difference

K=thermal conductivity

 μ =viscosity

Fig. 5-8 compares the experimental heat transfer results with those predicted by Eq. (5-23). All the data can be predicted with Eq. (5-23) with a standard deviation within $\pm 10\%$.



Figure 5-8: Comparison of measured and predicted Nusselt number for PCME

5.5 Comparison with numerical results

The pressure drop and heat transfer coefficient of developed PCE-10 and water were recorded and compared with the numerical results (obtained in Chapter 4).

Fig. 5-9 compares the pressure drop for both the experimental and the numerical results. The experiments were carried out at the almost isothermal condition with the coil insulated to prevent air flow across the heat exchanger. In CFD simulations, the effect of air flow on pressure drop across the heat exchanger was neglected. Therefore, the validation results were taken under conditions similar to the CFD model. Generally, the experimental result displayed similar trend as the numerical profile. But, at low flow rate (<0.2m/s), CFD tended to underestimate the pressure drop caused by fluid.



Figure 5-9: Pressure drop

Tab. 5-4 compares the simulation boundary conditions and experimental settings for heat transfer study. It is safe to say the simulation and experiments were performed under similar conditions and therefore, the results obtained were regarded as comparable.

	Air temperature	Air heat transfer coefficient	Fluid inlet temperature
Experimental settings	27-28°C	38.7, 49.9 59.3W/m ² K	7-8°C
Simulation settings	27°C	40, 50, 60W/m ² K	7°C

Table 5-4: Experiment and simulation settings

Fig. 5-10 shows the heat transfer coefficients of PCE-10 under different boundary conditions. The predicted values were in good agreement with that of the experiments. The differences in heat transfer coefficient number were all within $\pm 10\%$. In general, the errors were within an acceptable range and the results agreed well with the numerical predictions.



Figure 5-10: Heat transfer coefficient

5.6 Concluding remarks

The flow behaviour and heat transfer performance of PCME in fin-and-tube heat exchangers were evaluated experimentally. The results were used to validate the numerical prediction. The errors were within an acceptable range and the results agreed well with the numerical predictions.

The rheological performance of PCME in a coiled pipe was characterized by pressure drop across the test section. Measured pressure drop for PCE-10 was 3-6 times of that for water. The pressure drop for emulsion increased significantly with increasing in flow velocity. The friction factor for PCE-10 was determined by Equivalent Length Method and the correlation between friction factor and Reynolds number had been proposed.

Thermal performance of PCE-10 in FCU was evaluated under different air flow velocities and the results compared with water and result suggested that PCME did help to improve the heat transfer rate of heat exchanger by factor of 1.1-1.2 at the same flow rate compared with water. Relationships between Nusselt number and Reynolds number in coiled pipes have been proposed for further studies.

In the present study, the focus is on the application in air conditioning system where the PCME flow fell in the laminar regime. But in some other possible application areas such as waste heat recovery from a heat and steam generator, the flow velocity will be higher and flow is likely to be turbulent. Therefore, studies into PCME flow at higher Reynolds numbers are also recommended.

CHAPTER 6

EVALUATION OF PCE-10 IN AN AIR CONDITIONING SYSTEM

6.0 General

Chilled-water system is used widely in large-scale air-conditioning project around the world. Chilled-water system is an all water system. The chilled water is produced in the central plant and flows through the air-handling unit (AHU), where it absorbs sensible and latent heat from the passing air. The air is cooled and dehumidified after the heat transfer [111]. The AHU themselves are basically fan coil unit which are comprised of a finned-tube coil, an insulated drain pan under the coil to collect condensate, a fan to move air through the coil, filters, and a cabinet to house these components. An additional air distribution system may also be employed to distribute only fresh air to meet fresh air requirements [112].

In previous chapters, the flow rate and heat transfer characteristics of PCE-10 were investigated and found to be promising for application as a heat transfer medium in an air conditioning system within temperature range of 7-12°C. In this chapter, its performance in an air conditioning system was evaluated by using a whole building simulation software TRNSYS version 16. TRNSYS is a transient system simulation program with a modular structure. it includes a graphical interface, a simulation engine, and a library of components that range from various building models to standard Heating, Ventilation and Air Conditioning (HVAC) equipment to renewable energy and emerging technologies. The incorporation of weather files, control systems, HVAC systems, scheduling strategy and the modelling of multi-zone buildings makes TRNSYS a flexible tool in the evaluation of building energy systems.

In addition to TRNSYS an interface program called Simulation Studio 2006,

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version 4.0.0.0 was employed to facilitate the whole process. In Simulation Studio, the components are linked together and the input and parameters are given, which is very illustrative and easy to follow and there are several functions for using different colours or layers to distinguish between different types of connections like hydraulics, controls, weather or output.

6.1 Integrated phase change emulsion air conditioning system

6.1.1 Fan coil unit with phase change emulsion (FCU-P)

Fig. 6-1 shows the view of the self-assembled fan coil unit operating with phase change emulsion (named as FCU-P) considered for this study. It consists of a finand-tube heat exchanger and a circulation fan. The FCU-P is equipped with 3stage control over circulation fan at a ratio of 1: 0.75: 0.5 as requested by the national standard for Fan Coil Unit [113].



Fan

Figure 6-1: Self-assembled air handling unit

The phase change emulsion PCE-10 was used as the cooling medium. Listed in Tab. 6-1 are the simulation data for the PCE-10 which was extracted from Tab. 3-8 and 3-9. The heat capacities were calculated by using Eq. (6-1)

$$C_{p,m,eff}(T) = \begin{cases} X_{W}C_{p,w} + X_{PCM}C_{p,PCM} & T_{s} < T_{e} \\ X_{W}C_{p,w} + X_{PCM}C_{p,PCM} + X_{PCM}\Delta h / \Delta T & T_{e} < T_{s} < T_{c} \\ X_{W}C_{p,w} + X_{PCM}C_{p,PCM} & T_{c} < T_{s} \end{cases}$$
(6-1)

Where $C_{p,PCM}$ and Δh are respectively the specific heat of paraffin in the sold state and latent heat of fusion.

Properties	Measured	Unit	
Density	940	kg/m3	
Smaaifia Haat	T<4°C	3800	
Specific Heat	4°C ≤T≤11°C	8718	J/kgK
Capacity	11°C <t< td=""><td>3800</td><td></td></t<>	3800	

Table 6-1: Physical Properti	ies of PCE-10
------------------------------	---------------

The cooling capacity of the FCU-P was tested first under standard test condition as specified in Tab. 6-2. For comparison, the system with water (named as FCU-W) was also rated. The cooling capacity of a fan coil unit is the average of air-side and fluid-side capacities [113], which is calculated using Eq. (6-2).

$$Q_{FCU} = 0.5 \times (Q_W + Q_{Air}) \tag{6-2}$$

Which is considered reliable only if

$$\frac{\left|Q_{W}-Q_{Air}\right|}{Q_{FCU}} \le 5\% \tag{6-3}$$

Chapter 6

	Standard Test Condition [113]		
DCME volume flow rate	Determined by inlet/outlet temperature		
(l/hr)	difference		
	Requirement: $T_{outlet} - T_{inlet} = 5 \circ C$		
Inlet air temperature(°C)	27		
Inlet air velocity(m/s)	0.8, 1, 1.5		
Inlet PCME temperature	7		
(°C)	1		

Table 6-2: Standard test conditions

The design fluid flow rates for FCU-W and FCU-P are 304kg/h and 130kg/h, respectively, which fulfils the requirement of $T_{outlet} - T_{inlet} = 5$ °C for medium level. The complete technical data for FCU-P and FCU-W are listed in Tab. 6-3. Compared with FCU-W, it was found that FCU-P could cut the amount of cooling media by 50% but still achieve the same amount of cooling capacity.

	FCU-W			FCU-P		
-	High	Medium	Low	High	Medium	Low
Cooling capacity	1.72	1.57	1.39	1.60	1.59	1.44
(kW)						
Wind flow rate (kg/h)	516	345	258	516	345	258
Cooling media flow		304			130	
rate (kg/h)		504			150	
Heat transfer	243	283	278	246	303	275
coefficient (W/K)						
Pressure drop (Kpa)		30.04			23.8	

Table 6-3: Fan coil capacity rating under standard condition

6.1.2 System description

The concept of integrated PCME/air conditioning system is shown in Fig. 6-2.

The system consists of a central plant (chiller and cooling tower) and numbers of air handling unit FCU-P. The system operates with chilled PCE-10 at a temperature around 7-12°C. During working hours, phase change emulsion is directly circulated through a room air handling units by a centrifugal pump, and it completes the phase change process from solid to liquid and extracts heat from room, then it returns to central plant.



Figure 6-2: Schematic Diagram of Central Air Conditioning System

6.2 Modelling of air handling unit with PCE-10 (FCU-P)

6.2.1 TRNSYS model

The AHUs are basically fan coil units which are comprised of a finned-tube coil, an insulated drain pan under the coil to collect condensation, a fan to move air through the coil, filters, and a cabinet to house these components.

Therefore, it was modelled as a single equivalent unit made up of the following components:

• The cooling coil (a custom component Type 850d)

- A variable speed pump (Type 741)
- A variable speed fan (Type 662)

Pumps and fans can be modelled using standard TRNSYS components which have already developed by the TRNSYS Company. However, for all available heat exchangers in model library, specific heat capacities of hot side and cold side fluids are set as constant parameters. It is impossible to model phase change effect. Therefore, a custom component Type850d was built to simulate heat exchanger with variable heat capacities. Incorporating with a forcing function block, this Type can be used to model the cross flow heat exchangers working with fluids with variable heat capacities, such as phase change emulsion.

Type 850d relies on an effectiveness minimum capacitance approach to model a heat exchanger. Under this assumption, the user is asked to provide the heat exchanger's UA and inlet conditions. The model then determines whether the cold (load) or the hot (source) side is the minimum capacitance side and calculates effectiveness based upon the specified flow configuration and on UA. The heat exchanger outlet conditions are then computed, for all flow configurations using Eqs. (6-4) and (6-5). The capacitance of each side of the heat exchanger is calculated according to the following four equations. A schematic of the heat exchanger is shown in Fig. 6-3.

$$C_{\rm c} = m_{\rm c} C p_{\rm c} \tag{6-4}$$

$$C_{\rm h} = m_{\rm h} C p_{\rm h} \tag{6-5}$$



Figure 6-3: Heat Exchanger Schematic

Effectiveness of cross flow heat exchanger is calculated at each time step using Eq. (6-6) to Eq. (6-11). C_{max} and C_{min} are defined as the maximum value and minimum value of C_h and C_C , respectively.

If
$$C_h < Cc$$
, then, $C_{\min} = C_h$ and $C_{\max} = C_c$

$$\varepsilon = \frac{C_{\max}}{C_{\min}} \{1 - \exp[-1 - \exp(-\frac{UA}{C_{\min}})\frac{C_{\min}}{C_{\max}}]\}$$
(6-6)

If C_h >Cc, then, C_{max} = C_h and C_{min} = C_{c_s}

$$\varepsilon = 1 - \exp\left\{-\left[1 - \exp\left(-\frac{UA}{C_{\min}} * \frac{C_{\min}}{C_{\max}}\right)\right] \frac{C_{\min}}{C_{\max}}\right\}$$
(6-7)

The energy transfer in the coil and outlet temperatures of hot and cold fluid are

$$Q_{\max} = C_{\min}(T_{h,i} - T_{c,i})$$
 (6-8)

$$Q_{\text{total}} = Q_{\text{max}} \times \varepsilon \tag{6-9}$$

$$T_{h,o} = T_{h,i} - \frac{Q_{\text{total}}}{C_{\text{h}}}$$
(6-10)

$$T_{c,o} = T_{c,i} + \frac{Q_{\text{total}}}{C_c}$$
(6-11)

In Type850d, heat capacities of hot-side and cold-side fluid are modified with effective heat capacity method to model the phase change effect. The specific heat is given according to the temperature based on Eq. (6-1).

6.2.2 Simulation and experimental validation

Experiment setup and procedure

Before modelling a complete HVAC system, it is preferable to validate the component just developed with experimental result. Therefore, a small-size test bench equipped with a self-assembled fan coil unit was built.

The test facility is illustrated schematically in Fig. 6-4. The model room measures 1m x 0.8m x 1m. The full and exploded views of the complete system are shown in Figs. 6-5 and 6-6, respectively. 70% of the front surface is glazed with 12mm plain glass. The walls of the box were insulated with 12mm gypsum board which serves as thermal capacitance. The whole rig was housed inside a controlled laboratory environment which acts as a buffer controlling heat loss and gain through fabric. 1500W heat energy source was introduced to test the facility by using a heat convector and simulated internal heat gains (people and electrical equipment). K-type thermocouples were employed to measure the air and wall temperatures.



Figure 6-4: Schematic view of the air conditioning system



Self-assembled FCU

Figure 6-5: Photographic image of the test rig



Figure 6-6: Exploded view of the test box

The model room was equipped with a self-assembled fan coil unit operating with PCE-10 (named as FCU-P). During working times, cooling medium (PCE-10) was pumped from a storage tank, which was connected to a chiller.

The air-handling unit was set at Medium level with 100% fresh air during the test whilst the internal temperature of the test box was allowed to float between 25-28°C. The experimental parameters and TRNSYS inputs are listed in Tab. 6-4.

Parameters	Inputs	
Dimension	1*0.8*1m	
Thermal capacity	13 Wh/K	
Air flow rate	345 kg/h	
Specific heat of air	1000 J/kgK	
PCE-10 flow rate	130 kg/h	
	T<4°C	3800 J/kgK
Specific heat of PCE-10	4°C ≤T≤11°C	8718 J/kgK
	11°C <t< th=""><th>3800 J/kgK</th></t<>	3800 J/kgK
UA	303 W/K	
Pressure drop across FCU-W	23.8 kPa	

Table 6-4: TRNSYS input parameter for validation of experimental performance

Result and analysis

From repeated trials, the PCE-10 has been proved that it can effectively be used to replace water in air conditioning systems. It helps to reduce the volume flow rate of cooling medium whilst delivering same amount of cooling energy. To keep the box temperature around 27°C, 300kg/hr chilled water supply was required while only 130kg/h PCE-10 was needed to keep the box temperature at the same level. The average heat capacity for PCE-10 was calculated with the equation below as:

$$C_{p,e} = \frac{C_{p,air}m_{air}(T_{air-in} - T_{air-out})}{m_e(T_{e-out} - T_{e-in})}$$
(6-12)

The average heat capacity of PCE-10 in the working temperature range of 7-12°C is 8.67kJ/kgK, which was slightly lower than predicted number of 8.718kJ/kgK, but it was twice of water's specific heat.

Fig. 6-7 shows the comparisons between the experimental data and the numerical modelling for AHU performance and indoor room temperature. Overall, the

numerical modelling was in good agreement with the experimental data. The mean difference was within 0.2°C. The biggest difference occurred in the early stage of cooling. It actually took longer time for temperature differences across FCU-P to be stabilized than prediction.



Figure 6-7: Performance recording of model air conditioning system with PCME (Experiment and simulation)

6.3 Simulation of a typical HVAC system with PCE-10

6.3.1 Model description

Building description and Cooling load

In order to evaluate the performance of the PCE-10 in an air conditioning system,

a typical office building in a south-facing high rise building in Ningbo, China

(N29° 49' 19" E121° 31' 15") was considered. The reference year weather data for

Ningbo was based on a 1982-1997 period of record obtained from the Chinese

National Climatic Data Centre [114].

The geometry of the office room used in this analysis is illustrated in Fig. 6-8. It is a light-weight construction measuring 4.5m high by 4.5m wide and 3.6m deep with only one external facade of 70% glazing. The constructions and physical properties of building components are specified in Tab. 6-5. Windows are single-glazed (south facing) and the U-values follow those set by the National Standard for the region [115] for Type B (<2000m² of total floor area).



Figure 6-8: Office room (shaded) to be simulated in TRNSYS model

	Interior Wall	External Wall	Window	Floor	Ceiling
Construction	Light steel keel gypsum board	Reinforced concrete wall	Single glazing flat glass	Reinforced concrete wall	Reinforced concrete wall
Thickness (m)	0.008	0.3	0.012	0.2	0.2
U-value (W/m2K)	1.78	1.18	5.5	1	0.7
Density (kg/m3)	1100	470	2580	620	620
Thermal conductivity (W/mK)	0.68	0.19	1.05	0.2	0.2
Heat capacitance (Wh/m2K)	2.8	5	-	4	4

Table 6-5: Physical properties of office room structures to be simulated

Building thermal capacitance describes the ability of building component to store heat, providing inertia against temperature fluctuations. Building thermal capacitance includes the capacitance of building material and furnishings. In accordance to the HVAC system design manual [102], the average internal thermal capacitance for office building is 4.2Wh/m²K and the total thermal capacitance of studied office room is calculated as:

$$\begin{split} C &= C_{v,wall} \times S_{wall} + C_{v,furnitur} \times S + C_{v,air} \times V_{room} \\ &= 4Wh/m^2 K \times (4.5m \times 3.6m) + 4Wh/m^2 K \times (4.5m \times 3.6m) + 5Wh/m^2 K \times (4.5m \times 4.5m \times 30\%) \\ &+ 2.8Wh/m^2 K \times (4.5m \times 3.6m \times 2 + 4.5m \times 4.5m) \\ &+ 4.2Wh/m^2 K \times (4.5m \times 3.6m) + 0.36Wh/m^2 K \times (4.5m \times 4.5m \times 3.6m) \\ &= 400Wh/K \end{split}$$

The building is occupied daily from 07:00 to 18:00. The cooling load equals to the

sum of internal gains, such as lighting, equipment and occupants, and solar radiation. For simplification, the cooling load was estimated by using cooling load index for Chinese office buildings [<u>116</u>]. For normal office buildings, cooling load per square meter is 125W. The average cooling load is therefore,

$$125W/m^2 \times 4.5m \times 3.6m = 2000W$$

HVAC system

The 100% cooling load was covered by a central air conditioning system. The central air conditioning system was switched on from 06:00 to 18:00 in summer months (June to August). The complete TRNSYS system model is shown graphically in Fig. 6-9. The developed custom component Type 850d was used to describe the AHUs and generic TYPES were used for all other components. Standard component models have been introduced in details later.

Two separate cases were simulated in this study. **Case 1**, as a control group, simulated a conventional air conditioning system with water as a cooling medium. In **Case 2**, the structure of the system was kept the same as in Case 1, but the phase change material emulsion PCE-10 was used as the cooling medium instead of water.

With the aim of approaching the reality, the design of the two air conditioning systems were based on the guidelines in the Practical heating and air conditioning system design manual [116].

• Determination of cooling load.

The design cooling load is 2000W.

• Determination of fresh air requirement

In office building, the room area requirement is $4m^2$ /person and a minimal $30m^3$ /h/person of fresh air is required. In a 4.5*3.6m office room, the number of occupant is 4 persons and it will need 120m3/h (154.8kg/h) fresh air.

• Determining the number of AHU

The technical data of the self-assembled FCU-P was used for this study. The number of AHUs is related with the cooling load and cooling capacity of AHU at medium level. Thus, the number of AHU required is

$$n = \frac{2000W}{1570W} = 1.27 \approx 2$$
 For FCU – W
 $n = \frac{2000W}{1590W} = 1.25 \approx 2$ For FCU – P

Two AHUs can therefore supply 690kg/h fresh air in total, which meets the required fresh air supply without the need for additional ventilation system.

• Determining the size of other primary equipment

Based on the cooling load and number of AHUs, following primary equipment was selected accordingly, which are listed in Tab. 6-6.



Figure 6-9: TRNSYS Model for Central Air Conditioning System
Chiller	
Model No.	Carrier 19XR 742kW/5.42COP/VSD
Chilled water set point	7°C
Rated capacity	742kW
Rated COP	5.42
Specific hat of CHW	4.19kJ/kgK for water
	8.7kJ/kgK for PCE-10 at 4-12°C
Specific hat of CW	4.19kJ/kgK
Cooling Tower	
Model No.	AEC cooling tower model FG2003
Design air flow rate	5000kg/hr
Rated power	4000kJ/hr
Pump	
Overall Efficiency	0.6
Motor efficiency	0.9
Motor heat loss fraction	1
Fan	
Fan power (W)	134
Motor efficiency	0.8
Overall Efficiency	0.6
Motor efficiency	0.9

Table 6-6: Technical data for primary equipment from TRNSYS 16 [117]

6.3.2 Simulation component

Apart from the custom component Type 580d, other components used in this model as well as their mathematical models, are described in this section

The building model

The building was modelled in TRNSYS16 as an energy rate control building (Type690). This component models a single node lumped capacitance using a differential equation. It takes inputs of heating and cooling loads, and turns those loads into a modulating temperature. The component also has inputs for the

conditions of ventilation streams, which may by conditioned to control the modulating temperature. The structure is subject to internal gains but solar gains and building loss coefficient are taken as negligible. Type690 shows its usefulness in modelling new HVAC systems for buildings whose heating and cooling loads have already been obtained [118].

The component Type690 is governed by two balance equations. An energy balance that predicts the zone temperature and moisture balance that predicts the humidity content of the zone.

The energy balance for the zone is

$$\frac{dT}{dt} = \frac{m_{vent}Cp_{vent}}{cap}(T_{vent} - T_{initia}) + \frac{Q_{\text{cooling}}}{cap}$$
(6-13)

The moisture balance equation is similar in form to the energy balance:

$$\frac{d\omega}{dt} = \frac{m_{vent}}{\operatorname{cap}_{moissture}} (\omega_{vent} - \omega_{initia}) + \frac{Q_{\text{cooling}}}{\operatorname{cap}_{moissture}}$$
(6-14)

The chiller model

The chiller was modelled as a vapour compression style water cooled chiller (Type666). Type666 relies on a catalogue data lookup method to predict the performance of a vapour compression style water cooled chiller. Because of the data lookup approach, this component may be equally well used to model single and multi-stage chillers [117]. To set up the model, the user must provide two text based data files. The first of these files provides the chiller's performance data including capacity ratio and COP ratio for varying values of chilled water set

point temperature (in°C), and for varying entering cooling water temperatures (in °C). The second data file, part-load performance data, provides fraction of full load power for varying values of part load ratio. The data files were constructed from the performance data for the ElectricEIRChiller Carrier 19XR 742kW/5.42COP/VSD. A schematic diagram showing a single stage water cooled chiller is shown in Fig. 6-10.



Figure 6-10: Schematic diagram of a single stage water cooled chiller.

At a given time step, Type666 first performs a call to the performance data routine with the current cooling water (sink) temperature and the chilled water set point temperature, obtaining in return the COP ratio and capacity ratio for those conditions. The chiller's nominal COP and the capacity at current conditions are calculated:

$$COP_{nom} = COP_{rated} * COP_{ratio}$$
 (6-15)

$$Capacity_{nom} = Capacity_{rated} * Capacity_{ratio}$$
(6-16)

The chillers nominal COP and Capacity ratio is calculated by performing a call to the performance data with the current cooling water (sink) temperature and the chilled water set point temperature.

TRYNSYS 16 then calculates the chiller's load using

$$Q_{\text{load}} = m_{\text{chw}} * C p_{\text{chw}} (T_{chw,i} - T_{chw,set})$$
(6-17)

The Part Load Ratio (PLR) was calculated by:

$$PLR = \frac{Q_{\text{load}}}{Capacity} \tag{6-18}$$

The load met by the chiller is limited to the capacity of the machine if the calculated PLR is greater than unity. Once a valid PLR has been calculated (between 0-1), the Dynamic Data was called upon again, this time to specify the second data file, and the chiller power draw was given by:

$$P = \frac{Capacity}{COP_{nom}} FFLP \tag{6-19}$$

Where,

FFLP=Fraction of full load power

A corrected COP was then calculated as:

$$COP = \frac{Q_{\text{met}}}{P} \tag{6-20}$$

The energy rejected to the cooling fluid stream by the chiller was:

$$Q_{\text{rejected}} = Q_{\text{met}} + P \tag{6-21}$$

The outlet temperature of the chilled fluid stream is:

$$T_{\rm chw,o} = T_{\rm chw,i} + \frac{Q_{met}}{m_{chw}Cp_{chw}}$$
(6-22)

The outlet temperature of the cooling fluid stream is:

$$T_{\rm cw,o} = T_{\rm cw,i} + \frac{Q_{met}}{m_{cw}Cp_{cw}}$$
(6-23)

The cooling tower model

Type 510 closed circuit cooling tower was used to model the cooling tower where the working fluid (CW) circuit was separated from the ambient air and the process water. The TRNSYS simulation code was based on the model proposed by Zweifel [119] that matches the manufacturers catalogue data over a wide range of operating conditions. The performance evaluation was based on AEC cooling tower model FG2003. This model relies on the basic premise that the saturated air temperature is the temperature at the air-water interface and is also the temperature of the outlet fluid. With this assumption Zweifel [119] demonstrated that the saturated air enthalpy can be calculated from:

$$h_{\text{sat}}(T_{fluid,o}) = h_{air}(T_{air,i}) + \frac{Q_{fluid}}{m_{air}\{1 - \exp[-\lambda_{design}\psi_{air}^{y-1}]\}}$$
(6-24)

Where y is empirical constant, 0.6

$$\gamma_{air} = \frac{m_{air}}{m_{air,design}} \tag{6-25}$$

$$\lambda_{design} = \ln\left[\frac{h_{\text{sat}}T_{fluid,o,design} - h_{\text{air}}T_{air,i,design}}{h_{\text{sat}}T_{fluid,o,design} - h_{\text{air}}T_{air,o,design}}\right]$$
(6-26)

$$h_{\text{air}}T_{air,o,design} = h_{\text{air}}T_{air,i,design} + \frac{Q_{fluid,design}}{m_{air}}$$
(6-27)

The power drawn by the fan in operation is given by Eq. (6-28). Where a is coefficient defined by users.

$$P_{fan} = P_{fan, rated}[a_0 + a_1(\psi_{air}) + a_2\psi_{air}^2 + \dots]$$
(6-28)

For simplification in this study, the above equation is solved as:

$$P_{fan} = P_{fan, rated} \times \psi_{air} \tag{6-29}$$

The fan model

Variable speed fan Type 662 is able to turn at any speed between 0 (full stop) and its rated speed. While the mass flow rate of air moved by the fan is linearly related to the control signal, the power drawn by the fan at a given flow rate can be any polynomial expression of the control signal.

The power drawn by circulation fan was calculated using Eq. (6-30) as defined in Design standard for energy efficiency of public buildings [120]

$$P_{circulaton_fan} = P' \times V_{air}$$
(6-30)

P' is the power consumption of unit air volume, which is 0.6W/(m3/h).

The resulted energy transferred to the fan air stream was calculated using Eq. (6-31).

$$Q_{\text{air}} = P_{\text{circulation}_{fan}} [\eta_{\text{motor}} + (1 - \eta_{\text{motor}}) f_{motorlos}]$$
(6-31)

Where η_{motor} is the motor efficiency and $f_{motorloss}$ is a value between 0 and 1 which determines whether the fan motor inefficiencies cause a temperature rise in the air

stream that passes through the fan or whether they cause a temperature rise in the ambient air.

The mass flow rate of air delivered by the fan at any given time is linearly related with the control signal as shown in Eq. (6-32).

$$m_{\rm air} = \psi_{air} m_{air,rated} \tag{6-32}$$

The enthalpy of air exiting the fan can be calculated as shown in Eq. (6-33).

$$h_{o} = h_{i} + \frac{Q_{air}}{m_{air}}$$
(6-33)

The pump model

There are two pumps in this cooling system, chilled pump and cooling water pump. Chilled pump transport chilled cooling medium from chiller to RACU while cooling water pump delivers cooling water between chiller and cooling tower. These pumps were modelled with variable speed pump Type741.

Type741 models a variable speed pump that is able to produce any mass flow rate between zero and its rated flow rate. The pump's power was calculated from pressure rise, overall pump efficiency, motor efficiency and fluid characteristics. However, pump starting and stopping characteristics were not modelled. Type741 takes mass flow rate as an input but ignores the value except in order to perform mass balance checks. Type 741 also sets the downstream flow rate based on its rated flow rate parameter and the current value of its control signal input.

The total power drawn by the pump therefore includes the effects of motor inefficiency and inefficiency of the pumping process as shown in Eq. (6-34).

$$P = \frac{P_{shaft}}{\eta_{motor}} = \frac{W_{pumping}}{\eta_{pumping}\eta_{motor}} = \frac{\Delta p \times m_{fluid}}{\rho_{fluid} \times \eta_{motor} \times \eta_{pumping}}$$
(6-34)

The pumping efficiency was given by the equation below; both the motor and overall efficiency of the pump were entered by the user as Inputs to the model.

$$\eta_{\text{pumping}} = \frac{\eta_{overall}}{\eta_{motor}}$$
(6-35)

Energy transferred from the pump motor to the fluid stream and the resulted temperature of fluid exiting the pump can be calculated as

$$Q_{\text{fluid}} = P_{\text{shaft}} (1 - \eta_{\text{pumping}}) + (P - P_{\text{shaft}}) f_{\text{motorloss}}$$
(6-36)

$$P_{\text{shaft}} = P \times \eta_{\text{motor}} \tag{6-37}$$

$$T_{fluid,out} = T_{fluid,in} + \frac{Q_{\text{fluid}}}{m_{fluid}}$$
(6-38)

Where

 $\eta_{pumping} = pumping \text{ process efficiency}$

 $f_{motorloss}$ is a value between 0 and 1 which determines whether the pump motor inefficiencies cause a temperature rise in the fluid stream that passes through the pump or the ambient air surrounding the pump.

If the pump has an inline motor, in which case all waste heat would impact the fluid stream temperature and $f_{motorloss}$ would have a value of 1, or whether the pump motor is housed outside of the fluid stream such that it's waste heat impacts the ambient and $f_{motorloss}$ would have a value of 0.

Other component model

Another important component is the weather data processor that reads a weather

data file. In order to link whole system and incorporate features of PCME, controllers and equation blocks were required. The equation blocks give the user the ability to define equations within the input file that are not in a component. The equations can then be used as inputs to other components, or as parameters, initial values of inputs, or derivatives if the equations are time independent [117]. One main equation block was used in the simulation to make changing certain parameters easier and is especially useful for enabling free cooling in one easy step. The output files from TRNSYS16 were put into Excel so as to be analysed more in depth and to generate graphs. Please refer to Appendix 2 for the complete TRNSYS input file.

6.2.1 Result and discussion

Cooling performance

The performances of two air handling units have been simulated based on weather data for July 31st (the hottest day). Fig. 6-11 shows the performances of FCU-W and FCU-P. Air conditioning system started working at 6am (5070hour). The performances of two systems were fairly close. Temperatures of inlet air and fluid were found increased by 2°C and 0.2°C before entering the FCUs due to heat reject from the fan and pump. Fluid side temperature increased by roughly 4.8°C and 5.1°C, in FCU-P and FCU-W, respectively. Air was effectively cooled from 30°C to 13.6°C and 13.8°C for FCU-P and FCU-W, respectively. It was noticed that the effectiveness of FCU-P (0.83) was slightly lower than that of FCU-W (0.87), but, the two systems achieved similar cooling performance.





The performances of two complete systems are presented in form of comparison between office room equipped with chilled water system (Case 1) and one with chilled PCE-10 air conditioning system (Case 2).

The comparisons of hourly indoor air temperatures of Case 1and Case 2 are presented in Fig. 6-12 for a hottest week (July 20th to 26th). In both cases, the room temperature did not vary significantly with outdoor temperature. As the air conditioning system started working 1 hour early than the other equipment, the room temperature dropped to 18°C. Then it gradually rose to 20°C due to heat input from office equipment. At night when the air conditioning system was off, the heat capacitance inside the office released heat and kept the temperature at 21°C.

In Case 1, 608kg/h of chilled water was used to keep the room temperature at 20°C during day time when the air conditioning system is operating. In Case 2, it only took 260kg/h to achieve the same cooling effect.



Heat transfer rates

Temperatures

(a) Case 1: Conventional air conditioning system Figure 6-12: Indoor temperature comparison for a hottest week (July 20th to 26th))



(b) Case 2: Integrated air conditioning system Figure 6-12: Indoor temperature comparison for a hottest week (July 20th to 26th)

Energy consumption

The energy consumptions of Case 1 and Case 2 for summer months (June to August) are breakdowns in Fig. 6-13.

Case 1 consumed 4036MWh of energy in a 3-month operation, among which, 53% of energy was consumed by the chiller followed by 26% in the refrigerant distribution system (i.e. pumps). Chilled water pump power alone was responsible for 6.5% of total energy usage. In case 2, total energy consumption was 3751MWh, a reduction by 7% and the energy used by chilled water pump was reduced significantly, by roughly 50%.



Figure 6-13: Breakdown of energy consumption for summer period

6.4 Concluding remarks

In this chapter, the idea of applying developed PCE-10 into an air conditioning system as a cooling medium was explored, and the performance of PCE-10 in air conditioning system was evaluated with a transient system simulation program TRNSYS.

An integrated PCE/fan coil unit operating with PCE-10 has been assembled in lab. The cooling capacity for integrated PCE/ air handling unit (named as FCU-P) and a normal AHU with water (named as FCU-W) have been rated under the standard test conditions. FCU-P has a rated cooling capacity of 1.6kW and 516kg/h rated air flow rate. Compared with normal FCU, it was found that FCU-P could cut the amount of cooling media by 50% but still achieve the same amount of cooling capacity.

As the current TRNSYS components failed to model the performance of phase change emulsion in cooling coils, a new component was therefore developed. The custom component uses the effectiveness-NTU method to model the heat exchanger and effective heat capacity method to incorporate into the heat exchanger model by a force function. It has been validated that this component could model the cooling effect of FCU-P.

By incorporating the technical data of integrated PCE/air handling unit into the TRNSYS system, the annual energy consumptions of integrated PCME/air conditioning system and normal air conditioning system were analysed based on a model room located in Ningbo, China. The simulation results showed that the

PCME/air conditioning system could achieve the same cooling effect as the normal chilled water system with 50% less cooling media and 7% reduction in energy consumed by the air conditioning system.

CHAPTER 7

CONCLUSIONS AND FURTHER WORK

7.1 Conclusions

Global energy consumption has been increasing steadily over the past decades due to population growth and rapid urbanisation in developing countries such as China and India. This upward trend has led to growing demand for air conditioning systems in the building sector which dominates the energy consuming sectors. There is therefore the need for more energy efficient cooling systems since the demand is likely to continue for foreseeable future. Development and application of phase change material emulsions (PCMEs) in air conditioning systems has been identified as one of the key solutions towards improving the energy utilization efficiency and minimisation of energy consumption in the building sector.

PCMEs are binary systems where paraffin droplets are dispersed in water by a surfactant. PCME can work as a cooling media as well as a cold storage material simultaneously in air conditioning systems and offers many advantages including:

- Lower pumping power, as a consequence of the reduction in mass flow due to higher heat capacity,
- PCMEs do not lose the fluidity during phase transition due to the carrier fluid
- 3) High heat transfer rate due to enlarged surface-to-volume rate

However, there are some issues including sub-cooling, instability and other heat transfer characteristics which still affect the thermal performance of PCMEs which need to be addressed before they could be replace conventional chilled water systems. A comprehensive review was therefore carried out to establish the characteristics, challenges and advancement in the development and application of PCM emulsions in HVAC systems. The review also covered the composition, preparation methods and analysis of properties of various PCMEs. In general the PCMEs were found to be capable of significantly improving the heat transfer rate under laminar and turbulent flows. However, most of the developments did experience scientific and technical problems such as low thermal stability during a long-term storage period when subjected to mechanical–thermal loadings and high level of sub-cooling which tends to deteriorate the performance of refrigeration equipment.

To this end a novel paraffin-in-water phase change material emulsion (PCE-10) capable of minimising the effects of those highlighted barriers was developed and evaluated. The emulsion contained 25wt% of RT10 with a phase change range of 4-12°C and a non-ionic surfactant group of Tween 60 and Brij 52 with a HLB value of 10.9. To ensure stability in the emulsion, an estimated amount of a suitable surfactant (3.75 wt%) was also added to the emulsion. Analysis of the thermophysical properties revealed the particle size of the PCE-10 sample to be 3μ m with a heat capacity of twice as much as that of water and a negligible subcooling degree of 0.2°C. The test conducted on the sample achieved a good period stability for 9 months in storage and was able to withstand over 500 freeze-thaw cycles in a pumping system without any significant sign of degradation to particle size and heat storage capacity. The viscosity was however found to be 13 times higher than that of water, which could increase energy consumption in the

pumping system.

It was also noticed from the review that the lack of knowledge of PCME's flow behaviour and heat transfer characteristics in heat exchangers restricted the design of integrated PCE/ air conditioning system. Therefore, theoretical investigation into the forced convective heat transfer characteristics of phase change emulsion flow in a fin-and-tube heat exchanger, (which is the commonest type used in HVAC systems) was performed with computational fluid dynamics software ANSYS FLUENT. The results of the heat transfer parameters and rheological performance were then compared with results from a chilled water system under similar operating conditions. Pressure drop, Nusselt number and wall temperature were used to describe the flow behaviour and heat transfer enhancement in the phase change emulsion.

It was found that under identical conditions, PCE-10 did help to improve heat transfer rate by a factor of 1.1-1.3 in comparison with water, however, pressure drop for PCE-10 was obtained as 2-10 times higher than that of water. At high flow velocity, PCE-10 pumping power was reduced in comparison to water when delivering the same level of heat. However, at lower flow velocity, the pump power consumption of PCE-10 would still be higher than water. Simulation work also suggested the necessity of thermal conductivity enhancement as additional 10% increase in heat transfer rate could be achieved if the conductivity was increased to 0.6W/mK.

The heat transfer and pressure drop investigations into the laminar phase change

emulsion flow in the fin-and-tube heat exchanger have been presented in Chapter 5. An experimental rig for the determination of the pressure drop and heat transfer performance of a commercially available fin-and-tube heat exchanger using PCE-10 as a coolant was set up. High accuracy sensors were used to measure key parameters including inlet and outlet temperatures, flow rates and pressure drop throughout the test section. Thermal performance of PCE-10 was evaluated under different air flow velocities and the results compared with water. In general the experimental results were in agreement with the numerical predictions.

Concluding from the experimental studies, the measured pressure drop for PCE-10 was 3-6 times of that for water. The pressure drop for emulsion increased with increasing flow velocity. In addition, it was found that PCE-10 did help to increase the total heat transfer rate and seems to be more suitable at lower heat transfer coefficient while water based system was performing better with high heat transfer coefficient. Last but not least, the friction factor for PCE-10 in coiled pipes was determined, and the relationship between friction factors and Reynolds number has been derived for further studies.

In order to assess the potential of the developed PCE-10 sample, it was integrated into a model air conditioning system and theoretically evaluated as presented in Chapter 6. The integrated PCE-10/air handling unit (named as FCU-P) had a rated cooling capacity of 1.6kW and flow rate of 516kg/h. The technical data were therefore incorporated into a transient system simulation program TRNSYS which enabled its annual energy consumptions to be analysed and compared with that of a conventional air conditioning system for a typical office space location in Ningbo, China as a test case. The simulation results showed that the PCE/air conditioning system could achieve the same cooling effect as the normal chilled water system with 50% less cooling medium which represents a 7% reduction in energy consumption.

In conclusion the study has demonstrated the potential of the developed PCE-10 as a cooling medium for application in air conditioning systems. There is however the need for further improvements in the areas of thermal conductivity enhancement and reduction in pressure drop in order to establish its full potential.

7.2 **Recommendation for future studies**

In view of the conclusions the following areas are recommended for future investigations:

The viscosity of the PCE-10 was found to be much higher than water which could contribute to high pressure drop in a pumping system. Its thermal conductivity was also found to be about 30% lower than the value of water which could influence heat transfer processes. Further enhancements of these thermophysical properties are therefore recommended

Investigations into flow behaviour at higher Reynolds numbers are also recommended. This is because the study was limited to low velocity fan coil units which do not sufficiently meet the requirements for wide engineering applications.

The final recommendation is to conduct a full scale experiment on a real size air conditioning system covering a full life cycle analysis.

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APPENDIXES

Flow rate	Cooling capacity	Fluid side temperature difference	Air side temperature difference	LMTD	F	Fin efficiency
l/hr	kW	Cel	Cel			
144	1.65	9.90	15.62	13.43	0.85	0.88
136	1.61	10.21	15.29	13.49	0.85	0.88
128	1.57	10.57	15.71	13.26	0.85	0.88
120	1.53	11.01	15.12	13.10	0.85	0.88
112	1.49	11.44	14.92	13.11	0.85	0.88
104	1.44	11.89	15.86	13.14	0.85	0.88
96	1.39	12.44	14.57	12.50	0.85	0.88
88	1.32	12.89	13.92	12.50	0.80	0.88
80	1.26	13.51	13.42	12.03	0.80	0.88
72	1.19	14.24	13.85	12.11	0.84	0.88
64	1.11	14.92	13.53	11.76	0.85	0.88
56	1.02	15.69	13.78	11.96	0.85	0.88
48	0.92	16.54	12.20	10.57	0.85	0.88
40	0.81	17.37	14.24	12.31	0.85	0.88

Appendix 1: Test records for heat transfer performance of PCME in coiled pipes Table A1 (a): Test records for water heat transfer performance at air flow velocity 1.5 m/s

Flow rate	Cooling capacity	Fluid side temperature difference	Air side temperature difference	LMTD	F	Fin efficiency
l/hr	kW	Cel	Cel			
144	1.79	6.00	17.11	14.67	0.90	0.88
136	1.74	6.33	16.67	14.01	0.90	0.88
128	1.73	7.09	16.91	14.21	0.90	0.88
120	1.67	7.47	16.12	13.83	0.90	0.88
112	1.62	8.03	15.13	13.23	0.88	0.88
104	1.58	8.71	15.58	13.26	0.88	0.88
96	1.52	9.34	14.90	12.57	0.88	0.88
88	1.46	10.15	13.89	12.25	0.88	0.88
80	1.40	11.00	13.99	11.87	0.88	0.88
72	1.32	11.93	12.97	11.29	0.88	0.88
64	1.25	13.05	12.57	10.80	0.95	0.88
56	1.16	14.34	11.35	10.24	0.95	0.88
48	1.06	15.67	10.66	9.51	0.95	0.88
40	0.95	17.21	10.90	9.47	0.95	0.88

Table A1 (b): Test records for PCME heat transfer performance at air flow velocity 1.5 m/s
Flow rate	Cooling capacity	Fluid side temperature difference	Air side temperature difference	LMTD	F	Fin efficiency
l/hr	kW	Cel	Cel			
144	1.50	8.95	17.07	14.56	0.70	0.90
136	1.46	9.25	17.48	14.54	0.70	0.90
128	1.43	9.61	17.03	14.18	0.80	0.90
120	1.40	10.07	17.34	14.05	0.80	0.90
112	1.36	10.44	17.03	14.12	0.85	0.90
104	1.32	10.91	16.42	13.44	0.80	0.90
96	1.27	11.43	16.27	13.70	0.85	0.90
88	1.23	12.02	15.10	13.00	0.80	0.90
80	1.17	12.61	14.70	12.93	0.80	0.90
72	1.10	13.21	14.71	12.23	0.85	0.90
64	1.04	14.00	13.82	11.80	0.85	0.90
56	0.96	14.80	13.81	11.62	0.85	0.90
48	0.88	15.75	13.25	11.27	0.85	0.90
40	0.78	16.75	13.28	11.56	0.90	0.90

Table A1 (c): Test records for water heat transfer performance at air flow velocity 1 m/s

Flow rate	Cooling capacity	Fluid side temperature difference	Air side temperature difference	LMTD	F	Fin efficiency
l/hr	kW	Cel	Cel			
144	1.67	5.20	18.42	15.24	0.75	0.90
136	1.61	5.45	18.93	15.44	0.75	0.90
128	1.56	5.83	17.93	15.06	0.85	0.90
120	1.51	6.20	17.47	14.40	0.85	0.90
112	1.46	6.70	16.95	14.23	0.85	0.90
104	1.43	7.38	16.21	13.80	0.85	0.90
96	1.38	8.02	15.91	13.43	0.80	0.90
88	1.33	8.80	15.17	13.02	0.80	0.90
80	1.28	9.65	13.83	12.34	0.80	0.90
72	1.32	12.07	14.77	13.05	0.85	0.90
64	1.16	11.72	12.84	11.28	0.85	0.90
56	1.08	13.03	12.19	10.84	0.85	0.90
48	0.99	14.41	11.46	10.12	0.85	0.90
40	0.89	16.04	10.85	9.72	0.90	0.90

Table A1(d): Test records for PCME heat transfer performance at air flow velocity 1 m/s

Flow rate	Cooling capacity	Fluid side temperature difference	Air side temperature difference	LMTD	F	Fin efficiency
l/hr	kW	Cel	Cel			
144	1.16	6.95	18.11	14.83	0.80	0.92
136	1.15	7.25	17.35	14.68	0.80	0.92
128	1.13	7.61	17.71	14.48	0.80	0.92
120	1.12	8.07	17.81	14.60	0.80	0.92
112	1.10	8.44	17.96	14.43	0.85	0.92
104	1.08	8.91	17.28	14.42	0.85	0.92
96	1.05	9.43	16.98	13.98	0.80	0.92
88	1.02	10.02	16.17	14.03	0.80	0.92
80	0.99	10.61	15.75	13.65	0.85	0.92
72	0.94	11.21	16.39	13.54	0.85	0.92
64	0.89	12.00	15.10	12.97	0.90	0.92
56	0.83	12.80	15.35	13.16	0.90	0.92
48	0.77	13.75	15.06	12.72	0.90	0.92
40	0.68	14.75	14.82	12.61	0.90	0.92

Table A1(e): Test records for water heat transfer performance at air flow velocity 0.8 m/s

Flow rate	Cooling capacity	Fluid side temperature difference	Air side temperature difference	LMTD	F	Fin efficiency
l/hr	kW	Cel	Cel			
144	1.48	4.52	23.75	18.55	0.90	0.92
136	1.47	4.75	24.48	18.67	0.90	0.92
128	1.46	5.02	24.07	18.63	0.88	0.92
120	1.44	5.28	24.61	18.40	0.88	0.92
112	1.34	5.61	22.56	17.18	0.88	0.92
104	1.29	6.05	21.57	16.61	0.88	0.92
96	1.25	6.61	20.38	15.94	0.90	0.92
88	1.21	7.41	19.36	15.80	0.90	0.92
80	1.17	8.23	18.08	15.19	0.85	0.92
72	1.12	9.14	17.62	14.33	0.85	0.92
64	1.07	10.30	16.48	13.76	0.85	0.92
56	1.01	11.63	15.12	13.29	0.85	0.92
48	0.93	13.05	14.66	12.61	0.85	0.92
40	0.84	14.75	13.41	11.64	1.00	0.92

Table A1(f): Test records for PCME heat transfer performance at air flow velocity 0.8 m/s

(TIME XIN OUTPT DTI	SUBROUTINE T	YPE850
C*****************	***************************************	:
C Object: Cross flow HX C IISiBat Model: Type85	for PCME 0d	
C C Author: Jingjing SHAO)	
C Editor: C Date: January 13, 201	5 last modified: January 13, 2015	
C Date. January 15, 201	s last mounted. Sandary 15, 2015	
С		
C ***		
C *** Model Parameters		
C ***	Cross flow mode [Inf+Inf]	
C ***	closs now mode - [-m,+m]	
C *** Model Inputs		
C ***		
C	Hot side inlet temperature C	
C	Hot side flow ratekg/hr [-Inf;+Inf]	
C	Cold side inlet temperature C [-Inf;+Inf]	
C	Specific heat of cold side flow kI/kg K [-Inf:+Inf]	
C	Overall heat transfer coefficient kJ/hr.K [-Inf:+Inf]	
C	Specifc heat of hot side flow kJ/kg.K [-Inf;+Inf]	
C ***		
C *** Model Outputs		
C ***		
C	Hot-side outlet temperature C [-Inf;+Inf]	
С	Hot-side flow rate kg/hr [-Inf;+Inf]	
C	Cold-side outlet temperature C [-Inf;+Inf]	
C	Cold-side flow rate kg/hr [-Inf;+Inf]	
C	Heat transfer rate kJ/hr [-Inf;+Inf] Effectiveness - [-Inf;+Inf]	
C ***		
C *** Model Derivatives		
C (Comments and routine	: interface generated by TRNSYS Studio) ************************************	¢
C		
C TRNSYS acess functi	ions (allow to acess TIME etc.)	
USE TrnsysConstants		
USE THISYSFUNCTIONS		
C		
C REQUIRED BY THE	E MULTI-DLL VERSION OF TRNSYS	
!DEC\$ATTRIBUTES	S DLLEXPORT :: TYPE850 !SET	THE
CORRECT TYPE NUME	3ER HERE	
C		
C TRNSYS DECLARA	TIONS	
IMPLICIT NONE	!REQUIRES THE USER TO DEFINE	E ALL
VARIABLES BEFORE U	JSING THEM	
DUIBIE DEC	USION XIN THE ARRAY FROM WHICH THE INDI	ITS TO
DOUDLE I KEU		15 10
	201	

Appendix 2: Source Code for TRNSYS TYPE850d

THIS TYPE WILL BE RETRIEVED
DOUBLE PRECISION OUTP !THE ARRAY WHICH WILL BE USED TO STORE
THE OUTPUTS FROM THIS TYPE
DOUBLE PRECISION TIME !THE CURRENT SIMULATION TIME - YOU MAY
USE THIS VARIABLE BUT DO NOT SET IT!
DOUBLE PRECISION PAR !THE ARRAY FROM WHICH THE PARAMETERS
FOR THIS TYPE WILL BE RETRIEVED
DOUBLE PRECISION STORED !THE STORAGE ARRAY FOR HOLDING
VARIABLES FROM TIMESTEP TO TIMESTEP
DOUBLE PRECISION T !AN ARRAY CONTAINING THE RESULTS FROM
THE DIFFERENTIAL EQUATION SOLVER
DOUBLE PRECISION DTDT !AN ARRAY CONTAINING THE DERIVATIVES
TO BE PASSED TO THE DIFF.EQ. SOLVER
INTEGER*4 INFO(15) !THE INFO ARRAY STORES AND PASSES
VALUABLE INFORMATION TO AND FROM THIS TYPE
INTEGER*4 NP,NI,NOUT,ND !VARIABLES FOR THE MAXIMUM NUMBER OF
PARAMETERS, INPUTS, OUTPUTS AND DERIVATIVES
INTEGER*4 NPAR,NIN,NDER !VARIABLES FOR THE CORRECT NUMBER OF
PARAMETERS, INPUTS, OUTPUTS AND DERIVATIVES
INTEGER*4 IUNIT,ITYPE !THE UNIT NUMBER AND TYPE NUMBER FOR
THIS COMPONENT
INTEGER*4 ICNTRL !AN ARRAY FOR HOLDING VALUES OF
CONTROL FUNCTIONS WITH THE NEW SOLVER
INTEGER*4 NSTORED !THE NUMBER OF VARIABLES THAT WILL BE
PASSED INTO AND OUT OF STORAGE
CHARACTER*3 OCHECK !AN ARRAY TO BE FILLED WITH THE
CORRECT VARIABLE TYPES FOR THE OUTPUTS
CHARACTER*3 YCHECK !AN ARRAY TO BE FILLED WITH THE
CORRECT VARIABLE TYPES FOR THE INPUTS
REAL THI, FLWH, TCI, FLWC, CPC, UA, CPH,CC, CH, CMAX, CMIN, RAT
REAL THI, FLWH, TCI, FLWC, CPC, UA, CPH,CC, CH, CMAX, CMIN, RAT REAL UC. GAM. EFF. OMAX. OT. THO. TCO
REAL THI, FLWH, TCI, FLWC, CPC, UA, CPH,CC, CH, CMAX, CMIN, RAT REAL UC, GAM, EFF, QMAX, QT, THO, TCO C
REAL THI, FLWH, TCI, FLWC, CPC, UA, CPH,CC, CH, CMAX, CMIN, RAT REAL UC, GAM, EFF, QMAX, QT, THO, TCO C
REAL THI, FLWH, TCI, FLWC, CPC, UA, CPH,CC, CH, CMAX, CMIN, RAT REAL UC, GAM, EFF, QMAX, QT, THO, TCO C
REAL THI, FLWH, TCI, FLWC, CPC, UA, CPH,CC, CH, CMAX, CMIN, RAT REAL UC, GAM, EFF, QMAX, QT, THO, TCO C C USER DECLARATIONS - SET THE MAXIMUM NUMBER OF PARAMETERS (NP), INPUTS (NI).
REAL THI, FLWH, TCI, FLWC, CPC, UA, CPH,CC, CH, CMAX, CMIN, RAT REAL UC, GAM, EFF, QMAX, QT, THO, TCO C
REAL THI, FLWH, TCI, FLWC, CPC, UA, CPH,CC, CH, CMAX, CMIN, RAT REAL UC, GAM, EFF, QMAX, QT, THO, TCO C
REAL THI, FLWH, TCI, FLWC, CPC, UA, CPH,CC, CH, CMAX, CMIN, RAT REAL UC, GAM, EFF, QMAX, QT, THO, TCO C
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REAL THI, FLWH, TCI, FLWC, CPC, UA, CPH,CC, CH, CMAX, CMIN, RAT REAL UC, GAM, EFF, QMAX, QT, THO, TCO C
REAL THI, FLWH, TCI, FLWC, CPC, UA, CPH,CC, CH, CMAX, CMIN, RAT REAL UC, GAM, EFF, QMAX, QT, THO, TCO C
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REAL THI, FLWH, TCI, FLWC, CPC, UA, CPH,CC, CH, CMAX, CMIN, RAT REAL UC, GAM, EFF, QMAX, QT, THO, TCO C
REAL THI, FLWH, TCI, FLWC, CPC, UA, CPH,CC, CH, CMAX, CMIN, RAT REAL UC, GAM, EFF, QMAX, QT, THO, TCO C
REAL THI, FLWH, TCI, FLWC, CPC, UA, CPH,CC, CH, CMAX, CMIN, RAT REAL UC, GAM, EFF, QMAX, QT, THO, TCO C
REAL THI, FLWH, TCI, FLWC, CPC, UA, CPH,CC, CH, CMAX, CMIN, RAT REAL UC, GAM, EFF, QMAX, QT, THO, TCO C
REAL THI, FLWH, TCI, FLWC, CPC, UA, CPH,CC, CH, CMAX, CMIN, RAT REAL UC, GAM, EFF, QMAX, QT, THO, TCO C

C-----C READ IN THE VALUES OF THE PARAMETERS IN SEQUENTIAL ORDER

Cross_flow_mode=PAR(1)

C
C RETRIEVE THE CURRENT VALUES OF THE INPUTS TO THIS MODEL FROM THE XIN ARRAY IN SEQUENTIAL ORDER
I HI=XIN(1)
FLWH=XIN(2) TCI=VIN(3)
FIWC=XIN(4)
CPC=XIN(5)
UA=XIN(6)
CPH=XIN(7)
IUNIT=INFO(1)
ITYPE=INFO(2)
C SET THE VERSION INFORMATION FOR TRNSVS
C SET THE VERSION INFORMATION FOR TRINSYS IE(INEO(7) EO(2)) THEN
INFO(12)=16
RETURN 1
ENDIF
C
C DO ALL THE VERY LAST CALL OF THE SIMULATION MANIPULATIONS HERE
IF (INFO(8).EQ1) THEN
KETUKN I ENDIE
C
C PERFORM ANY 'AFTER-ITERATION' MANIPULATIONS THAT ARE REQUIRED HERE
C e.g. save variables to storage array for the next timestep
IF (INFO(13).GT.0) THEN
NITEMS=0
C STORED(1)= (if NITEMS > 0)
C CALL SET_STORAGE_VARS(STORED,NITEMS,INFO)
KETUKIN I ENDIE
C
C DO ALL THE VERY FIRST CALL OF THE SIMULATION MANIPULATIONS HERE
IF (INFO(7).EQ1) THEN
C SET SOME INFO ARRAY VARIABLES TO TELL THE TRNSYS ENGINE HOW THIS
TYPE IS TO WORK
INFO(6)=NUUI INFO(0)=1
INFO(9) = 1 INFO(10)=0
ISTORAGE FOR VERSION 16 HAS BEEN CHANGED
C SET THE REQUIRED NUMBER OF INPUTS, PARAMETERS AND DERIVATIVES
THAT THE USER SHOULD SUPPLY IN THE INPUT FILE
C IN SOME CASES, THE NUMBER OF VARIABLES MAY DEPEND ON THE VALUE OF
PARAMETERS TO THIS MODEL
NIN=NI NDAD-ND
NDFR=ND
C CALL THE TYPE CHECK SUBROUTINE TO COMPARE WHAT THIS COMPONENT
REQUIRES TO WHAT IS SUPPLIED IN
C THE TRNSYS INPUT FILE
CALL TYPECK(1,INFO,NIN,NPAR,NDER)
C SET THE NUMBER OF STORAGE SPOTS NEEDED FOR THIS COMPONENT
NITEMS=0

С	CALL SET_STORAGE_SIZE(NITEMS,INFO)
С	RETURN TO THE CALLING PROGRAM RETURN 1
	ENDIF
C C ITE	DO ALL OF THE INITIAL TIMESTEP MANIPULATIONS HERE - THERE ARE NO RATIONS AT THE INTIAL TIME IF (TIME .LT. (getSimulationStartTime() + getSimulationTimeStep()/2.D0)) THEN
С	SET THE UNIT NUMBER FOR FUTURE CALLS IUNIT=INFO(1) ITYPE=INFO(2)
C SUI C	CHECK THE PARAMETERS FOR PROBLEMS AND RETURN FROM THE BROUTINE IF AN ERROR IS FOUND IF() CALL TYPECK(-4,INFO,0,"BAD PARAMETER #",0)
C THI C C	PERFORM ANY REQUIRED CALCULATIONS TO SET THE INITIAL VALUES OF E OUTPUTS HERE Hot-side outlet temperature OUTP(1)=0 Hot-side flow rate
С	OUTP(2)=0 Cold-side outlet temperature
С	OUTP(3)=0 Cold-side flow rate OUTP(4)=0
С	Heat transfer rate OUTP(5)=0
С	Effectiveness OUTP(6)=0
C VAI C	PERFORM ANY REQUIRED CALCULATIONS TO SET THE INITIAL STORAGE RIABLES HERE NITEMS=0 STORED(1)=
C C	PUT THE STORED ARRAY IN THE GLOBAL STORED ARRAY CALL SET_STORAGE_VARS(STORED,NITEMS,INFO)
С	RETURN TO THE CALLING PROGRAM RETURN 1
C	ENDIF
С	
С С	*** ITS AN ITERATIVE CALL TO THIS COMPONENT ***
C C C C	RETRIEVE THE VALUES IN THE STORAGE ARRAY FOR THIS ITERATION NITEMS= CALL GET_STORAGE_VARS(STORED,NITEMS,INFO) STORED(1)=
C C C C C	CHECK THE INPUTS FOR PROBLEMS IF() CALL TYPECK(-3,INFO,'BAD INPUT #',0,0) IF(IERROR.GT.0) RETURN 1

Appendix

C-	
C C	*** PERFORM ALL THE CALCULATION HERE FOR THIS MODEL. ***
!	CALCULATE MINIMUM AND MAXIMUM CAPACITY RATES CH=CPH*FLWH CC=CPC*FLWC CMAX = max(CC,CH) CMIN = min(CC,CH) RAT=CMIN/CMAX UC=UA/CMIN
IF	CMAX .EQ. CC)then GAM=1.d0-exp(-UC) EFF=(1.d0-exp(-GAM*RAT))/RAT
	else GAM=1.d0-exp(-UC*RAT) EFF=1.d0-exp(-GAM/RAT) END IF
!	PERFORM ENERGY CALCULATIONS QMAX=CMIN*(THI-TCI) QT=EFF*QMAX THO=THI-QT/CH TCO=TCI+OT/CC
C-	
C C C C C	SET THE STORAGE ARRAY AT THE END OF THIS ITERATION IF NECESSARY NITEMS= STORED(1)= CALL SET_STORAGE_VARS(STORED,NITEMS,INFO)
	REPORT ANY PROBLEMS THAT HAVE BEEN FOUND USING CALLS LIKE THIS: CALL MESSAGES(-1,'put your message here','MESSAGE',IUNIT,ITYPE) CALL MESSAGES(-1,'put your message here','WARNING',IUNIT,ITYPE) CALL MESSAGES(-1,'put your message here','SEVERE',IUNIT,ITYPE) CALL MESSAGES(-1,'put your message here','FATAL',IUNIT,ITYPE)
C- C	SET THE OUTPUTS FROM THIS MODEL IN SEQUENTIAL ORDER AND GET OUT
С	Hot-side outlet temperature OUTP(1)=THO !outlet temperature on hot (source) side
С	Hot-side flow rate OUTP(2)=FLWH_loutlet flowrate on hot (source) side
С	Cold-side outlet temperature
С	Cold-side flow rate
С	OUTP(4)=FLWC !outlet flowrate on cold (load) side Heat transfer rate
	OUTP(5)=QT !total instantaneous energy transfer across exchanger
С	Effectiveness OUTP(6)=EFF !heat exchanger effectiveness
C-	
C	EVERYTHING IS DONE - RETURN FROM THIS SUBROUTINE AND MOVE ON RETURN 1 END
U-	

Appendix

Appendix 3: Test records for experimental evaluation of model air

conditioning system

10 min)					
No.	Box temperature	Tair,diff Cel	Tw,diff Cel		
1	32.86	15.04	13.24		
2	30.58	16.28	8.25		
3	29.36	16.78	6.86		
4	28.45	17.38	6.09		
5	27.85	17.67	5.45		
7	28.00	16.81	5.27		
8	27.35	17.54	5.13		
9	27.36	17.51	5.11		
10	27.43	17.67	5.04		
11	27.15	17.90	5.15		
12	27.20	17.77	5.40		
13	27.19	17.09	5.44		
14	27.18	18.04	5.29		
15	27.11	17.37	5.111		
16	27.13	18.51	5.09		
17	27.17	18.39	5.07		
18	27.19	17.86	5.33		
19	27.16	17.51	5.41		
20	27.12	17.32	5.38		
21	27.10	17.95	5.21		
22	27.19	18.85	5.06		
23	27.15	17.78	4.99		
24	27.14	18.06	5.07		
25	27.14	18.11	5.10		
26	27.12	18.45	5.029		
27	27.16	18.83	5.11		
28	27.12	18.46	5.08		
29	27.11	18.14	5.11		
30	27.16	17.84	5.13		
31	27.15	18.63	5.19		
32	27.10	18.51	5.18		
33	27.18	18.36	5.03		

Table A3 (a): Test records for model air conditioner with water @ 300 kg/h (every

No.	Box temperature Cel	T _{air,diff} Cel	T _{emulsion,diff} Cel
1	30.86	7.537	4.05
2	28.44	14.51	4.47
3	28.70	14.71	4.31
4	28.33	15.13	4.59
5	27.99	15.55	4.73
7	27.66	15.93	4.89
8	27.23	16.30	5.00
9	26.93	16.53	4.99
10	26.82	16.75	4.94
11	26.61	16.83	5.24
12	26.57	16.92	5.22
13	26.43	17.04	5.25
14	26.37	17.16	5.10
15	26.26	17.22	5.08
16	26.19	17.28	5.21
17	26.32	17.16	5.10
18	26.21	17.04	5.02
19	26.2	17.07	5.17
20	26.40	17.09	5.18
21	26.35	17.09	5.06
22	26.33	17.09	5.15
23	26.39	17.09	5.06
24	26.24	17.09	5.20
25	26.25	17.05	5.125
26	26.32	17.01	5.13
27	26.40	17.04	5.13
28	26.36	17.08	5.07
29	26.36	17.02	5.06
30	26.18	16.97	5.04
31	26.28	17.01	5.25
32	26.20	17.06	5.15
33	26.22	17.04	5.22

Table A3 (b): Test records for model air conditioner with PCME @ 130kg/h

(every 10 min)