TECHNICAL MEMORANDUM

Field Validation of Microencapsulated Phase-Change Material Slurries as Heat-Transfer Fluids

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14. ABSTRACT

Thermal management of facilities and processes on military installations requires the continual pumping of large volumes of a heat-transfer fluid (HTF). Laboratory studies have shown that HTFs infused with microencapsulated phase-change materials (MPCMs) have a higher heat-carrying capacity than conventional HTFs, which could reduce the required fluid-flow rate and, thus, system power costs. This study demonstrated the performance of MPCM slurries under continuous pumping conditions in full-scale applications at two U.S. military installations. An instrumented test loop was temporarily integrated with ground-source heat-exchange systems at Fort Hood, TX, and Fort Dodge, IA, to compare the thermal performance and pumping-power requirements of conventional HTFs and MPCM slurries.

Premature capsule breakage during early tests was addressed by thickening capsule walls and replacing the centrifugal circulation pump with a progressive cavity pump to reduce mechanical shear stresses. MPCM heat-carrying capability varied with mass ratio between capsule wall and phase-change material. The heat-transfer coefficient of performance for MPCMs was seen to improve at least 10% over conventional HTFs, with system power-consumption reductions of at least 3%. An economic assessment indicates that implementation of MPCMs as a drop-in application could provide a 1.77 return on investment ratio. However, results suggest that the technology is not mature enough to recommend for widespread drop-in implementation at this time.

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microencapsulated phase-change materials (MPCMs); Fort Hood, TX; Fort Dodge, IA; thermal energy; testing; heat transfer; energy research

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EXECUTIVE SUMMARY

Thermal management of facilities and processes on military installations requires the continual pumping of large volumes of a heat-transfer fluid (HTF), typically water with chemical additives. Laboratory studies have shown that slurries infused with microencapsulated phase-change materials (MPCMs) have a higher heat-carrying capacity than conventional HTFs, which could reduce the required fluid-flow rate and, thus, total power costs for the system. The present study was performed to investigate the performance of MPCM slurries under continuous pumping conditions in full-scale heat-transfer applications at two U.S. military installations. An instrumented test loop was temporarily integrated with ground-source heat-exchange systems at Fort Hood, TX, and Fort Dodge, IA, to measure and compare the thermal performance and pumping-power requirements of conventional HTFs and MPCM slurries.

Premature capsule breakage during early tests was addressed by thickening capsule walls and replacing the centrifugal circulation pump with a progressive cavity pump to reduce mechanical shear stresses. MPCM heat-carrying capability varied with mass ratio between capsule wall and phase-change material. The heat-transfer coefficient of performance for MPCMs was seen to improve at least 10% over conventional HTFs, with system power-consumption reductions of at least 3%. An economic assessment indicates that implementation of MPCMs as a drop-in application could provide a 1.77 return on investment ratio. However, results suggest that the technology is not mature enough to recommend for widespread drop-in implementation at this time.

1. INTRODUCTION

1.1 BACKGROUND

U.S. military installations are equipped with district heating and cooling systems to handle a large percentage of thermal-management requirements for space conditioning and other operational processes. These systems include large pipe loops filled with a heat-transfer fluid (HTF), typically water. Because the pipe loops contain a high volume of water, a considerable amount of energy is consumed to pump the HTF between thermal source and sink. Technologies that can reduce pumping costs for large-scale military heating and cooling applications are desirable to promote cost reduction and energy sustainability. One way to approach this goal using existing piping networks and systems is to improve the heat-transfer capabilities of the HTF. This could reduce the required flow rate through the system, consequently saving energy by decreasing pumping power consumption (Alvarado et al. 2007).

Over the past decade, researchers have been developing novel HTFs that incorporate phase-change materials (PCMs) engineered with a high latent heat of fusion. These materials need to absorb (or release) more heat than water to change from a solid to liquid phase (or the converse) before the material bulk temperature begins to change. Consequently, these PCMs can absorb and transfer greater amounts of thermal energy than water or propylene glycol when properly dispersed as part of an engineered carrier fluid (Sabbah et al. 2008, Wang et al. 2008). It has been shown that these PCMs can be enclosed in durable micro-scale capsules, or *microcapsules*, and delivered at high volume fractions with water to serve as a slurry in heat-transfer loops. Aqueous HTF slurries loaded with these microencapsulated phase-change materials (MPCMs) also have been shown to have heat-transfer coefficients and pressure-drop values similar to water, meaning that they absorb and release heat at approximately the same rate as conventional HTFs and have similar pumpability characteristics (Yamagishi et al. 1996; Yamagishi et al. 1999). At high volume fractions in the HTF, these microencapsulated phase-change materials (MPCMs) have been shown in laboratory testing to increase heat-carrying capacity up to 50% as compared to water in both turbulent and laminar flow conditions (Alvarado et al. 2007).

The durability of microcapsules exposed to the stresses of pumping in a real-world heat-transfer application is important because free PCMs released into the water as a result of capsule rupture could lead to fouling of the piping, heat exchangers, and pumps. In the laboratory heat-transfer apparatus, microcapsules did not show signs of significant damage through 600 – 1,000 cycles of pumping through the loop (Thies and Alvarado 2007). However, conditions within a large-scale heating or cooling loop would be considerably more severe than in a laboratory test loop, so a demonstration of MPCMs in a real-world application was needed.

The Environmental Security Technology Certification Program (ESTCP) funded the U.S. Army Engineer Research and Development Center, Construction Engineering Research Laboratory (ERDC-CERL) to demonstrate the use of MPCM slurries in full-scale heating and cooling applications on Army installations. The purpose of the work was to determine whether laboratory-scale heat-transfer performance and characteristics could be verified in field use, and whether the technology could reduce energy requirements for real-world district heating and cooling systems.

1.2 OBJECTIVE OF THE DEMONSTRATION

The objective of this project was to validate the thermal and physical performance of selected MPCMs used in HTF slurries for commercial/industrial-scale ground-source heat pump systems. The energy sustainability metric for success was validation of a substantial improvement of heat-transfer performance, as previously observed in the laboratory studies.

1.3 DEMONSTRATION SITES

The demonstration was intended to verify the performance of this technology in real-world heat-transfer applications. Currently operating ground-source heat pumps (GSHPs) were selected as the most compatible systems for "drop-in" demonstration of MPCM HTFs.

Fort Hood, TX, was selected as the warm-weather site for testing MPCMs in an operating GSHP system used primarily for cooling applications. The system consists of 108 wells of 1 in. (nominal) bore holes that are 300 ft deep. The HTF is water circulated from the well field to the heat pumps.

Fort Dodge, IA, is located in a temperate Midwestern region. The ground-source temperature remains high enough during winter to use as a heating source. The Fort Dodge GSHP system consists of 30 wells that are 300 ft deep. The HTF is an 80/20 volume-percent mixture of water and propylene glycol, respectively.

2. TECHNOLOGY DESCRIPTION

2.1 OVERVIEW

The MPCM species used in this demonstration were developed by Thies Technology, Inc., Henderson, NV (see Figure 1). A significant performance improvement in pumped-flow conditions has been achieved with new MPCM slurry formulations that contain a high payload (85 wt%) of MPCMs. The demonstrated PCM is an engineered, biodegradable wax-like material that is contained in durable microcapsules. When transported as part of an aqueous slurry in a heating or cooling loop, the PCM changes between solid and liquid phases, melting as the slurry absorbs heat and solidifying as it releases heat through thermal cycling. The PCM has higher heat-carrying capacity than water because the material's latent heat of fusion—the heat required to change material phase before the bulk material temperature begins to rise.

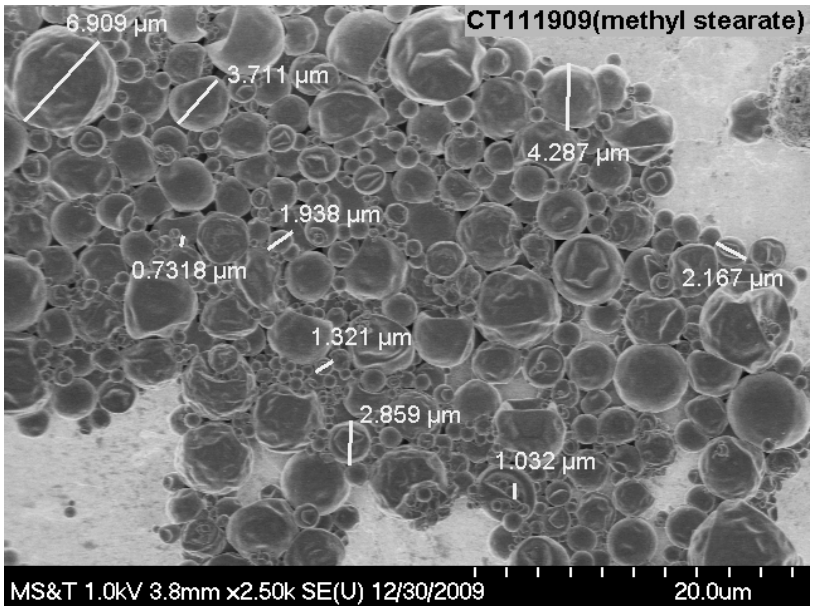

Figure 1. Microraph of MPCM capsules labeled with representative size range.

Microencapsulation of materials is a well developed technology, used prominently for many years in the carbonless copy paper industry. However, that conventional technology is not well suited to combining with a phase-change material for heat-transfer purposes owing to the thermal environment, turbulence, and mechanical stresses to which capsules would be exposed in heating and cooling systems. Over the past 10 years, Thies Technology has developed a process for fabricating more durable microcapsules that are suitable for use with micro-volumes of PCM in pumped flow conditions. In collaboration with Texas A&M University, the company has developed MPCMs that can be pumped continuously for thousands of cycles with little degradation (Thies and Alvarado 2007) This accomplishment represents a significant advance because capsule durability is essential for economical operation and to avoid fouling the pumps, pipes, and exchangers. With support from the National Science Foundation (NSF), Texas A&M researchers recorded data using an instrumented laboratory unit to establish that the subject MPCM improves thermal performance for heat-exchange applications (Thies and Alvarado 2007).

2.2 PROSPECTIVE BENEFITS

Conservative calculations based on laboratory heat-transfer data as well as processing and MPCM costs show that MPCM-based systems are expected to save at least $25.10 per year per 100 W of pumping power consumed. MPCMs could provide significant energy savings from pumping energy reduction not only to DoD facilities, but also to a range of government and commercial facilities.

If MPCMs are verified to be practical in heating and cooling applications, new systems exploiting this technology could be designed in the future to reduce capital and operational costs related to installation heating and cooling.

2.3 LIMITATIONS

Several potential technical risks are associated with using MPCMs in pumped flow systems. The primary concern is capsule breakage due to action of the pump, resulting in release of PCMs into the system. Free PCM could solidify during the cooling cycle to cause clogging. A second potential risk is the possibility that intact MPCMs could agglomerate at points in the loop where flow is restricted, such as a heat exchanger. Both risks could be aggravated by pipe fouling, which is a typical problem in many heating and cooling systems.

Custom fabrication of MPCMs for the conditions in individual systems will increase initial costs, especially compared with water. (However, the water in heating and cooling loops is typically treated with chemicals to prevent corrosion or scaling, so those costs will affect the economic analysis of the demonstrated technology.)

3. PERFORMANCE OBJECTIVES AND METRICS

The technology performance objectives for MPCM slurry HTF are listed below:

- The facility energy use coefficient of performance (COP) for a heat pump using MPCM slurry ($COP_{MPCM\ HTF}$) is greater than or equal to 1.1 as compared with traditional (water) HTF for both heating and cooling applications.
- Hours of system maintenance required using MPCM HTF is equal to or less than traditional HTF.
- Life-cycle cost of MPCM HTF, including material and operational cost, is equal to or less than traditional HTF.
- Estimated annual load of system (in kilowatt-hours) using MPCM HTF is equal to or less than traditional HTF.
- Heat load to pumping power ratio, η, is greater than 4.0.
- A simple payback period of less than 10 years when equipment is used continually.
- Acceptable ease of use, based on feedback from demonstration field technicians.
- Acceptable user (technician) satisfaction with MPCM HTF as compared to traditional HTF.

Table 1 presents the technical basis for each performance metric and success criterion, and Table 2 defines the parameters used in the objectives.

Table 1. Basis of Performance Objectives.

Performance Objective	Metric	Data Requirements	Success Criteria	Criteria Met?
COP for Facility Energy Use, Heat Pump Heating	$\mathrm{COP_h} = \frac{\text{Net Heating Rate}}{\text{Electric Energy Rate of Heat Pump and Pump}} = \frac{\dot{Q}_{\text{ground}} + \dot{W}_{\text{heat pump}} + \dot{W}_{\text{pump}}}{\dot{W}_{\text{heat pump}} + \dot{W}_{\text{pump}}} = \frac{\dot{m} \cdot c_p \cdot \Delta T_{\text{HTF}} + (V \cdot I)_{\text{heat pump}} + (V \cdot I)_{\text{pump}}}{(V \cdot I)_{\text{heat pump}} + (V \cdot I)_{\text{pump}}}$	HTF trended data of temperature and flow rate. Power consumption of heat pump unit and HTF circulator pumps.	$\frac{\mathrm{COP}_{\text{MPCM-HTF}}}{\mathrm{COP}_{\text{traditional-HTF}}} \geq 1.1$	Yes
COP for Facility Energy Use, Heat Pump Cooling	$\mathrm{COP_c} = \frac{\text{Net Cooling Rate}}{\text{Electric Energy Rate of Heat Pump and Pump}} = \frac{\dot{Q}_{\text{ground}} - \dot{W}_{\text{heat pump}} - \dot{W}_{\text{pump}}}{\dot{W}_{\text{heat pump}} + \dot{W}_{\text{pump}}} = \frac{\dot{m} \cdot c_p \cdot \Delta T_{\text{HTF}} - (V \cdot I)_{\text{heat pump}} - (V \cdot I)_{\text{pump}}}{(V \cdot I)_{\text{heat pump}} + (V \cdot I)_{\text{pump}}}$	HTF trended data of temperature and flow rate. Power consumption of heat pump unit and HTF circulator pumps.	$\frac{\mathrm{COP}_{\text{MPCM-HTF}}}{\mathrm{COP}_{\text{traditional-HTF}}} \geq 1.1$	Yes
System Maintenance	Number of hours required to service the system while in operation	Number of hours of service	$\mathrm{Hours}_{\text{MPCM}} \leq \mathrm{Hours}_{\text{existing fluid}}$	No
System Economics	Life cycle cost of heat-transfer fluid including material and operational cost: LCC = Σ(cost of materials + cost of energy + others)	Total estimated cost for a 10 year period	$\mathrm{Cost}_{\text{MPCM}} \leq \mathrm{Cost}_{\text{existing fluid}}$	Yes
Load Reduction	Estimated annual load of system with heat-transfer fluid: AL = Σ(daily loads)	Amount of kW-hr per system per year	$\text{kW-hr}_{\text{MPCM}} \leq \text{kW-r}_{\text{existing fluid}}$	Yes
Heat load-to-pumping ratio	$\eta = \frac{\text{Heat Load}}{\text{Pumping Power}}$ $\eta = \frac{\dot{m} \cdot c_p \cdot \Delta T_{\text{HTF}}}{(V \cdot I)_{\text{pump}}}$	Inlet and outlet temperatures and pressures, flow rate, current, and voltage	$\eta > 4.0$	Yes
Simple Payback Period	$SPBP = \frac{\text{Initial Cost}}{(\text{Annual Savings - Annual Cost})}$	Cost of MPCM and pumping cost savings	Less than 10 years for equipment used almost continuously	No
Ease of use	User satisfaction survey	Feedback from field technician on how to fill up the system with MPCM slurry	Acceptable user satisfaction	No
User Satisfaction	Annual customer satisfaction survey	Survey questions	Acceptable satisfaction when compared with existing fluids	No

Table 2. Nomenclature of Parameters in Performance Objectives.

Parameter	Definition
COP_h	Coefficient of performance for heating
COP_c	Coefficient of performance for cooling
Q_{ground}	heat rate received from the ground source (if heating) and delivered to the ground source if cooling
$\dot{W}_{heat\ pump}$	energy consumed by the heat pump system in Fort Hood
$\dot{W}_{pump}$	energy consumed by the HTF circulator pump
$\dot{m}$	mass flow rate
c_p	heat capacity
ΔT_{HTF}	temperature difference between input and output of the ground source
η	heat load to pumping ratio
$COP_{MPCM\text{-}HTF}$	The coefficient of performance when MPCM is used as HTF
$COP_{traditional\text{-}HTF}$	The coefficient of performance when water is used as HTF
V	voltage measured for HTF circulation pump and heat pump
I	current measured for HTF circulation pump and heat pump

4. APPROACH

4.1 CONCEPTUAL DESIGN

Because MPCM slurries are an emerging technology without a long-term record of application, use in a live pumped-flow system poses several potential risks. The primary concern is capsule breakage and the release of free phase-change material, which could form large agglomerates during the cooling cycle. This could lead to clogging that would interfere with operation of the pumps and, consequently, the entire GSHP system. Intact capsules might also agglomerate at certain points in the loop. In order to reduce the risk of potential problems in this demonstration, a conceptual design was developed to isolate the MPCM slurry from the HTF in the main heat-transfer systems at Fort Hood and Fort Dodge. This involved construction of an isolated heat-transfer loop to be placed between the ground source and the heat pump being tested. The design is illustrated in Figure 2. The demonstration loop components are described below.

4.2 INSTRUMENTED LOOP DESIGN AND COMPONENTS

An instrumented heat-transfer loop was constructed for use in both laboratory and field testing (Figure 3 and Figure 2). This apparatus was designed to supply an HTF at specified flow condition for heat exchange. It includes an HTF circulation pump, heat exchanger, a heat pump and a heat sink, two flow meters, two differential pressure transducers, and thermocouples. The data acquisition system obtains and records HTF flow rate, differential pressure drop, and power consumption of both the fluid pump and the heat pump unit.

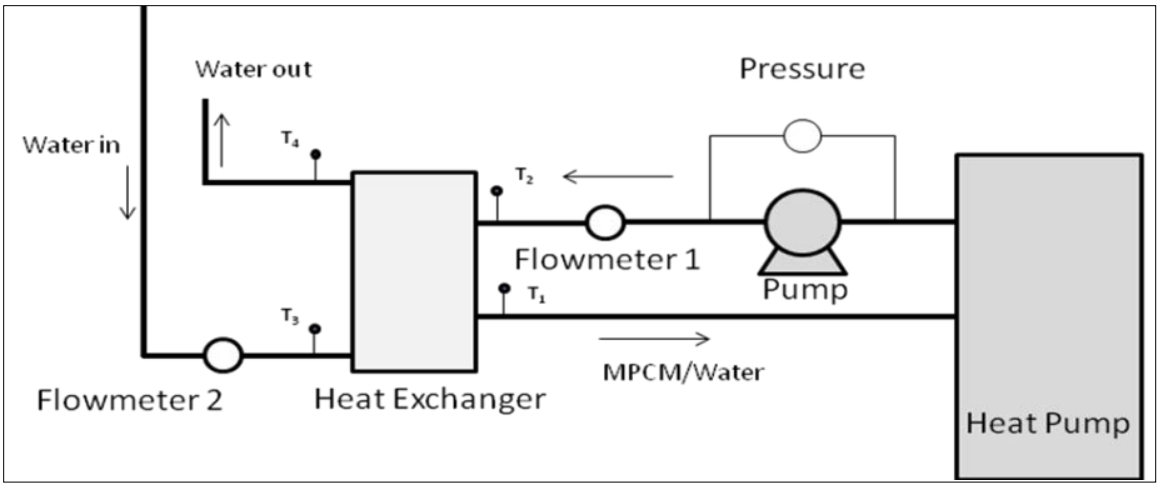

Figure 2. Schematic diagram of instrumented slurry loop.

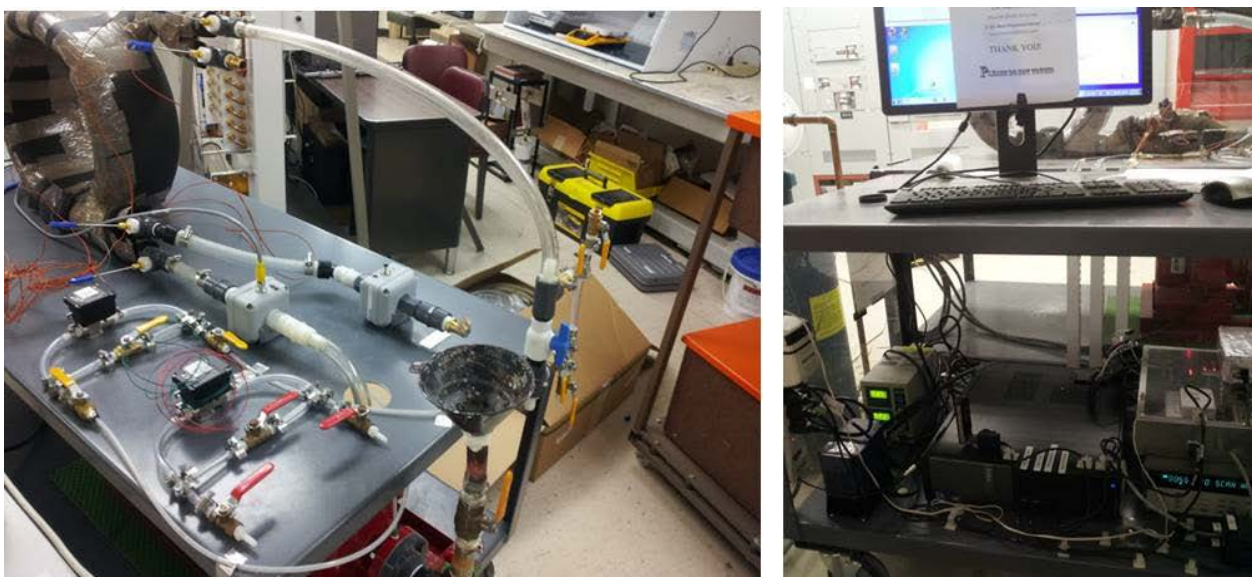

Figure 3. Instrumented loop (left) and DAQ system (right) in laboratory.

This instrumented loop was transported to the two field sites and integrated with the ground-source systems for purposes of collecting data from full-scale heat-transfer loops (see section 4.3 below).

4.2.1 Pumps

Two different types of commercial circulation pumps were used in this study. In the laboratory setting and, subsequently, in most of the field tests, a Moyno 500 progressive-cavity pump was used. This pump is capable of flow rates up to 5 GPM and output pressure up to 100 psi, and was driven by a 1 horsepower motor manufactured by Century Motors. Motor speed was controlled using a variable-frequency drive (VFD) capable of operating at 0 – 1,800 revolutions per minute. In two early field runs at Fort Hood, an Armstrong centrifugal pump was used. This pump design is commonly used in heating, ventilating, and air-conditioning (HVAC) systems, and is capable

of flow rates up to 50 GPM and output pressures up to 10 psi. Use of the Armstrong pump was discontinued in the field for reasons discussed in section 5.1.

4.2.2 Heat exchanger

The selected heat exchanger was a coil-type unit made by Packless Industries to exchange heat between MPCM slurry and the standard HTF used in the main installation ground-source loop. The heat-transfer capacity is 4 tons (48,000 BTUh), and the material is copper in both inner and outer tubes. The MPCM slurry flowed through the inner copper tube of the coil heat exchanger and the other side of the coil heat exchanger was used for the existing HTF.

4.2.3 Flow meter

Two flow meters were used to measure the flow rates in both MPCM slurry loop and the existing installation ground-source loop to calculate the amount of heat-transfer rate. Omega FMG84 electromagnetic flow meters were selected to avoid direct mechanical contact between the flow meter and MPCM capsules. The flow rate range is 0 – 20 GPM and the accuracy is ±1% reading scale.

4.2.4 Differential-pressure transducer

Two differential-pressure transducers were used in the heat-transfer loop to monitor flow conditions. The pressure drop across the heat pump unit was measured by a Cole-Parmer 0 – 5 psid differential pressure transducer and the pressure drop across the MPCM slurry loop was measured by a Cole-Parmer 0 to 25 psid differential pressure transducer. The accuracies of both DPTs are ±0.25% full scale and linear calibration curves were obtained to determine the differential pressure drop.

4.2.5 Thermocouple

For the field tests, Omega T-type immersion thermocouples (TCs) were used to measure the bulk fluid temperatures of both the MPCM slurry and existing HTF in the test site ground-source loops. All thermocouples were calibrated by using water as the HTF under isothermal conditions.

4.2.6 Power measurement

Digital volt and ampere meters were used to measure voltage and current from the motor of the pump and the compressor of the heat pump unit. Pumping-power voltage and current measurements were obtained using a National Instruments (NI) module.

4.2.7 Data acquisition (DAQ)

An Agilent Data Logger (34970A) with one multiplexer card with the Agilent Benchlink Data Logger software was used to obtain the bulk fluid temperatures. A National Instruments Module with the Labview software was used to obtain flow rate, pressure drop, and power consumption

data. The DAQ system uses a graphical user interface to display and analyze the results in real time.

4.3 SLURRY LOOP IMPLEMENTATION AT TEST SITES

4.3.1 Fort Hood

A 2.5 ton capacity heat pump unit (Florida Heat Pump Model GT030-2VTC) in the mechanical room was selected for the MPCM performance tests. A schematic flow diagram of an individual Fort Hood heat pump unit is shown in Figure 4. The existing heat pump loop was connected with the instrumented heat-transfer loop in a way to prevent the existing HTF from mixing with the MPCM slurry during the demonstration. A flow diagram and of the onsite test setup with the instrumented test loop in place is shown in Figure 5.

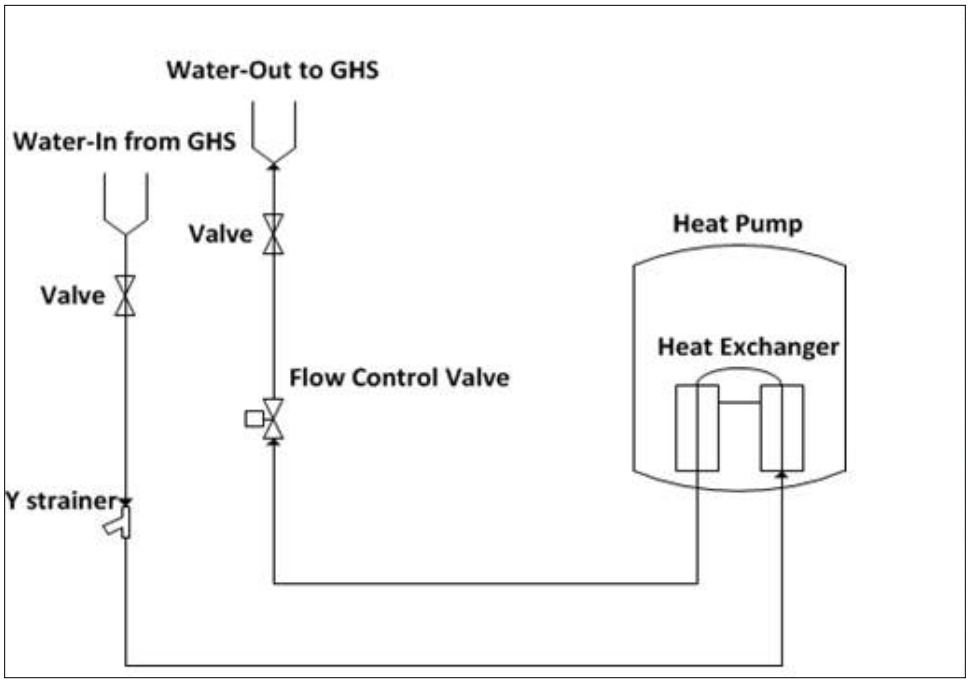

Figure 4. Flow diagram of existing Fort Hood heat pump loop.

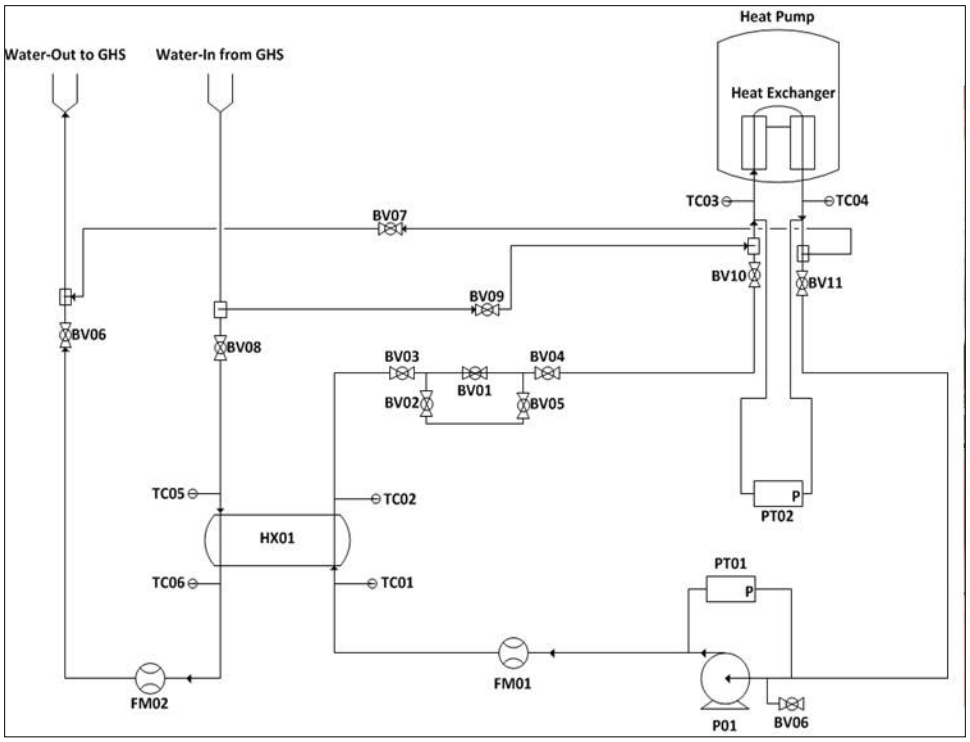

Figure 5. Flow diagram of slurry loop integrated with Fort Hood system.

The MPCM slurry supplied by a centrifugal pump (P01 in Figure 5) was circulated through a co-axial heat exchanger (HX01) and the heat pump's heat exchanger to ensure a complete PCM melting-solidifying cycle. Since the MPCM slurry transfers energy from the heat pump heat exchanger (upper right in Figure 5) for cooling mode, that exchanger is where the MPCM melting takes place. Cold water supplied from the ground source circulates through HX01 to cool the MPCM slurry, so the solidifying process occurs in that exchanger. Flow meters FM01 and FM02 measure the flow rates of the MPCM slurry and ground-source HTF, respectively. Pressure transducers PT01 and PT02 measure pressure drops across the entire heat-transfer loop and the heat exchanger of the heat pump, respectively. Ball valves BV01 to BV05 were designed for taking MPCM slurry samples and measuring thermal properties. Bypass lines that include BV6 to BV11 were installed to allow for continued heat pump operation in the event of a failure of the MPCM slurry system.

4.3.2 Fort Dodge

The Fort Dodge GSHP system uses an 80/20 weight-percent solution of water and propylene glycol as the HTF. A mechanical room heat pump with a 1 ton heating/cooling capacity (Florida Heat Pump Model EM012) was selected for the demonstration. It is used for both heating and cooling purposes depending on the season. Other than the standard installation HTF and the model of the heat pump, all essential design and operating details were the same as the Fort Hood setup.

5. DATA COLLECTION AND ANALYSIS

Tests were conducted to study the enhancement of heat-transfer performance of systems in place at Fort Hood and Fort Dodge using MPCM slurries as compared with the traditional HTFs used there (i.e., water and 80/20 water/propylene glycol, respectively). First, the tests were run with the traditional HTFs in the instrumented test loop to obtain the baseline data needed for comparison with the results of the MPCM slurries. In addition, the operating temperature range obtained from the existing HTFs tests provided guidance to decide the melting and solidifying temperatures of the MPCM capsules. Then, different MPCM slurries with different mass fractions of PCM were used and a certain flow rate of the MPCM slurries was selected, which could carry the same amount of heat-transfer rate as the traditional HTFs tests to compare with the baseline.

5.1 FORT HOOD

Initial tests using water as the HTF were performed in the instrumented loop using the rated flow rate for the heat pump of 6.0 GPM. These results served as baseline data used for later comparisons to MPCM slurry tests. Three different slurries provided by Thies Technology were used at different PCM mass fractions; Table 3 shows MPCM properties based on a differential scanning calorimeter (DSC) test. An Armstrong circulation pump was installed in the instrumented loop for the first and second tests, but was replaced with the Moyno progressive pump in all later tests for reasons explained in section 5.1.3. Test conditions for the slurry and ground source loops at Fort Hood are shown in Table 4. Table 5 and Table 6 summarize the test results, which are discussed in sections 5.1.1 – 5.1.4. As shown in Table 6, a COP improvement of at least 10% was demonstrated, and the heat load to pumping power ratio was also significantly enhanced.

Table 3. PCM batch properties for three Fort Hood tests.

		Test 1 (Oct. 31, 2012 ~ Nov. 9, 2012)	**Test 2 (Dec. 13, 2012 ~ Jan. 8, 2013)**	**Test 3 (Aug. 14, 2013 ~ Oct. 18, 2013)**
Batch No.		CT100312	JCT091412	CT080513
Melting Temperature		38.4 ℃	36.9 ℃	41.7 ℃
Solidifying Temperature		38.3 ℃	36.5 ℃	36.7 ℃
Latent Heat of Fusion	@ Melting	190 J/g	169 J/g	139.6 J/g
	@ Crystallization	193 J/g	172 J/g	138.4 J/g

Table 4. Fort Hood HTF loop parameters.

Test No. (Date)		**Test 1 (Oct. 31, 2012 ~ Nov. 9, 2012)**	**Test 2 (Dec. 13, 2012 ~ Jan. 8, 2013)**	**Test 3 (Aug. 14, 2013 ~ Oct. 18, 2013)**
Batch No.		CT100312	JCT091412	CT080513
Hood MPCM Slurry Loop	Mass fraction	11%	6%	6.5%
	Flow rate	4.2 GPM	4.0 GPM	4.5 GPM
	Temperature range	36.2 ~ 43.7℃	34.5 ~ 41.5℃	39.6 ~ 47.9 ℃
Hood Ground Source Loop	Flow rate	7.9 GPM	6.2 GPM	6.5 GPM
	Temperature range	34.6 ~ 38.8 ℃	32.6 ~ 37.5 ℃	37.6 ~ 43.3 ℃

Table 5. Fort Hood pumping power data for three test periods.

Demonstration Period	**Test 1 (Oct. 31, 2012 ~ Nov. 9, 2012)**	**Test 2 (Dec. 13, 2012 ~ Jan. 8, 2013)**	**Test 3 (Aug. 14, 2013 ~ Oct. 18, 2013)**
Hood average pumping power for water (baseline)	216.4 W	123.0 W	109.4 W
Hood average pumping power for MPCMs	163.2 W	107.9 W	100.5 W
Difference between baseline HTF and MPCM	53.2	15.1	8.9

Table 6. Summary of Fort Hood MPCM test results.

Performance Objective	**Success Criterion**	**Experimental Results**
Coefficient of Performance (COP) for cooling mode	$\frac{COP_{MPCM\ slurry}}{COP_{Traditional\ HTF}} \geq 1.1$	1.08 (Test 1) 1.10 (Test 2) 1.06 (Test 3)
Heat load to pumping ratio	$\eta = \frac{Heat\ trasnfer\ rate}{Pumping\ power\ input} \geq 4.0$	55.2 (Test 1) 77.2 (Test 2) 95.5 (Test 3)
Durability		14,440 (Test 1) 38,880 (Test 2) 164,917 (Test 3) 53,500 (add. Test)

5.1.1 Hood Test 1

Figure 6 shows pumping power consumption for both HTFs. Figure 7 shows that the heat load to pumping power ratio increased when the heat-transfer rate increased and the ratios of the MPCM slurry were higher than those of water. Moreover, the improvement of the heat load to pumping ratio of the MPCM slurry was up to 36.8% compared to water. This indicates that the higher heat-carrying capacity of the MPCM enhanced by the phase change process reduced a required flow rate of a heat-transfer fluid in the system and consequently saved pumping energy.

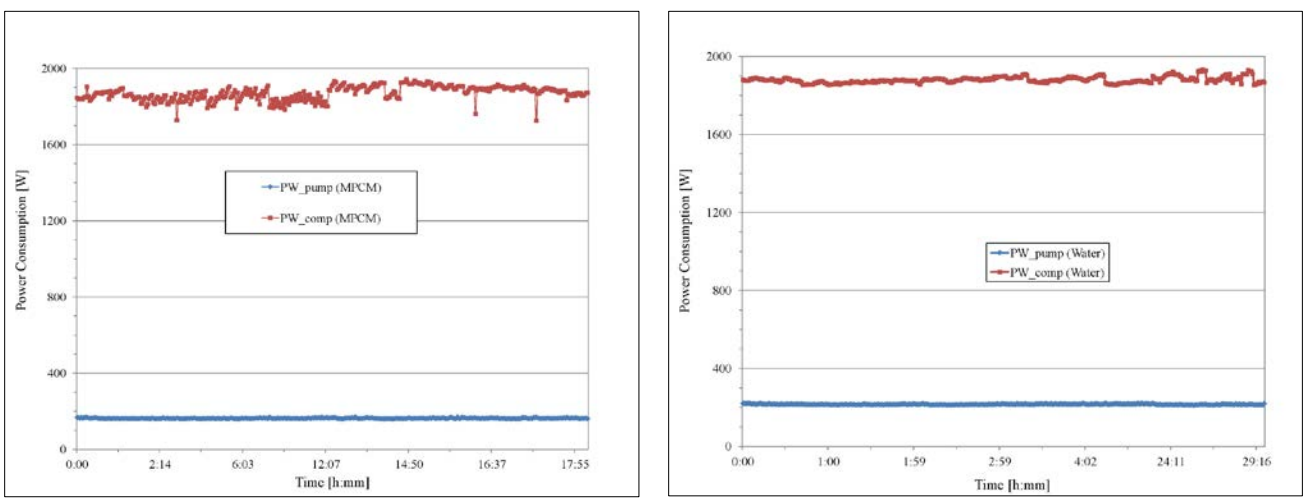

Figure 6. Hood Test 1: power consumption (PW) in MPCM slurry (left) and water (right).

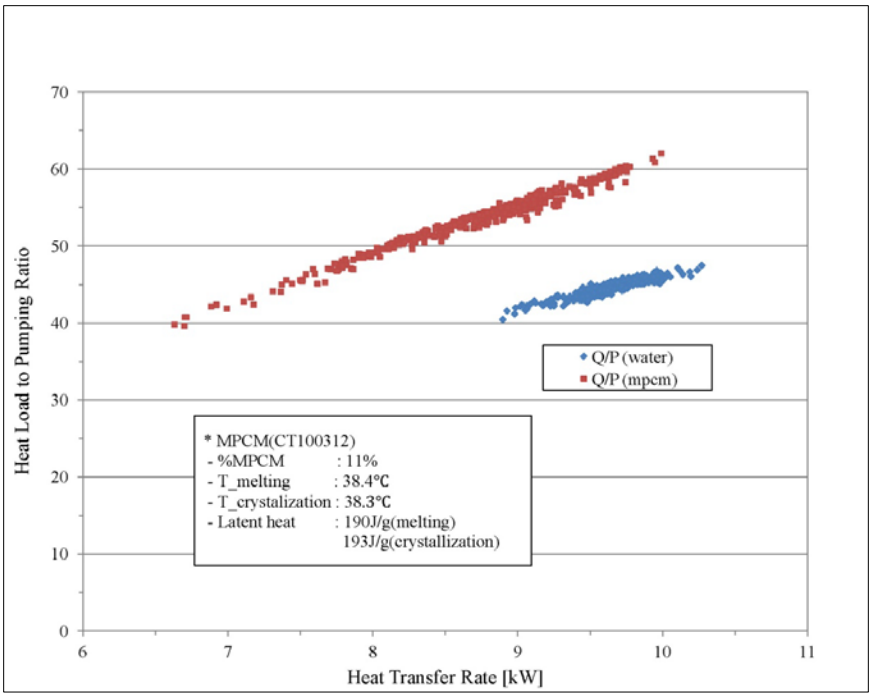

Figure 7. Hood Test 1: heat load to pumping ratio as a function of heat-transfer rate.

The use of MPCM slurry reduced the pumping power input to 62.4 W and reduced total power consumption by 3%. This reduction in the pumping power requirement represents over 85% of energy savings attributable to use of MPCMs. It was found that the improvement of the COP for the system with the MPCM slurry was mainly affected by reduced pumping power.

The first test at Fort Hood ran from October 31 – November 9, 2012, and had to be terminated early when sampling revealed that capsule breakage had led to debris accumulations in a gap between pump impeller and housing.

5.1.2 Hood Test 2

Figure 8 shows pumping power consumption for both HTFs. Figure 9 shows that the heat load to pumping ratio increased with increasing the heat-transfer rates and the ratios of the MPCM slurry were higher than those of water. The enhancement of the heat load to pumping ratio of the MPCM slurry was up to 18.3%. Like the results of Test 1, the reduced flow rate induced by the high heat carrying capacity of the MPCM saved energy consumed by the pump and then enhanced the heat-transfer performance of the heat pump system.

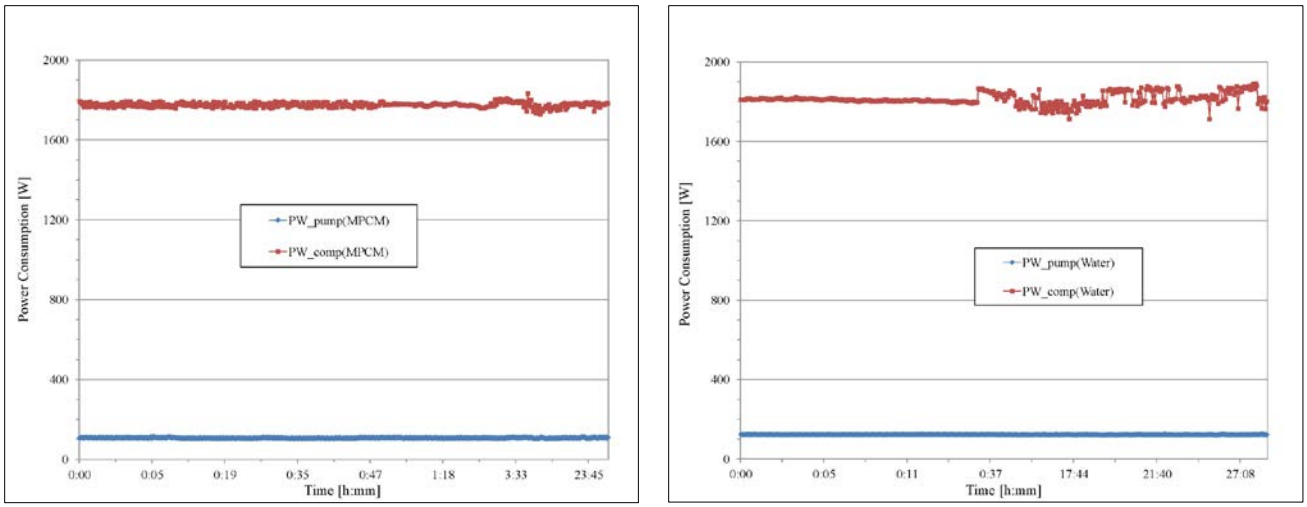

Figure 8. Hood Test 2: power consumption in MPCM slurry (left) and water (right).

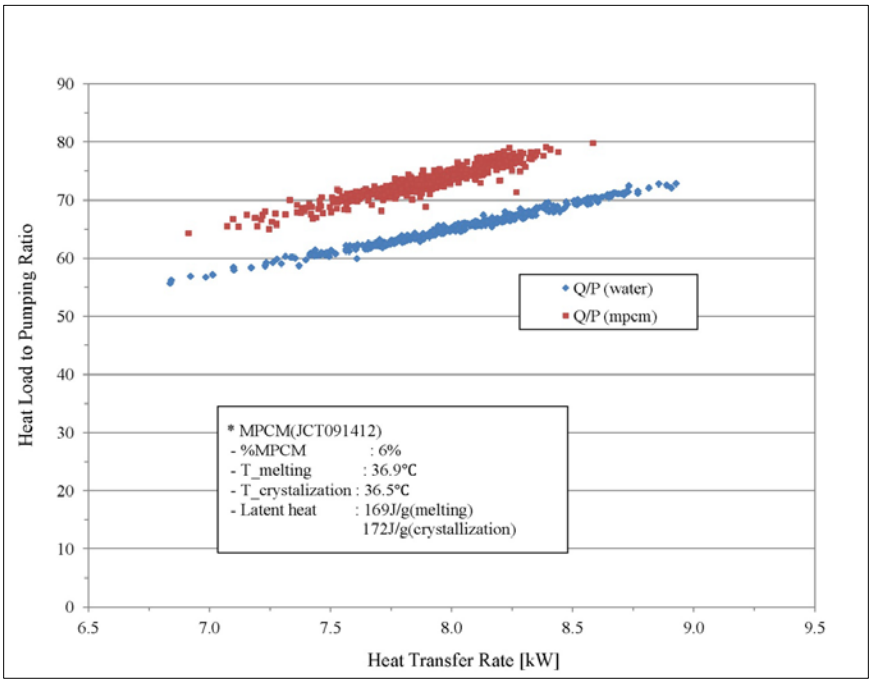

Figure 9. Hood Test 2: Heat load to pumping ratio as a function of heat-transfer rate.

Use of the MPCM slurry reduced pumping power input by 53.6 W, representing a 3% reduction in total power consumption.

The MPCM capsules were continuously pumped 38,880 cycles through a centrifugal pump. The MPCM capsules were damaged and increased the pressure drop across the heat-transfer loop. As in Test 1, sampling revealed an unacceptable level of capsule breakage.

5.1.3 Hood Pump Substitution Test

Sampling results from the first and second demonstrations indicated that the Armstrong centrifugal pump was not suitable for the supplied MPCM capsule batches, causing a high level of rupturing. Before the 3rd heat-exchange test was performed, a supplementary test focusing on pump type and capsule durability was performed. This test on MPCM batch CT071513 was performed from July 20 – August 13 in 2013. In this test, the MPCMs resisted continuous pumping conditions after circulating 53,500 cycles through a Moyno progressive cavity pump. During the test period, no MPCM breakage was observed in the samples.

5.1.4 Hood Test 3

Figure 10 shows pumping power consumption for both HTFs. Figure 11 shows that the heat load to pumping ratio increased when the heat-transfer rate increased and the ratios of the MPCM slurry were higher than those of water. Moreover, the heat load to pumping ratio of the MPCM slurry was improved by 9.2% compared to water. It indicates that the MPCM reduced a required flow rate of a heat-transfer fluid at the same heat-transfer rate and consequently saved pumping energy.

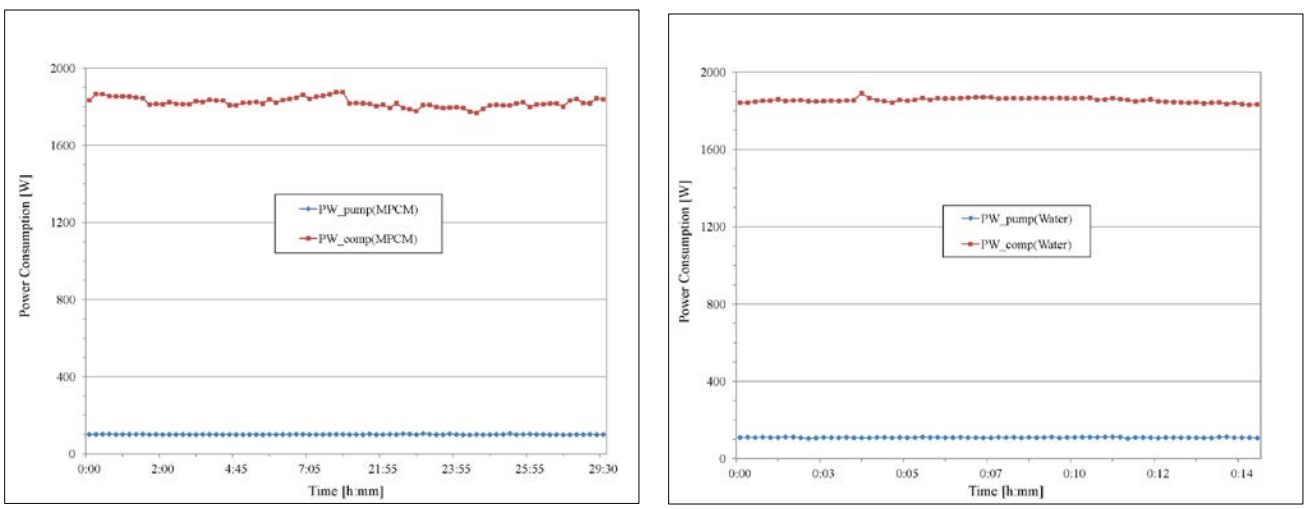

Figure 10. Hood Test 3: power consumption in MPCM slurry (left) and water (right).

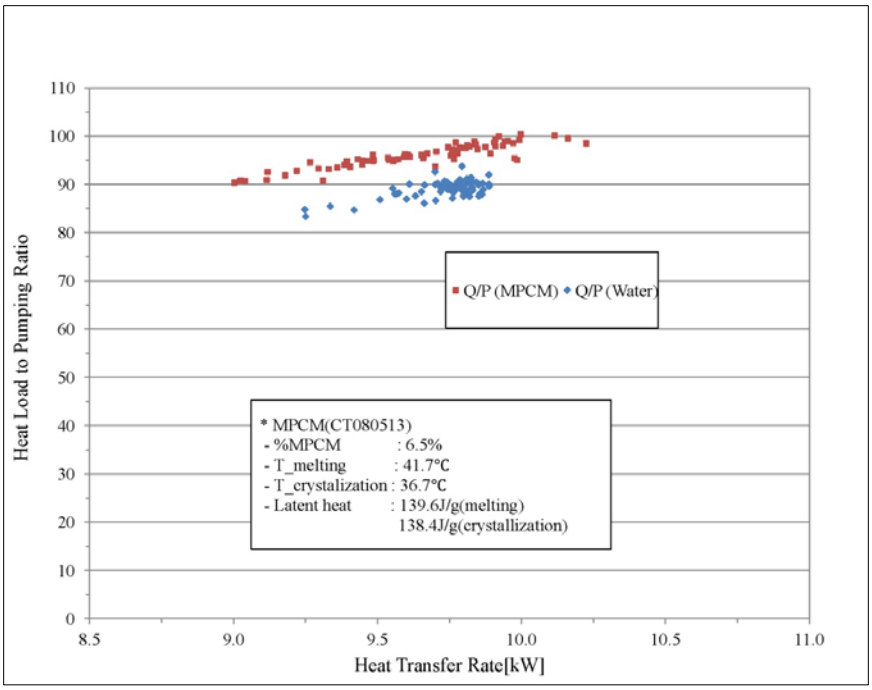

Figure 11. Hood Test 3: heat load to pumping ratio as a function of heat-transfer rate.

With respect to the durability of the MPCM, the third demonstration at Fort Hood was performed from August 14 to October 18 in 2013. MPCM capsules withstood continuous pumping conditions after circulating 164,917 cycles through a progressive cavity pump. However, the possible breakage and agglomeration of the MPCM could have occurred at the end of the test and increased the pressure drop of the system from 16 kPa to 21 kPa. Consequently, the heat-transfer performance of the system was negatively affected by the increased pumping power by 6W.

5.2 FORT DODGE

Heat-transfer tests in the instrumented loop using a mixture of water (80%) and propylene glycol (20%) as an existing heat-transfer fluid were conducted at 4.5 GPM for the baseline data. Then, three different MPCM slurries provided by Thies Technology were tested at different mass fractions of PCM. Table 7 shows the properties of each MPCM as DSC results, and test conditions for each test are shown in Table 8. The results, summarized in Table 9 and Table 10, are discussed in sections 5.2.1 – 5.2.3. As shown in Table 10, a COP improvement of more than 10% was demonstrated, and the heat load to pumping power ratio was also significantly enhanced.

Table 7. MPCM batch properties for Fort Dodge.

Test No. (Date)		**Test 1 (Feb. 26, 2013 ~ Mar. 11, 2013)**	**Test 2 (Mar. 11, 2013 ~ Apr. 2, 2013)**	**Test 3 (Apr. 02, 2013 ~ May 10, 2013)**
Batch No.		CT022313	CT030113	CT033013
Melting Temperature		-0.9 ℃	2.4 ℃	1.4 ℃
Solidifying Temperature		-0.4 ℃	1.9 ℃	-7.2 ℃
Latent Heat of Fusion	@ Melting	97 J/g	108 J/g	100 J/g
	@ Crystallization	97 J/g	104 J/g	105 J/g

Table 8. Fort Dodge HTF loop parameters.

Test No. (Date)		**Test 1 (Feb. 26, 2013 ~ Mar. 11, 2013)**	**Test 2 (Mar. 11, 2013 ~ Apr. 2, 2013)**	**Test 3 (Apr. 02, 2013 ~ May 10, 2013)**
Batch No.		CT022313	CT030113	CT033013
Dodge MPCM Slurry Loop	Mass fraction	15%	11%	10%
	Flow rate	3.3 GPM	3.1 GPM	3.0 GPM
	Temperature range	4.4 ~ 6.5 ℃	4.1 ~ 6.3 ℃	4.4 ~ 6.8 ℃
Dodge Ground Source Lop	Flow rate	9.6 GPM	9.3GPM	6.9 GPM
	Temperature range	7.0 ~ 6.2 ℃	6.7 ~ 5.9 ℃	7.4 ~ 6.4 ℃

Table 9. Fort Dodge pumping power data for three test periods.

Demonstration Period	**Test 1 (Feb. 26, 2013 ~ Mar. 11, 2013)**	**Test 2 (Mar. 11, 2013 ~ Apr. 2, 2013)**	**Test 3 (Apr. 02, 2013 ~ May 10, 2013)**
Dodge average pumping power for water/ propylene glycol (baseline)	170.6 W	169.5 W	169.5 W
Dodge pumping power for MPCMs	113.5 W	98.7 W	94.4 W
Difference between baseline HTF and MPCM	57.1 W	70.8 W	75.1 W

Table 10. Summary of Fort Dodge results.

Performance Objective	**Success Criteria**	**Experimental Results**
Coefficient of Performance (COP) for cooling mode	$\frac{COP_{MPCM\ slurry}}{COP_{Traditional\ HTF}} \geq 1.1$	1.09 (Test 1) 1.13 (Test 2) 1.09 (Test 3)
Heat load to pumping ratio	$\eta = \frac{Heat\ trasnfer\ rate}{Power\ input} \geq 4.0$	16.3 (Test 1) 19.2 (Test 2) 18.6 (Test 3)
Durability		31,953 cycles (Test 1) 49,047 cycles (Test 2) 32,593 cycles (Test 3)

5.2.1 Dodge Test 1

Figure 12 shows pumping power consumption for both HTFs. Comparison on heat load to pumping ratios between tests with the MPCM slurry and a mixture of water (80%) and propylene glycol (20%) is shown in Figure 13. Heat load to pumping ratios of MPCM slurry were higher than those of the existing HTF under constant heat-transfer rate and the improvement was up to 58.6%. These results indicate that employing MPCM slurry enhanced the heat-transfer performance of the heat pump system through reducing pumping power.

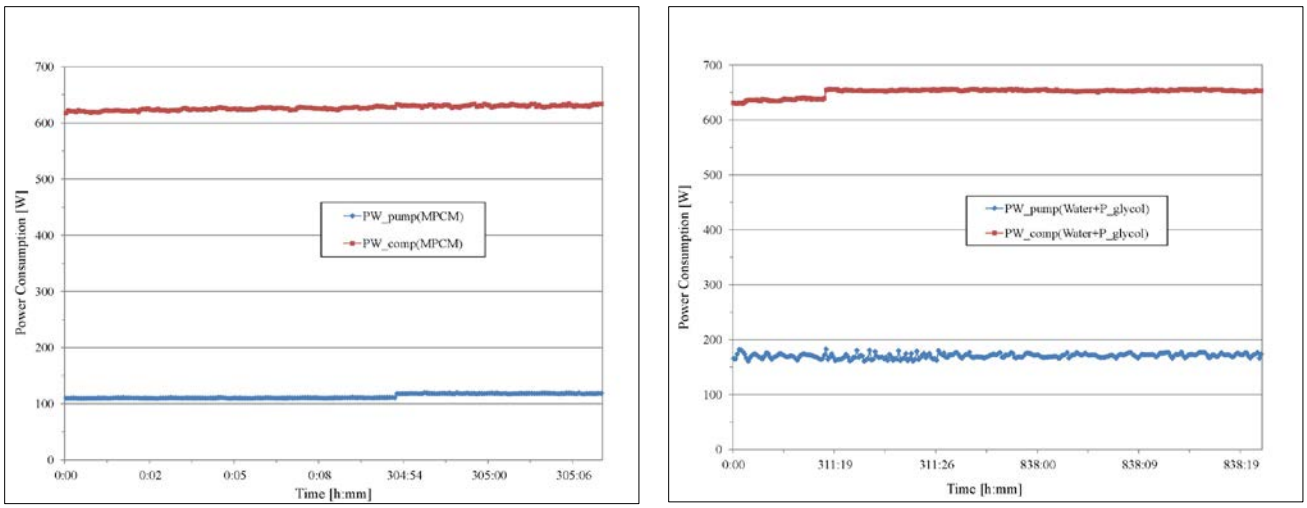

Figure 12. Dodge Test 1: power consumption in MPCM slurry and standard HTF.

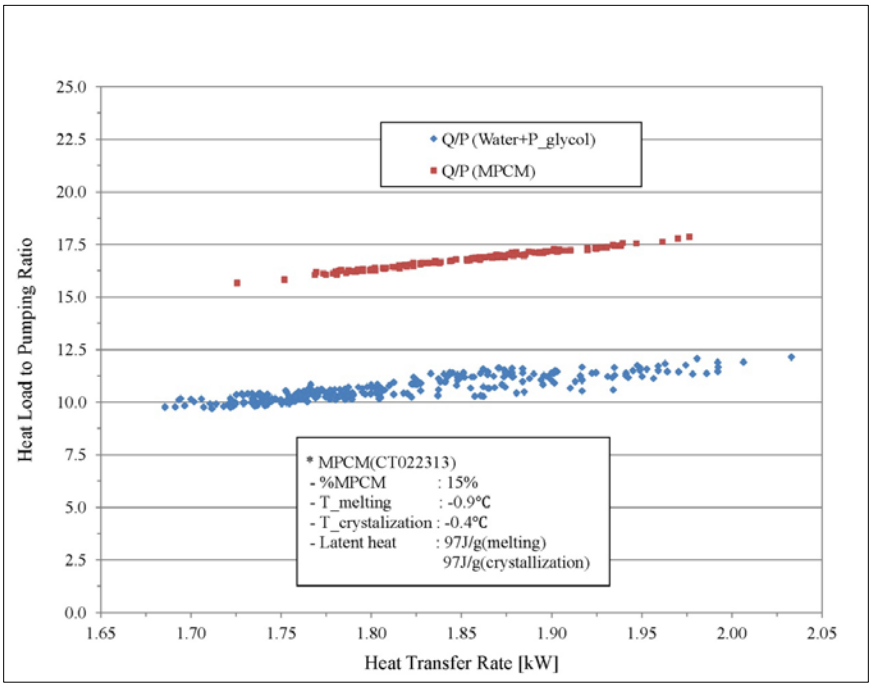

Figure 13. Dodge Test 1: heat load to pumping ratio as a function of heat-transfer rate.

Implementation of the MPCM slurry reduced 57.1 kW of the pumping power and it was 70.5% of the total save energy.

In terms of the durability of the MPCM slurry, the first demonstration at Fort Dodge was performed from February 26 to March 11, 2013. MPCM capsules withstood 31,953 continuous pumping cycles through a progressive cavity pump. There was no significant breakage of the MPCM during the first Fort Dodge test.

5.2.2 Dodge Test 2

Figure 14 shows pumping power consumption for both HTFs. Figure 15 shows that the heat load to pumping ratios in the MPCM slurry and the existing HTF tests increased when the heat transfer increased. The ratios of the MPCM slurry were higher and the improvement was up to 83.7% compared to the existing HTF. The reduced flow rate induced by the high heat carrying capacity of MPCM saved energy consumed by the pump and enhanced the heat-transfer performance of the heat pump system.

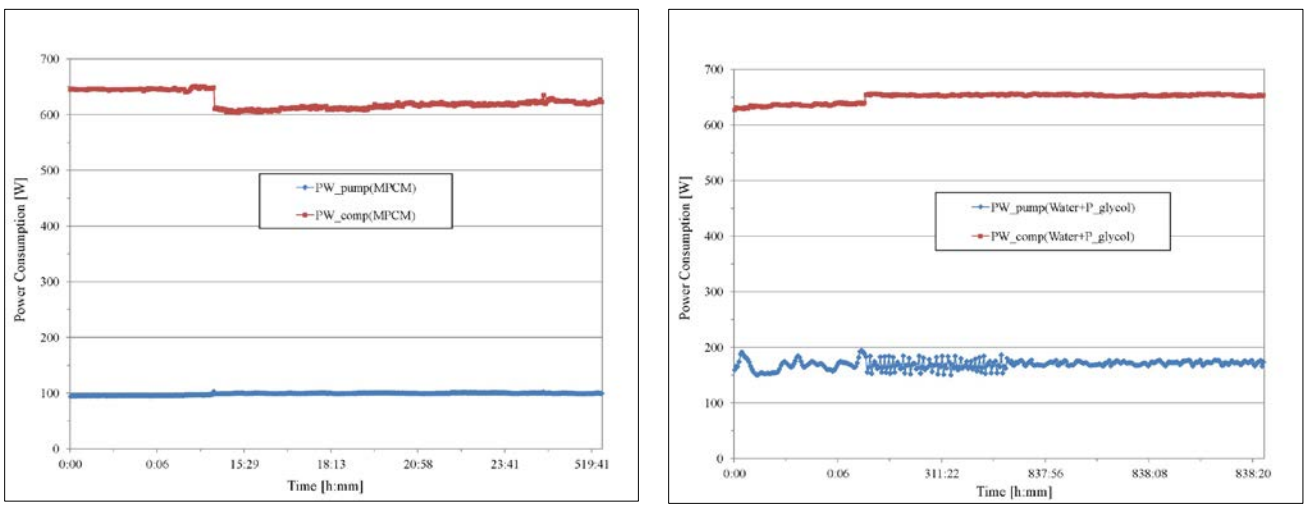

Figure 14. Dodge Test 2: power consumption in MPCM slurry (left) and standard HTF (right).

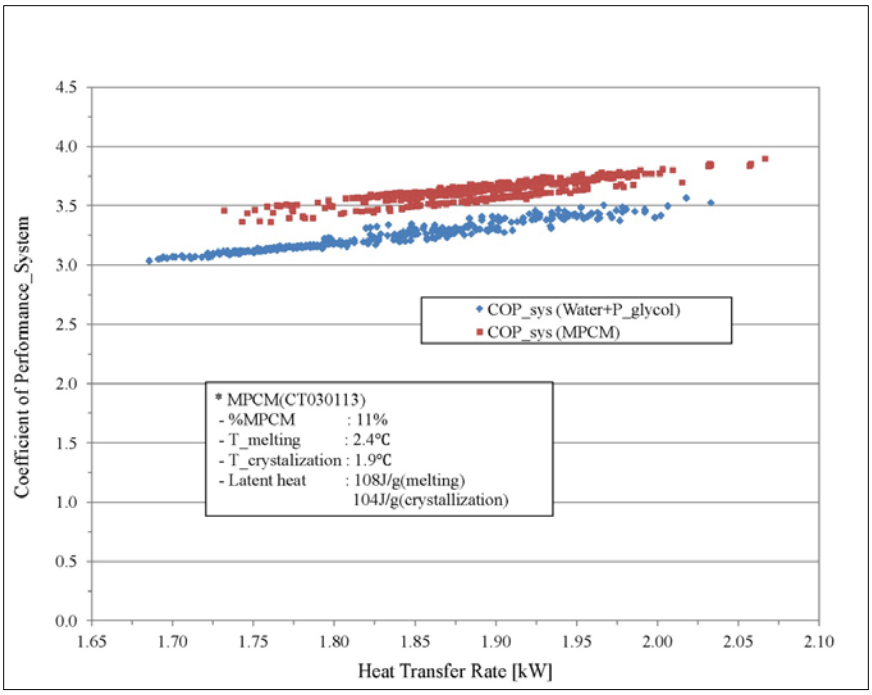

Figure 15. Dodge Test 2: heat load to pumping ratio as a function of heat-transfer rate.

With respect to the durability of the MPCM, the second demonstration at Fort Dodge was performed from March 11 to April 2 in 2013. MPCM capsules withstood continuous pumping conditions after circulating 49,047 cycles through the progressive cavity pump. Also, no breakage of the MPCM capsules was found during the test.

5.2.3 Dodge Test 3

Figure 16 shows pumping power consumption for both HTFs. Figure 17 shows comparison on the heat load to pumping ratios between the MPCM slurry and a mixture of water (80%) and propylene glycol (20%). The ratios of the MPCM slurry were higher than those of the existing HTF and the enhancement was up to 90.5%. The reduced flow rate induced by the high heat carrying capacity of the MPCM saved energy consumed by the pump. The heat load to pumping ratio of the MPCM slurry at the end of the test decreased because the pumping power increased by around 20 W. This was because a system pressure drop increased from 62.8kPa to 74.4kPa caused by the agglomeration of the MPCM. However, the heat load to pumping ratio at the end of the test was still higher than the existing HTF.

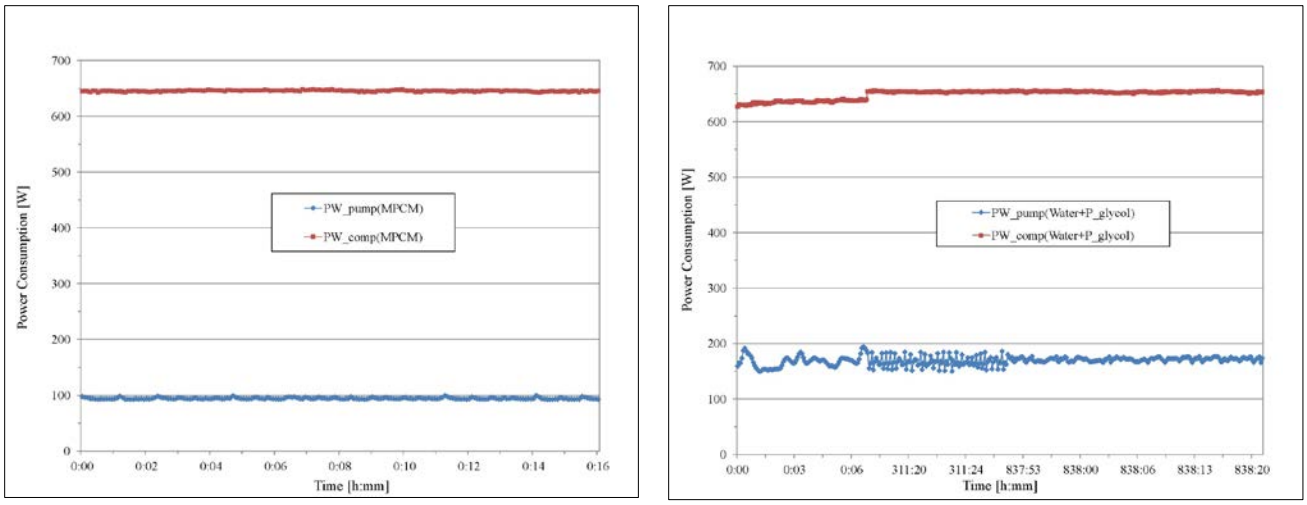

Figure 16. Dodge Test 3: power consumption in MPCM slurry (left) and standard HTF (right).

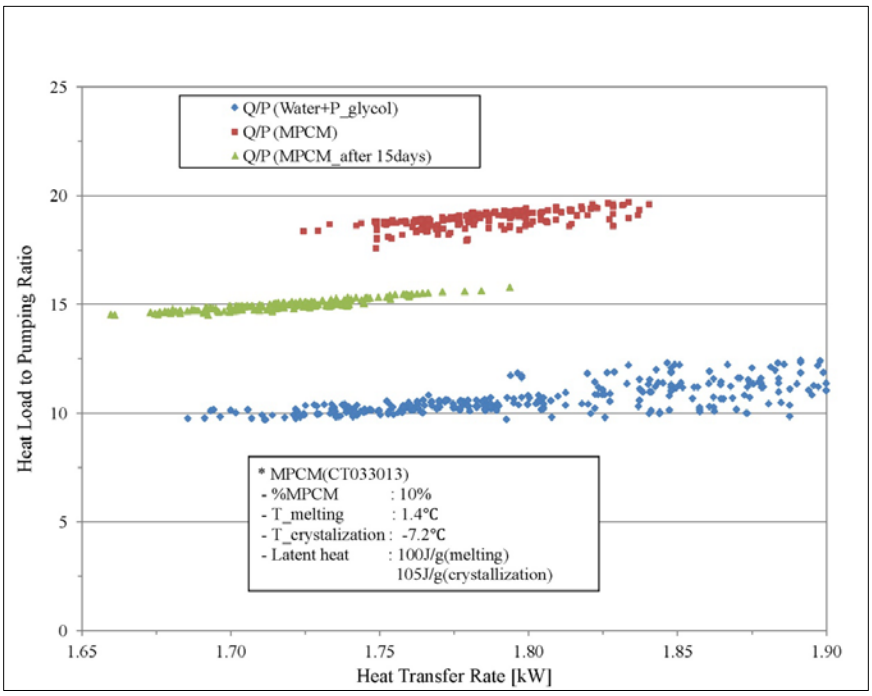

Figure 17. Dodge Test 3: heat load to pumping ratio as a function of heat-transfer rate.

The use of the MPCM slurry reduced the power input to 79.0 W and saved 10% of the pumping power consumption.

In terms of the durability of the MPCM, the capsules were circulated 32,593 cycles through the progressive cavity pump. At the end of this test, some capsule breakage and agglomeration of the MPCM occurred and caused an increased pressure drop of the system from 62.8kPa to 74.4kPa. This decreased the COP of the system through increased pumping power. In spite of this capsule degradation, the COP of MPCM slurry was still greater than the baseline COP.

6. SLURRY PERFORMANCE ASSESSMENT

The sampling data presented in Chapter 5 show that MPCM slurry enhanced the heat-transfer performance of two ground-source heat pump systems. Results indicate that MPCM slurries have the potential to become a viable heat-transfer fluid in heating and cooling applications if capsule durability can be improved and assured over long enough time scales to provide a significant economic return on investment.

The use of the MPCM slurry in the field-test loop improved the thermal performance of the ground source system. The improvement resulted from the slurry's higher heat-transfer rate, which reduced the required flow rate of the heat-transfer fluid and, consequently, saved energy through decreased pumping power.

6.1 FORT HOOD APPLICATION

The heat load to pumping ratio for MPCMs ranged from 55.2 to 95.5 and the improvement was up to 36.8% over the baseline HTF.

The MPCM capsules in each of the three trials experienced some degradation under continuous pumping conditions between 14,440 cycles through a centrifugal pump and 164,917 cycles through a progressive cavity pump.

The shortest estimated capsule service life was 1.2 years, the longest was 14 years.

6.2 FORT DODGE APPLICATION

The heat load to pumping ratio for MPCMs ranges of 16.3 to 19.2 and the improvement was up to 90.5% over the baseline HTF.

The MPCM capsules in one of the three trials experienced observable degradation. In two other trials, the MPCM capsules were circulated 31,000 and 49,047 cycles through the progressive-cavity pump with no observable degradation.

6.3 CONVERTING PUMPING CYCLES TO SLURRY SERVICE LIFE

In order to assess the performance and cost of pumped MPCM slurries in a real-world application it was necessary to calculate MPCM slurry service life when used in a full-scale GSHP system. Using the demonstration results and system parameters from the Fort Hood GSHP system, a calculation method was devised. Table 11 summarizes the results of the calculation, providing estimated MPCM service life as scaled for the entire Fort Hood ground-source loop. A detailed explanation of the calculation follows the table.

Table 11. Estimated service life of MPCM slurry.

Parameters	Unit	Values			
Total Volume of the GSHP loop at Fort Hood	Gal	6037.4			
Working flow rate of MPCM slurry flowing through the GSHP loop	Gal/min	135			
Time taken by MPCM slurry for flowing unit cycle of GSHP system	min/cycle	44.7			
		Test 1	**Test 2**	**Test 3**	**Add'l Test**
Number of possible cycles for MPCM slurry (obtained from the demonstrations)	cycle	14440	38880	164917	53500
Estimated service life of MPCM slurry	year	1.2	3.3	14.0	4.6

The following assumptions were applied:

- MPCM slurry is used as the pumped HTF throughout the entire loop of the ground source heat pump (GSHP) system at Fort Hood.
- The working flow rate of MPCM slurry is same as that of the existing HTF (i.e., water).
- The rate of any MPCM capsule degradation in the full-scale loop will be comparable to that recorded in the slurry loop tests because most shear stresses on the capsules are imposed at the pump, not by flow through the loop.

The service life of MPCM slurry (LS_{MPCMs}) used in a full-scale system is calculated using the following equation:

$$LS_{MPCMs}[year] = T_{cycle}[\min/cycle] \times N_{cycle}[cycle] \times \frac{1hr}{60\min} \times \frac{1day}{24hr} \times \frac{year}{365day}$$

T_{cycle} is time taken by MPCM slurry for flowing unit cycle of GSHP system, and is calculated by the total volume of the GSHP system loop ($V_{t,GSHP}$) divided by the working volumetric flow rate of the fluid (VFR_{GSHP}) as follows:

$$T_{cycle}[\min/cycle] = \frac{V_{t,GSHP}[Gal/cycle]}{VFR_{GSHP}[Gal/\min]}$$

$V_{t,GSHP}$ is calculated using the pipe diameters and lengths shown in the drawings for the GSHP loop provided by the Fort Hood Directorate of Public Works, which is approximately 6037.4 gallons. VFR_{GSHP} is determined by assuming the field pump (Armstrong Pump-4030BF) is working at the rated operating condition of 135 GPM at 110 ft.

N_{cycle} is the number of possible cycles for MPCM slurry, which is obtained from the demonstration results.

7. COST ASSESSMENT

This cost assessment assumes that an effective capsule service life of 10 years can be reached and assured. The longest effective service life estimated by closed-loop testing at Fort Hood was 14 years, but the variability of batch durability in this demonstration was too high for commercial application. It is reasonable to assume that at some time in the near term the capsule durability and variability will be solved. Major manufacturers in the United States and Europe are continuing to demonstrate improvements in capsule technology, and so an assumed service life of 10 years is considered reasonable for this economic assessment.

The cost of using MPCMs in a coolant loop is driven primarily by the cost of the added capsules, with an approximately 10% additional cost for an engineering assessment to optimize MPCM properties to a specific system. The cost of adding the capsules to a loop is the same as for adding other types of HTF enhancement products, such as propylene glycol. This scenario examines a heat transfer loop used in climate where water periodically freezes, so both the baseline HTF and the MPCM slurry are based on an 80/20 blend of water and propylene glycol.

The cost model for an energy-efficient application of pumped MPCM slurry in a GSHP system is summarized in Table 12.

Table 12. Cost model for an energy-efficiency technology.

Cost Element	Data Tracked During the Demonstration	Estimated Costs
Initial capital costs	Estimates made based on MPCM costs for demonstration and NREL cost survey data	$0.75 to $2.5 per pound
Installation costs	Labor and material required to install	Same as for baseline
Consumables	Estimates based on rate of consumable use during the field demonstration	Same as for baseline
Facility operational costs	Reduction in energy required vs baseline data	3% reduction in pumping energy
Maintenance	Frequency of required maintenance Labor and material per maintenance action	Same refresh rate as for baseline
Hardware lifetime	Estimate based on components degradation during demonstration	Same estimated life compared to baseline
Operator training	Operator training the same, but need additional Engineering analysis at the front end	$5 k

7.1 COST DRIVERS

As noted, the baseline HTF is assumed to be an 80/20 blend of water and propylene glycol. Baseline infrastructure costs of the coolant loops for a drop-in application is the same for the baseline HTF and the MPCM slurries. Both technologies used in this analysis are based on the 80/20 blend, so propylene glycol costs are common to both scenarios. The commercial cost of propylene glycol is about $20/gallon, so the cost of the baseline HTF becomes $4/gallon when diluted to a 20% concentration.

The cost difference between the baseline technology and the MPCM slurry, therefore, is attributable to the addition of MPCMs to the baseline HTF. One cost survey report detailed in a National Renewable Energy Laboratory report (Kosny 2013) cites the cost for organic-based MPCMs at $0.70 – $2.50 per pound. We assume a cost of $1 per pound, which is lower than the mean documented in this demonstration, but reflects a reasonable estimate given future efficiencies in manufacture. The density of MPCMs is about 7 lb per gallon (similar to paraffin). The slurry is 10% MPCMs by volume, so one-tenth of a gallon of MPCMs is needed for each gallon of the blended HTF. Therefore, each treated gallon requires about $0.70 of MPCMs. To provide a financially conservative estimate, this cost is rounded up to approximately $1 per treated gallon of slurry for raw material costs. To that cost we add $0.20 per gallon to account for the cost of the engineering assessment mentioned above, so the total slurry cost is assumed to be $1.20/gallon. The cost per gallon of coolant is expected to be lower for larger implementations as the engineering assessment cost is fixed, and the cost per volume is much lower.

The primary advantage of MPCMs is the reduction of pumping power for a given heat-transfer demand. For both field demonstrations, total power requirements were reduced by at least 3%, so that minimum energy reduction is used in the analysis. It is assumed that 70% of the electrical pumping energy is used during peak demand hours, when the most of workers are present in the building. Peak rate used is $0.12 per kWh. The balance of pumping energy is calculated at off-peak rates. $0.04 per kWh is used.

7.2 COST ANALYSIS AND COMPARISON

There are some cost savings associated with using MPCMs from a material consideration, but that is a fixed cost per loop volume, and the cost scales linearly with loop volume. However, as loop volume increases, the increase in pumping power is greater than linear. Therefore, greater pumping power savings are also realized in larger HTF loops.

Life-cycle cost calculated in this analysis uses the methodology specified in Office of Management and Budget (OMB) Circular A-94, “Guidelines and Discount Rates for Benefit-Cost Analysis of Federal Programs.”

The model system for this analysis uses a 6,000 gallon HTF loop. The initial cost is $31,200, based on $24,000 for the 20% propylene glycol (baseline case) and $7,200 for adding 10% MPCMs. The total electricity requirement for circulation pump and heat pump compressor is assumed to be 750 kW per hour. Assuming a heating season of 2,160 hours of runtime per year, the total power requirement for the system would be 750 MWh. Assuming 70% runtime at the peak electric rate ($0.12 per kWh) and 30% runtime at off-peak rates ($0.04 per kWh), the annual cost would be $155,520.

Given the stated costs and assumptions, the use of MPCMs would result in annual cost savings of $4,665, or 3% of the system total power requirement. The resulting annual savings of $4,665 is entered as a cost savings each year for the demonstrated technology as a system savings over the estimated service life of 10 years.

The spreadsheet used to calculate net present values, based on the method prescribed in OMB Circular A-94 and a 7% discount rate, is shown in Table 13. The calculated return on investment using these assumptions is 1.77. Systems of larger scale may see a higher return, and smaller systems are likely to see a smaller return.

Table 13. Ten-year return on investment using MPCM slurry in ground loop.

Return on Investment Calculation

Investment Required			31,200
Return on Investment Ratio	1.77	Percent	177%
Net Present Value of Costs and Benefits/Savings		55,195	55,195

A Future Year	B Baseline Costs	C Baseline Benefits/Savings	D New System Costs	E New System Benefits/Savings	F Present Value of Costs	G Present Value of Savings	H Total Present Value
1	24,000			4,665		26,790	26,790
2				4,665		4,074	4,074
3				4,665		3,808	3,808
4				4,665		3,559	3,559
5				4,665		3,326	3,326
6				4,665		3,108	3,108
7				4,665		2,905	2,905
8				4,665		2,715	2,715
9				4,665		2,537	2,537
10				4,665		2,371	2,371

8. SUMMARY AND LESSONS LEARNED

For the real-world conditions encountered this study has demonstrated enhanced thermal performance and net energy savings through the use of pumped MPCM slurries for both heating and cooling applications. To the authors' knowledge this is the first-ever MPCM field test involving existing full-scale systems. Using an instrumented intermediate test loop, real time variations in thermal load demand were consistently met, and characterized, for the thermal energy transfer rate range of 1.7 to 10 kW.

Lessons learned include improved knowledge of the impact of pump type on effective capsule service life, and challenges with custom formulation of first-generation commercial MPCM capsules. Although some batches of capsules used in this study were not adequately durable, increasing capsule wall thickness and using a progressive cavity pump were found to improve MPCM capsule service life. For drop-in applications, a more complete characterization of the full thermal operational envelope is necessary. This will help to better engineer and optimize the operational phase change temperatures for melting and solidification. In future systems designed specifically for use with MPCM slurries, multiple other improvements should be possible. Here, again, designing for complete PCM melting and solidification entirely within the intended heat exchanger or delimited piping network region is critical.

While with further development pumped convective thermal transfer using MPCM slurries promises significant benefits, based on the results of this demonstration they are not recommended for immediate widespread implementation.

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