Final Report

Use of MicroPCM Fluids as Enhanced Liquid Coolants in Automotive EV and HEV Vehicles

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James C. Mulligan and Richard D. Gould
Department of Mechanical and Aerospace Engineering
North Carolina State University
Raleigh, NC, 27695-7910
Voice: 919-515-5236; FAX: 919-515-7968; gould@eos.ncsu.edu

DOE Program Manager: David Hamilton
(202) 586-2314; fax: (202) 586-1600; email: david.Hamilton@ee.doe.gov

DOE Patent Clearance Granted
MP Dvorscak
Mark P. Dvorscak
(630) 252-2393
E-mail: mark.dvorscak@ch.doe.gov
Office of Intellectual Property Law
DOE Chicago Operations Office
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I. Introduction

The Department of Energy awarded Phase I funding (DE-FC02-00EE50631) for the project entitled "Use of MicroPCM Fluids as Enhanced Liquid Coolants in Automotive EV and HEV Vehicles," under the Cooperative Automotive Research for Advanced Technology (CARAT) program to Drs. J. C. Mulligan and R. D. Gould in the Mechanical and Aerospace Engineering Department at North Carolina State University (NCSU). The original project period of 04/01/2000 – 04/01/2001 was extended at no cost to 10/31/2001 due to the incremental funding schedule which did not allow NCSU to administer the subcontract to our teaming partners, David Colvin and Yvonne Byant at Triangle Research and Development Corporation (TRDC) of Raleigh, NC. Since TRDC was responsible for obtaining the microPCM materials this delayed the project schedule by approximately 6 months

The objectives of this Phase I project were to:

1. Develop a high performance phase-change material (PCM) for pumped loop cooling of EV & HEV electrical systems and for the provision of environmental cabin comfort heat,
2. To specifically investigate the efficacy of a 60°C microPCM suspension fluid consisting of microencapsulated octacosane capsules suspended in a 50/50 glycol/water carrier fluid, and
3. To evaluate the effectiveness of the suspension as a heat transfer working fluid utilizing computational methods and proof-of-concept experiments.

The following five specific tasks were identified in this Phase I effort and were completed.

1. Preparation of approximately two gallons of 23% by mass microencapsulated octacosane in a 50/50 glycol-water carrier fluid for demonstration and testing. Small quantities of four different microPCM/glycol-water mixtures were also prepared for viscosity tests.
2. Conduct differential scanning calorimeter(DSC) tests to determine the melting and freezing characteristics of octacosane phase change material.
3. Conduct viscosity tests of the 50/50 ethylene glycol/water baseline fluid and various mixture fractions of microPCM suspension fluids to determine flow characteristics.
4. Carry out computational analysis of heat transfer of baseline 50/50 ethylene glycol/water fluid and 23% microPCM suspension fluid.
5. Construct a bench scale pumped-loop demonstration to determine heat transfer and pumping characteristics of baseline and 23% microPCM suspension fluids.

II. Phase I Results

The microPCM suspension fluid has the appearance of whole milk as indicated by the photograph in Figure 1a, although it is more viscous. Two batches of dry-cake microPCM suspension fluid were tested for viscosity, suspension stability, and phase-change physics. Additives (BASF) were utilized in various concentrations to enhance suspension stability and decrease separation of the microencapsulated particles and the carrier fluid. One last batch of microPCM fluid in a wet-cake supply rather than a dry-cake supply was ordered and tested. According to the supplier, this wet-cake process has the potential of producing a stronger and more impermeable capsule wall structure. The wet cake microPCM suspension fluid proved to be more stable (i.e. less separation) and was used for all the pumped-loop studies. A photomicrograph of the microencapsulated octacosane particles developed as part of this work is shown in the photograph in Figure 1b. A 50X optical microscope was used to obtain this picture at room temperature and thus the octacosane is in the solid state. As indicated by the scale in this photograph the particle diameters vary between 10 and 30 μm . During the manufacturing process of these particles, liquid octacosane droplets ($T > 61^\circ\text{C}$) are encapsulated with a polymer coating. The dimples shown in this photograph occur due to contraction of the octacosane upon solidification.

Figure 1. a.) 23% PCM in 50/50 EG/water carrier with 1.5% BASF surfactant, b.) 10-30 μm diameter microencapsulated octacosane with $\sim 1\text{ }\mu\text{m}$ thick polymer wall – 87% core

A Perkins-Elmer differential scanning calorimeter (DSC) was used to determine the phase change properties of bulk laboratory grade octacosane. The output of this instrument is given in Figure 2. This figure shows the heat flow into the octacosane versus the temperature. A DSC controls the heat rate such that a constant temperature rise per unit time results. The scan rate used here was $5^\circ\text{C}/\text{min}$. The two peaks indicate phase change upon heating. A similar curve was measured for cooling. The left peak shows the behavior of the solid-to-solid phase change transition associated with octacosane, while the right peak shows the behavior of the solid-to-liquid phase change transition. The area under these peaks is the latent heat which for the solid-to-liquid transition was found to be approximately 260 kJ/kg . This figure also indicates that liquid starts to appear at 61.2°C and that octacosane is completely liquid at 63.7°C .

Figure 2. DSC measurements of octacosane

Viscosity measurements of the 50/50 by volume ethylene glycol/water baseline fluid and four microPCM fluids with solid mass fractions varying from 10% to 28% over a 25 - 65°C temperature range were made using a Saybolt viscometer. These measurements are shown in Figure 3 and indicate that the 23% solid fraction microPCM suspension fluid has a viscosity approximately three times greater than the baseline 50/50 ethylene glycol/water fluid. Thus the pumping power required for microPCM suspensions will be greater than for the baseline fluid and must be factored in when comparing the performance of these coolants.

Figure 3. Viscosity measurements of microPCM fluid

It is important to realize that two heat exchangers are necessary for a pumped loop cooling system; the source side heat exchanger and the rejection side heat exchanger (usually to air in an EV). A schematic of a pumped-loop cooling system is shown in Figure 4. The source side heat exchanger performance with the baseline ethylene glycol/water coolant and the microPCM suspension coolant was focus of this Phase I research. For a given coolant and mass flow rate, important parameters include the applied heat flux, $\dot{q}''$, the axial distance, x , and the inside wall surface temperature at the exit of the source side heat exchanger, $T_{s,max}$. The heat rejection side was modeled as Kays and London's air core model CF-624-5/8J heat exchanger. The average air temperature used in this study was defined as $T_{air} = (T_{air,in} + T_{air,out})/2$. Thus, T_{air} is a measure of the cabin heat air temperature.

Figure 4. Schematic of pumped loop cooling system

Figure 5 shows characteristic temperature distributions in the source side heat exchanger of pumped-loop cooling systems. The left plot of temperature versus axial distance is for a single phase fluid (i.e. the ethylene glycol/water carrier fluid) while the right plot is for a microPCM suspension fluid. The bulk fluid temperature increases with axial distance in a linear fashion for a single phase fluid, but stays nearly constant when using a microPCM suspension fluid at the proper flow rate. This is due to the nearly isothermal energy absorption of the suspended microencapsulated octacosane particles during phase change as indicated in Figure 2. In addition, the surface temperature is higher for a single phase fluid than for a microPCM suspension fluid. This is due to the higher convective heat transfer coefficient of the microPCM suspension fluid. Thus, via Newton's law of cooling, $\dot{q}'' = h(T_s - T_f)$, a smaller temperature difference is required to transfer a given amount of heat with the microPCM suspension fluid. This enhanced heat transfer effect also occurs on the heat rejection side. Thus, higher exit air temperatures result when using a microPCM suspension fluid. It is desirable for air temperatures to be greater than 35°C for use as environmental cabin heating.

Figure 5. Operation of a pumped-loop cooling system

Property	Water	Ethylene glycol, EG	50/50 EG mixture	Octacosane	23% microPCM suspension
Prandlt number	3.0	51	27	---	30
Density, ρ (kg/m ³)	985	1088	1036	900	1000
Dynamic viscosity, μ (kg/m-s)	.000471	.00517	0.00282	---	.00562
Specific heat, C_p (J/kg-K)	4184	2562	3370	600	2730
Thermal conductivity, k (W/m-K)	0.651	0.260	0.456	0.7	0.51
Kinematic viscosity, ν (m ² /s)	4.78×10^{-7}	4.75×10^{-6}	2.61×10^{-6}	---	5.64×10^{-6}
Thermal diffusivity, α (m ² /s)	1.55×10^{-7}	9.32×10^{-8}	1.24×10^{-7}	---	1.86×10^{-7}

A series of numerical experiments were conducted using this computer simulation to help understand the physics of the problem and to guide in the development of the experimental pumped-loop. Various coolant flow rates, heat inputs and tube lengths were used. The majority of these numerical experiments were conducted at a flow rate of 0.04 kg/s (5.3 lbm/min), although other flow rates were studied. Heat rates of 400, 600 and 800 watts for a 64 inch (1.61 m) long tube and 150 W for a 24 inch (.67 m) long tube were considered. Figure 6 shows temperature contour plots of the 50/50 ethylene glycol/water baseline fluid and the 23% microPCM suspension fluid in the 64 inch long tube with a heat rate of 600 W (which gives a heat flux of 8.78 kW/m^2) and mass flow rate of 0.04 kg/s.

Figure 6. Temperature contour plots in $\frac{1}{2}$ inch diameter tube ($L=1.6\text{m}$, 600 W , $\dot{m}=0.04 \text{ kg/s}$)

The same scale is used for both fluids and thus it is easy to see that the microPCM suspension fluid is cooler next to the wall than the 50/50 ethylene glycol/water fluid. In addition, the thinner thermal boundary layer for the microPCM suspension fluid indicates a higher convective heat transfer coefficient. In fact, the numerical experiments indicate that heat transfer coefficient increases by nearly 55% when using this microPCM suspension fluid. Since the viscosity of the microPCM suspension fluid is approximately 3 times that of the baseline 50/50 ethylene glycol/water fluid the augmented heat transfer is at the expense of approximately a factor of 3 higher pumping power.

Figures 7 and 8 show the predicted surface and bulk fluid temperatures as a function of axial distance for various heat rates in the 64 inch long heated tube for the baseline 50/50 ethylene glycol/water fluid and the 23% microPCM suspension fluid, respectively. Figure 7 shows that the surface temperature is approximately 10°C lower at the exit of the tube when using the microPCM suspension as opposed to the baseline fluid. It also shows that for a constant heat flux

boundary condition the surface temperature continues to rise with axial distance. Finally, these results show that surface temperatures may exceed the reliability limits of the electronic components being cooled. As mentioned earlier, the effect of the copper wall was not included in this analysis. Figure 7 indicates the temperature rise of the coolant is approximately 2.5°C less for the microPCM suspension than with the baseline fluid. This is a 65% reduction in the fluid temperature rise – a significant change.

Figure 6. Predicted tube surface temperature versus axial position

Figure 7. Predicted bulk fluid temperature versus axial position

A standard way to compare the performance of thermal systems is plot the pumping power versus temperature rise. Thus, enhanced heat transfer due to rough wall, ribs, fins, or a higher viscosity fluid (like the microPCM suspension used here) can be accounted for when comparing different systems. It is important to note that the pumping power reported in these figures is that necessary to overcome frictional effects in the source heat exchanger only, and not the entire pumped-loop system. Figures 8 and 9 show the predicted performance of the microPCM suspension fluid and the baseline ethylene glycol/water fluid. Figure 8 is for the 24 inch long tube (.67 m) and a heat rate of 150 watts, while figure 9 is for the 64 inch long tube (1.6 m) and a heat rate of 400 watts. These predictions use Kays and London's air core model CF-624-5/8J heat exchanger correlations to predict air temperatures. The ordered pairs in these figures denote the maximum surface temperature and the average air temperature as it blows over the air-side of the heat rejection heat exchanger. A cooling system that maintains reliable electronic component temperatures ($T_{s,max} < 70^{\circ}\text{C}$) and provides environmental cabin heat at temperatures greater than 35°C (95°F) would be ideal for electric vehicles as no electrical heating elements, which use crucial battery power, would be necessary for cabin heating requirements. Thus, a cooling system with $T_{s,max} \approx 70^{\circ}\text{C}$ and $T_{air} > 35^{\circ}\text{C}$ gives a $\Delta T = T_{s,max} - T_{air}$ between 30 and 40 $^{\circ}\text{C}$ on these plots. It is clear from the predictions shown in figure 8 that the ethylene glycol/water fluid will not provide environmental heat at a high enough temperature to be useful, whereas the microPCM suspension fluid will. Figure 9 shows that the microPCM suspension fluid is beginning to work at the higher pumping powers, but that the ethylene glycol/water fluid will not provide hot air enough for cabin heating.

Figure 8. Predicted bulk fluid temperature versus axial position ($L = .67 \text{ m}$)

Figure 9. Predicted bulk fluid temperature versus axial position ($L = 1.6 \text{ m}$)

A fully instrumented bench scale test flow loop was constructed to validate the use of a microPCM suspension fluid for EV and HEV electronics cooling. Figure 10 shows a schematic of this system. A $\frac{1}{2}$ inch diameter, thick walled copper tube wrapped with an electrical heating tape was used to model the source side heat exchanger. The heating tape was used to simulate the heat dissipated by EV or HEV electrical components. The outer surface of the heating tape was well insulated so that virtually all of the heat flows into the coolant.

Figure 10. Bench scale test flow loop apparatus

Energy balances indicate that approximately 90% of the applied heat goes into the enthalpy rise of the fluid. This is quite good for flow loop experiments and thus validates the precision of the measured results. Two tubes, one 24 inches (.67 m) long and the other 64 inches (1.61 m) long were used. The 24 inch tube was used to model a 4 pass (each 6 inch long) cold plate heat exchanger while the 64 inch tube was used to model a 10 pass cold plate. A variable speed Laing pump was used to control the coolant flow rate. A Micro-motion mass flow meter was used to measure the mass flow rate to within $\pm 1\%$. Finally, a variable resistor was used to set the heat input, while a power meter was used to measure the electrical power going to the heater. Important parameters include the applied heat rate and the mass flow rate. Pressure transducers were used to measure the pressure drop around the flow loop and thermocouples were used to measure fluid temperatures at the inlet and exit of the source and sink heat exchanges. In addition, the exit source side heat exchanger surface temperature was also measured. All transducers and thermocouples were interfaced to an HP 34970A datalogger which was interfaced to a PC using an IEEE 488 interface. A liquid cooled heat exchanger was used for the heat sink in this study since the focus of this work was on the source side heat exchanger.

A series of experiments were conducted using the pumped flow loop system described above. Tests using 50/50 ethylene glycol/water (single phase fluid) were conducted to obtain baseline performance data. These were compared to the measurements when using the 23% mass fraction microPCM suspension fluid (phase change fluid). Figure 11 shows the temperature difference between the outlet and inlet fluid temperature at various fluid flow rates when 600 W of heat were applied to the 64" (1.61 m) long tube.

Figure 11. Bench scale test flow loop apparatus

At the lowest flow rate the temperature difference is 9°C for the microPCM suspension fluid and 11°C for the baseline fluid. It should be mentioned that the measured enthalpy change of the fluid between the exit and inlet agreed to within 10% of the applied heat rate. As the mass flow rate is increased the outlet to inlet temperature difference decreases as expected, however, the

microPCM suspension fluid has a smaller temperature difference for all flow cases. Thus, the microPCM suspension fluid yields a more uniform temperature throughout the source heat exchanger as illustrated in figure 5. In fact, the temperature of the microPCM suspension fluid increases by only 1.43°C at a mass flow rate of .042 kg/s, whereas the baseline fluid has a temperature rise of 4°C. Although these values may seem small it is important to note that a 63% decrease in temperature rise results when substituting the baseline fluid for the microPCM suspension fluid. The ability of microPCM suspension fluids to absorb energy in a nearly isothermal manner is clearly demonstrated. It should also be noted that the numerical predictions shown in figure 4 agree extremely well with these experimental measurements and thus the predictive capability of our computer simulations are extremely valuable.

The next set of experiments, using the 23% microPCM suspension fluid, involved varying the source side inlet temperature and measuring the outlet fluid temperature and the maximum outside surface temperature of the copper tube (i.e. source side heat exchanger). The mass flow rate and the applied heat rate were maintained at 0.04 kg/s and 600 W, respectively, throughout these experiments. Figure 12 shows these measurements with the lower curve denoting the measured outlet fluid temperature and the upper curve denoting the measured maximum surface temperature. The dashed lines indicate the temperatures that would result if a single phase fluid having the same bulk thermophysical properties as the microPCM suspension fluid were used.

Figure 12. Bench scale test flow loop apparatus

The following important observations can be made: 1.) the outlet temperature is 3-4°C lower than with a single phase fluid, 2.) the exit fluid temperature is nearly constant as the inlet temperature is varied from 57 – 59.5°C, 3.) the maximum surface temperature is 4-5°C lower than with a single phase fluid, and 4.) the temperature difference between the maximum surface temperature and the exit fluid temperature decrease from approximately 5°C to 3°C indicating a 66% increase in convective heat transfer coefficient. The fact that the maximum source surface temperature decreases by approximately 5°C results in a lower electronic component temperature

(and similarly a higher heat rejection surface temperature which allows for provision of environmental cabin heat). The discrepancy of measured surface temperatures with predicted surface temperatures seems to suggest that the computer simulations are in error. In fact, the computer simulations consider an infinitely thin tube wall whereas the experimental flow loop uses thick walled copper tubing. Thus, a direct comparison of this value should not be made, but the predicted trends should be correct.

III. Summary of Phase I Accomplishments

1. A 60°C micro-PCM fluid consisting of a suspension of microencapsulated octacosane and glycol-water has been successfully produced and tested.
2. Numerical experiments and actual pumped-loop testing indicate significantly reduced surface temperatures and reduced fluid temperatures. THIS IS A MAJOR MILESTONE because it confirms that the fluid will operate in such a way to provide the possibility of environmental air heating at reduced electrical component temperatures.
3. The microPCM fluid is currently being circulated without incidence of degradation or microcapsule damage, which can occur due to pumping stress and/or expansion and contraction fatigue. THIS IS A MAJOR MILESTONE.
4. The numerical experiments indicate the fluid is a candidate alternative for electric systems which need to be maintained below 70 C and which are the source of 35 C heat for environmental temperature control.

In summary, the octacosane based microPCM suspension fluid developed in this study accomplishes the dual technology goals of cooling EV electronic components and provision of environmental cabin heat. The major barriers to its adoption in EV and HEV cooling systems are cost and long term durability. The small quantity cost of the octacosane based microPCM suspension fluid developed here was approximately \$630/gallon. The breakdown costs are as follows

Material	Lot size	Cost
Pure octacosane	11 lb	\$68/lb $\Rightarrow$ \$117/gal of 23% suspension
Microencapsulation	5 lb	\$300/lb $\Rightarrow$ \$516/gal of 23% suspension

IV. Cost Projection Estimates of MicroPCMs

Microencapsulation

Triangle Research and Development Corporation (TRDC), our teaming partner, has been working in the microencapsulation area for approximately 15 years and have tracked the price of microencapsulation with lot time and lot size. Figure 13 shows the actual year 2000 cost of microencapsulation versus lot size for macro-capsules (diamond symbol). A power law equation fits this cost data (solid blue line) with excellent agreement. The square symbol shows the small quantity (5 lb) cost for the microencapsulation of pure octacosane in this project, while the

triangle shows the small quantity (5 lb) cost for microencapsulation of pure eicosane (melts at 20 C and thus is easier to microencapsulate). Assuming that microencapsulation of octacosane is similar to that for micro-capsules, allows one to realistically extrapolate the large lot microencapsulation cost to approximately \$20/lb (\$34/gal of 23% suspension)

Figure 13. Microencapsulation costs versus lot size in the year 2000

Figure 14 shows small lot size microencapsulation cost versus year. The cost of microencapsulation appears to decrease by a factor of two every ten years. Thus, one can realistically expect today's microencapsulation cost to drop by a factor of two over the next decade, when electric and hybrid electric vehicles will become a larger part of the vehicle fleet.

Figure 14. Small lot size (5 lb) microencapsulation costs versus year.

Based on this data and very conservative estimations, we believe that large lot microencapsulation costs will be less than \$10/lb (\$17/gal of 23% suspension) within the next decade. Emerging technologies may accelerate this reduction in cost.

Phase change materials (PCMs)

Pure paraffins (99% pure) are relatively expensive compared with technical grade paraffins (90-95% pure). Our experience with eicosane is as follows:

Material	Purity	Lot size	Cost
Pure laboratory grade eicosane	99%	5 gal (liquid)	\$24.50/lb
Technical grade eicosane	90 – 95%	5 gal (liquid)	\$3.20/lb

The technical grade paraffin will have a wider phase change temperature range due to impurities, but are approximately a factor of seven less expensive. Technical grade octacosane is available upon request, and should have a cost reduction similar to that of the eicosane. Thus we believe that the small quantity cost for technical grade octacosane should be approximately \$10/lb (\$17/gal of 23% suspension). Large quantities purchases should reduce this cost to between \$5-7/lb (\$9-12/gal of 23% suspension).

In summary, it is believed that the prices will drop for large quantity purchases to approximately \$9-12/gal for technical grade octacosane and to \$10-17/gal for microencapsulation. This corresponds to approximately \$19-29/gal for a technical grade octacosane based microPCM suspension fluid. We have recently found a low-cost alternative to octacosane called Polywax (Baker Petrolite, Sugar Land, Texas). It is currently produced in large quantities and costs approximately \$2/gal. A Polywax based microPCM suspension produced in large quantities would cost approximately \$12-19/gal which we believe makes this suspension economically viable for large scale EV and HEV usage.

V. Future directions

Development of a low cost microPCM suspension fluid and long term durability studies will be the primary goal of future work. Development of a Polywax® based microPCM suspension fluid consisting of microencapsulated Polywax® in an environmentally friendly propylene glycol/water carrier fluid at the extremely attractive cost of ~\$12/gal is the goal. In addition, a long term durability study will be conducted to determine when the microPCM fluid should be replaced. A user friendly computer simulation design tool will be developed for microPCM based cooling systems. Finally, a full scale microPCM based prototype cooling system, sized for the EV-1 (i.e. 5 gal/min, 2 kW steady state dissipation with 10 kW instantaneous heat dissipation) will be developed, tested and delivered to an interested commercial entity.