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DEVELOPMENT OF ADVANCED LOW-TEMPERATURE HEAT TRANSFER FLUIDS FOR DISTRICT HEATING AND COOLING

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A Novel Concept for Heat Transfer Fluids Used in District Cooling Systems

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ABSTRACT

Low-temperature phase-change materials (PCMs) were mixed with water to enhance the performance of heat transfer fluids. Several PCMs were tested in a laboratory-scale test loop to check their suitability to district cooling applications. The phase-change temperatures and latent heats of fusion of tetradecane, pentadecane, and hexadecane paraffin waxes were measured using a differential scanning calorimeter. The heat of fusion of these materials is approximately 60% of that of ice. They exhibit no supercooling and are stable under repeated thermal cycling. For 10% and 25% PCM-water slurries, the heat transfer enhancement was found to be approximately 18 and 30 percent over the value of water, respectively. It was also found that, in the turbulent region, there is only a minor pumping penalty from the addition of up to 25% PCM to the water. It was demonstrated that pentadecane does not clog in a glass-tube chiller, and continuous pumping below its freezing point (9.9°C) was successfully carried out in a bench-scale flow loop. Adding PCM to water increases the thermal capacity of the heat transfer fluid and therefore decreases the volume that needs to be pumped in a district cooling system. It also increases the heat transfer rate, resulting in smaller heat exchangers. Research is continuing on these fluids in order to determine their behavior in large-size loops and to arrive at optimum formulations.

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Low-temperature phase-change materials (PCMs) were mixed with water to enhance the performance of heat transfer fluids. Several PCMs were tested in a laboratory-scale test loop to check their suitability to district cooling applications. The phase-change temperatures and latent heats of fusion of tetradecane, pentadecane, and hexadecane paraffin waxes were measured using a differential scanning calorimeter. The heat of fusion of these materials is approximately 60% of that of ice. They exhibit no supercooling and are stable under repeated thermal cycling. For 10% and 25% PCM-water slurries, the heat transfer enhancement was found to be approximately 18 and 30 percent over the value of water, respectively. It was also found that, in the turbulent region, there is only a minor pumping penalty from the addition of up to 25% PCM to the water. It was demonstrated that pentadecane does not clog in a glass-tube chiller, and continuous pumping below its freezing point (9.9°C)was successfully carried out in a bench-scale flow loop. Adding PCM to water increases the thermal capacity of the heat transfer fluid and therefore decreases the volume that needs to be pumped in a district cooling system. It also increases the heat transfer rate, resulting in smaller heat exchangers. Research is continuing on these fluids in order to determine their behavior in large-size loops and to arrive at optimum formulations.

INTRODUCTION

This paper described the development of low-temperature heat transfer fluids to be used in district heating systems to yield increased performance. It focuses on studying the potential of using phase-change-material (PCM) particles suspended in water at temperatures used in district cooling loops. The objective of the project is to develop heat transfer fluids that (1) have improved heat transfer properties and (2) reduce the volume of fluid that needs to be pumped.

The proposed method is to add to the carrier fluid a large quantity (up to twenty-five percent) of a PCM whose phase-change or phase-transition (melting/freezing) temperature is between the send-out and the return temperature of conventional district cooling systems. The increased thermal capacity of the PCM/water mixture decreases the pumping volume required and therefore results in savings in pumping energy, which constitutes a large fraction of the total input energy of district cooling systems. Better mixing of the carrier fluid due to the presence of solid particles tends to increase the convective heat transfer coefficient, which decreases the size of the heat exchangers required at both ends of a district cooling system. It is hoped that the proposed heat transfer enhancement technique does not require any hardware retrofits or modification of heat exchangers, and that it can be used concurrently with other passive or active heat transfer augmentation techniques.

SELECTION AND TESTING OF PHASE-CHANGE MATERIALS (PCMs)

In order to maximize the saving in pumping energy that can be obtained from using a phase-change material (PCM) slurry as the primary heat transfer medium in a district cooling system, the phase transition temperature of the PCM should be between 3°C and 10°C, depending on the design temperature of the primary cooling loop. Its latent heat should be as high as possible, it should form a slurry with water, it should exhibit little or no supercooling, be stable under repeated thermal cycling, be inexpensive, and preferably be chemically inert and non-toxic. Paraffin wax satisfies most of these conditions, and it has been previously used as thermal storage material (Ref's. 1, 2). These waxes are well characterized. They consist of simple, long molecular chains of CH_2 units with a CH_3 unit at each end. The exact phase transition temperature depends on the length of the chain, increasing with increasing chain length.

The desirable phase transition temperature for the district cooling application is straddled by the phase transition temperatures of tetradecane (5.8°C) and pentadecane (9.9°C). Thus it seems plausible that a mixture of these two waxes may represent a good choice. Since pentadecane is relatively expensive, mixtures of hexadecane (phase transition temperature of 18.1°C) with tetradecane were also believed to be worthy of investigation. Tests of different mixtures of tetradecane with both pentadecane and hexadecane were therefore performed in a differential scanning calorimeter (DSC) (Fig. 1, Ref. 3). Tests in that instrument simultaneously yield values for phase transition temperature and latent heat during melting. Figure 2 shows a typical test result for pure tetradecane. The indicated values of 6.3°C and 207 J/g compare well with values from Reference 4 of 5.8°C and 206 J/g, respectively. Phase transition temperatures for tetradecane/pentadecane and tetradecane/hexadecane mixtures as a function of the PCM mix are shown in Figures 3 and 4, respectively. Exploratory tests for latent heat showed that some of the mixtures have slightly lower latent heats than either of the two pure waxes used in the mixture, but further testing is required to determine the exact variation of latent heat with the percentage of tetradecane in the mix.

The significant increase in heat capacity of the heat transfer fluid with increasing PCM content is illustrated in Figure 5 for a temperature change of 15°C. The figure shows that the addition of 25% pentadecane by volume increases the volumetric heat capacity by almost 40%.

BENCH-TYPE FLOW LOOP

Experimental Setup

In order to determine whether slurries containing high fractions (up to 25%) of the selected phase change material could be pumped without solidifying inside the piping, tests were performed using 0, 10, and 25% slurries of pentadecane in water. Figure 6shows the schematic diagram of the test flow loop. It consists of a 5-gallon reservoir, a micro pump, a surge tank, a main test section (labelled "Heat Exchanger" in the figure), and a constant-temperature bath that acts as a chiller. The slurry is cooled below the freezing temperature of the wax in the main test section, and it is reheated above that temperature in the 5-gallon reservoir by using building service hot water. The test section consists of a coiled glass tube, while the remainder of the loop consists of tygon tubing. The behavior of the test slurry can therefore be observed through the entire test loop. The test section is a one-pass single-tube heat exchanger with a shell diameter of 3.5 cm and a tube diameter of 0.59 cm. The radius of curvature and the length of the coiled glass tube are 1.88 cm and 2.35 m, respectively. To minimize the effect of pumping-head fluctuation on pressure drop measurements, a surge tank with an inside diameter of 5 cm and a height of 30 cm was installed between the pump and the test section.

To measure the bulk temperatures of the well-mixed test fluid, four mixing chambers were installed at the inlets and outlets of the heat exchangers, into which four 20-gauge thermocouple wires were mounted as shown in Figure 6. Two 2-mm diameter holes are located at the inlet and outlet of the coiled tube to mount pressure taps. The test fluid inside the reservoir is stirred by a mixer mounted from the top of the reservoir. Flow rates are controlled by two valves, one located on a by-pass line and the other on the recirculating flow loop. The reservoir, the test section, and all tygon tubing are insulated with a one-inch-thick fiberglass blanket to minimize parasitic heat losses.

Calibration test runs were performed with distilled water to provide reference data for comparison with those of PCM mixtures. Test runs with two different concentrations of PCM slurry (10% and 25%) were carried out at two different inlet temperatures (20°C and 11°C) to the main test section. For the high-temperature case (indicated by 10%H in Figures 8 through 10), the inlet temperatures of the tube-side and shell-side were maintained at 20°C and 13-14°C, respectively. For the low temperature case (indicated by 10%L), the inlet temperatures of the tube-side and shell-side were maintained at 11°C and 4-5°C, respectively.

Analytical Evaluation

The essential quantities to be measured during the tests were bulk temperatures at four locations (i.e., inlets and outlets at both tube-side and shell-side), pressure drop across the test section, and mass flow rate for the calculation of the Reynolds number. The overall heat transfer coefficient U was obtained from

$$U = Q/(A\ \Delta T_m) \tag{1}$$

where Q is the heat transferred from the water chiller to the test fluid, A is the contact area between the two fluids, and ΔT_m is the log mean temperature difference defined by

$$\Delta T_m = [(T_{out,t}-T_{out,s}) - (T_{in,t}-T_{in,s})]/\ln[T_{out,t}-T_{out,s})/(T_{in,t}-T_{in,s})] \tag{2}$$

where the subscripts s and t refer to the shell and the tube of the heat exchanger, respectively. The heat Q was calculated calorimetrically from

$$Q = m\ C_p\ \Delta T \tag{3}$$

where m is the mass flow rate of the constant-temperature bath (measured and kept constant by means of a circulator pump not shown), Cp is the specific heat of water, and ΔT is the temperature difference between T/C 2 and T/C 1.

The use of the log mean temperature difference, ΔT_m, in a situtation where both sensible and latent heats are involved may, at first, appear open to question. The concept, however, is so universally used in heat exchanger analysis and design that the authors felt no useful purpose would be accomplished by defining a different parameter. They are aware of the fact that its use in equations (1) and (2) may not be theoretically correct and that the coefficient, U, derived by these equations is an "apparent heat transfer coefficient," which may not be directly relatable to heat transfer coefficients as usually defined for sensible heat transfer only. However, since the purpose of the present development is to improve total heat transfer, and since that will be compared to the heat transfer calculated in the usual manner according to equations (1) and (2), the authors believe that the use of the apparent heat transfer coefficient, U, is justified since it correctly describes the amount of heat transferred in the same heat exchnager with different heat transfer fluids.

The Fanning friction coefficient f was calculated from

$$f = \tau_w / (0.5\ \rho\ u^2) \tag{4}$$

where the wall shear stress τ_w is defined as $(\Delta P\ D)/4L$, D and L are the tube-side diameter and length, respectively, ΔP is the pressure drop across the test section, ρ is the density, and u is the flow velocity. The Reynolds number, Re, was calculated from the average mass flow rate m:

$$Re = 4\ m / (\pi\ \eta\ D). \tag{5}$$

ENLARGED FLOW LOOP

In preparation for full-scale testing, a somewhat larger flow loop than the bench-size loop of Figure 6 was constructed. It will incorporate a refrigeration-type water chiller of approximately 10.5-kW (3-ton) capacity. However, procurement of that chiller had to be postponed until additional funds became available to the project. The loop in its present configuration without the chiller is shown schematically in Figure 7.

The PCM/water mixture in the main test loop is cooled in Bath 3, thoroughly mixed in Bath 1, and heated in a copper tube wrapped by electric resistance heating wire. Valves 1 and 2 are used to modulate the output of the constant-speed Pump 1 to vary the loop flow rate. The heat exchanger in Bath 3 is designed to have sufficient surface for the future chiller. It consists of four parallel tubes to reduce pressure losses. The heating section consists of a 3/8-inch ID copper tube spiral-wrapped by three parallel nichrome wires, 0.65 mm in diameter. The space between two adjacent wires is 0.668 cm. The nichrome wires are insulated from the copper tube by a thin layer of epoxy. In order to operate the test loop before arrival of the chiller, a temporary installation comprising Bath 2 and a constant-temperature circulating bath (CTCB) was made. It is the function of the ice blocks and the cold water reservoir in Bath 2 to provide a heat sink far in excess of the capacity of the 0.5-kW (1700-Btu/h) capacity of the CTCB. The cooling rate of the test loop in Bath 3 is governed by the action of Pump 2 modulated by Valves 3 and 4.

RESULTS AND DISCUSSION

The pumpability of pentadecane-based PCM/water mixtures at various temperatures near the PCM freezing temperature of 10°C is a requirement for the suitability of the proposed improved heat transfer fluid. Prior to installing the insulation blanket on the heat exchanger, preliminary bench tests were therefore performed to determine whether PCM in water at that temperature would clog the tube. Even with a 25% slurry, the 0.59 cm tubing and glass coil never became clogged, and full flow was maintained throughout the test. This established the feasibility of using slurries containing paraffin wax up to 25% by volume for the desired application. While the tests were performed with pentadecane and water up to now, they will be repeated with tetradecane /pentadecane and tetradecane/hexadecane mixtures and water to vary the freezing temperature further below 10°C. Because of the similarity of the three waxes, no difference in flow behavior is expected.

During these preliminary experiments it was also observed that pentadecane does not readily dissolve in water. Most of it floated to the surface due to its low density of 0.8. Mechanical and chemical means were used to prevent this separation. Both were successful, and a suitable method will be developed once the optimum PCM/water mixture has been selected.

Figure 8 shows the friction coefficient f of distilled water and PCM/water mixtures at high inlet temperatures (without actual phase change). The data points for distilled water (solid circles) show a linear dependence on the Reynolds number at values above 3000. It is interesting to note as an aside that the friction coefficient for the coiled tube was found to be much higher than that for a straight tube, which may be attributable to the secondary flow caused by centrifugal force The figure shows that, in the fully turbulent region above Re=3000, the friction coefficients for 10% and 25% PCM/water mixtures are almost equal to those for water. Thus, only a small pumping penalty may have to be paid in order to use a PCM/water slurry for district cooling.

Figure 9 shows the apparent heat transfer coefficient U as a function of the Reynolds number for the "high" temperature case. The tube-side inlet temperature was 20°C, and the outlet temperature varied beween 13°C and 14°C depending on the flow rate. The solid circles represent data for water, while open circles and squares represent data for 10% and 25% PCM mixtures, respectively. The figure shows that the heat transfer enhancement is approximately 18 percent in the range of

Reynolds numbers from 3,000 to 8,000. It should be noted that, even though the tube-side outlet temperature is well above the freezing temperature of the PCM, local freezing of the PCM occurs at the tube wall, resulting in enhancement of heat transfer.

Figure 10 shows the apparent heat transfer coefficients obtained for the "low" temperature case, where the tube-side inlet and outlet temperatures were maintained at 11°C and 4°C to 5°C, respectively. For the 10% PCM/water mixture, the heat transfer enhancement is approximately 18 percent compared to the value for water, while for the 25% mixture it is approximately 30 percent in the range of Reynolds numbers from 3,000 to 8,000.

CONCLUSIONS

The performance enhancement resulting from the addition of a low-temperature phase change material (PCM) to the water in the heat transfer loop of a district cooling system was verified in small-scale laboratory tests. Adding 25% PCM by volume raises the volumetric heat capacity of a heat transfer fluid undergoing a Δt=15°C by almost 40%. For 10% and 25% PCM/water mixtures, the increase in apparent heat transfer coefficient over that of water is approximately 18 and 30 percent, respectively. In the turbulent region, there is only a small pumping penalty from the addition of up to 25% PCM to water. Pentadecane does not clog heat exchanger tubing in the turbulent flow region for mixtures containing as much as 25% PCM.

The concept of using low-temperature PCMs may represent a potentially practical and economical technology for district cooling. The enhancement technique presented here is not based on any heat exchanger modifications and, it is hoped, can be used concurrently with other passive or active heat transfer augmentation techniques.

FUTURE WORK

Future work involves increased heat transfer rates in the enlarged experimental flow loop and the use of tetradecane/pentadecane and tetradecane/hexadecane combinations to optimize the PCM/water mixture for district cooling applications. The use of friction-reducing additives, such as high-molecular-weight polymers, surfactants, and large-aspect-ratio fibers together with low temperature PCMs will also be investigated.

ACKNOWLEDGEMENT

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NOMENCLATURE

- A: contact area between two fluids, surface area of tube-side tube
- D: tube-side diameter
- f: Fanning friction coefficient, defined in Eq.(4)
- L: length of tube-side tube
- m: mass flow rate at tube-side
- ΔP: pressure drop across test section
- Q: heat transferred from tube-side to shell-side
- Re: Reynolds number, defined in Eq.(5)
- $T_{in,t}$: inlet temperature at tube-side

$T_{in,s}$: inlet temperature at shell-side
$T_{out,t}$: outlet temperature at tube-side
$T_{out,s}$: outlet temperature at shell-side
ΔT_m: log mean temperature difference, defined in Eq.(2)
u: average flow velocity at tube-side
U: overall heat transfer coefficient, defined in Eq.(1)
η : viscosity
ρ : density
τ_w : wall shear stress

REFERENCES

1. Altman, M., Yeh, H., and Lorsch, H.G., "Conservation and Better Utilization of Electric Power by Means of Thermal Energy Storage and Solar Heating," NTIS PB 210359 (1973).

2. Lane, G., ed., "Solar Heat Storage: Latent Heat Materials." vol II, CRC Press (1986).

3. DuPont Company, "Instruction Manual, 910 Differential Scanning Calorimter (DSC) System, (1980).

4. Handbook of Chemistry and Physics. The Chemical Rubber Co. (1986).

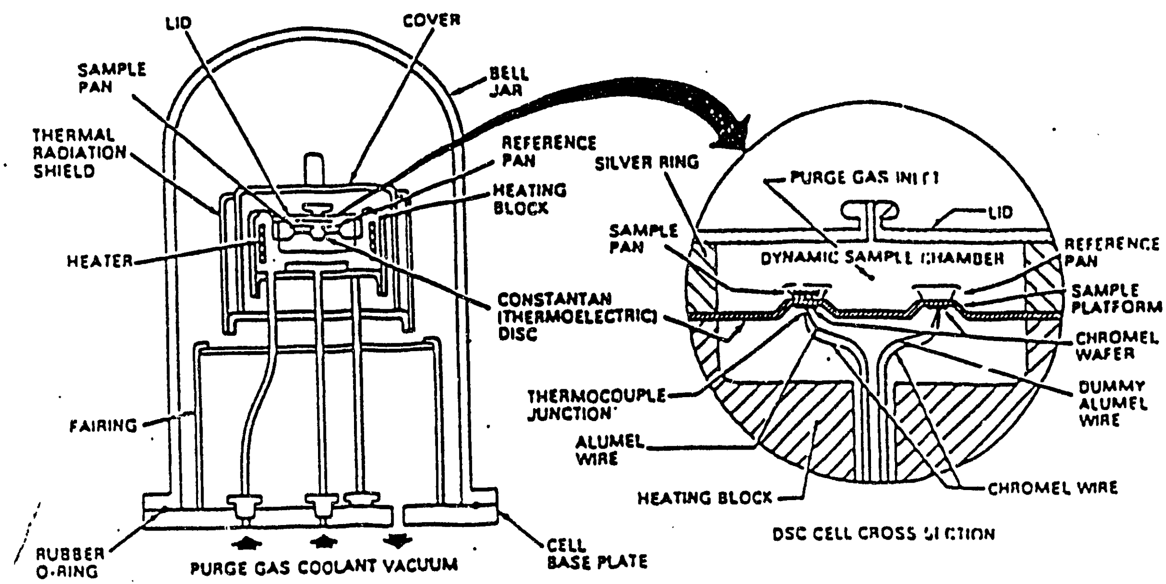

Fig. 1 Schematic diagram of differential thermal calorimeter

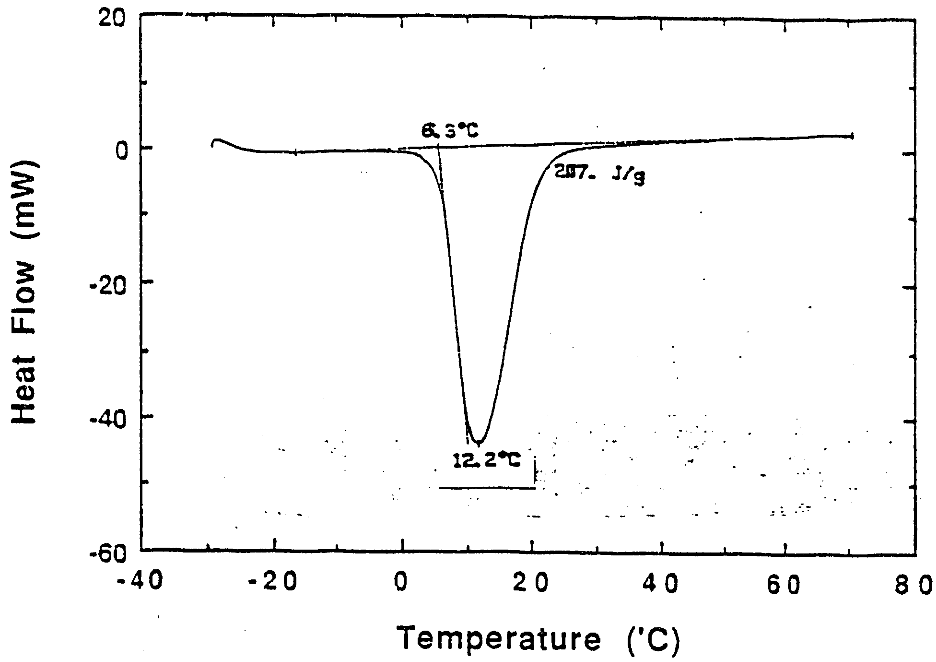

Fig. 2 Differential Thermal Calorimeter output plot for tetradecane

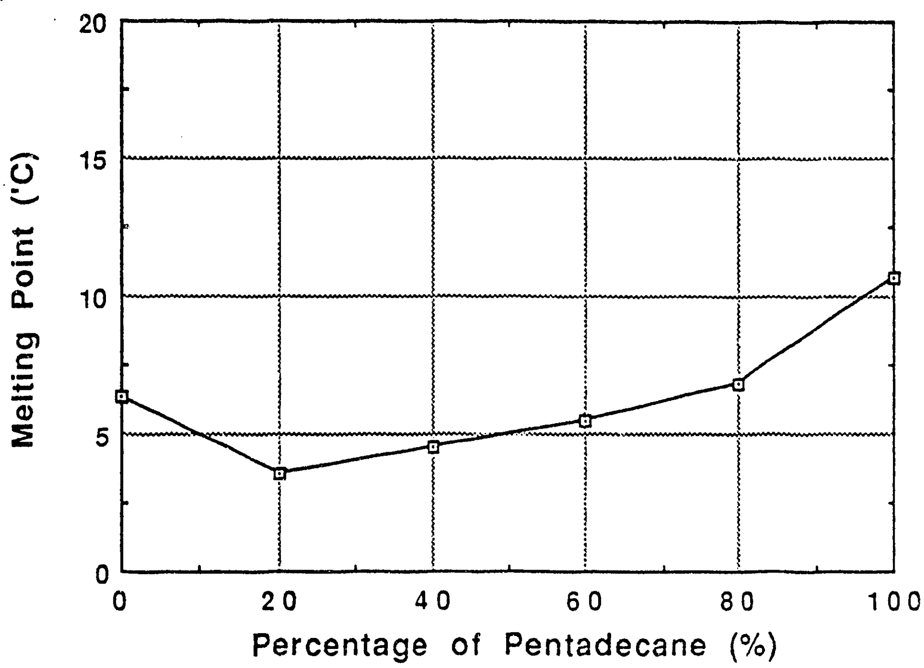

Fig. 3 Phase transition temperatures of tetradecane/pentadecane mixtures

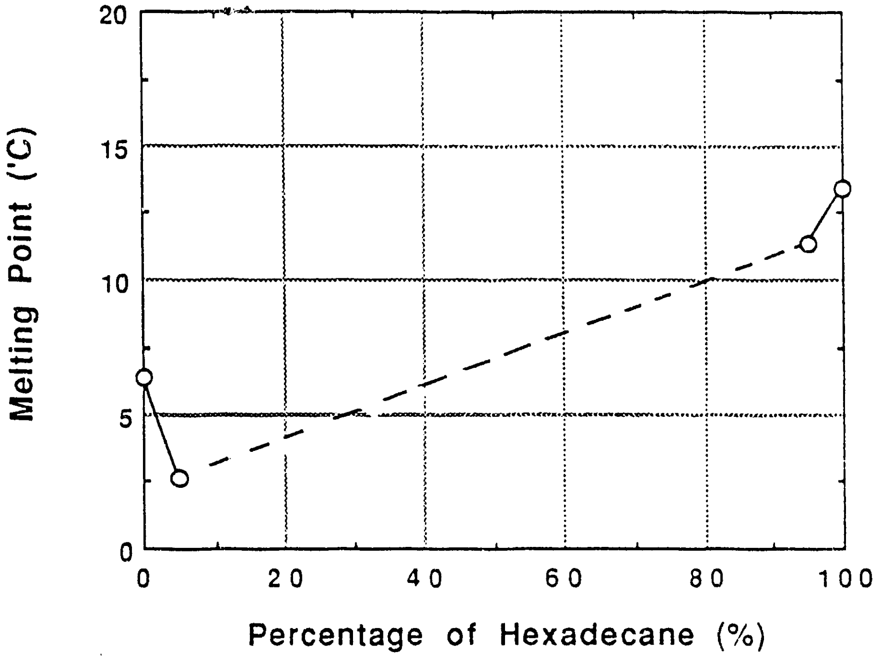

Fig. 4 Phase transition temperatures of tetradecane/hexadecane mixtures

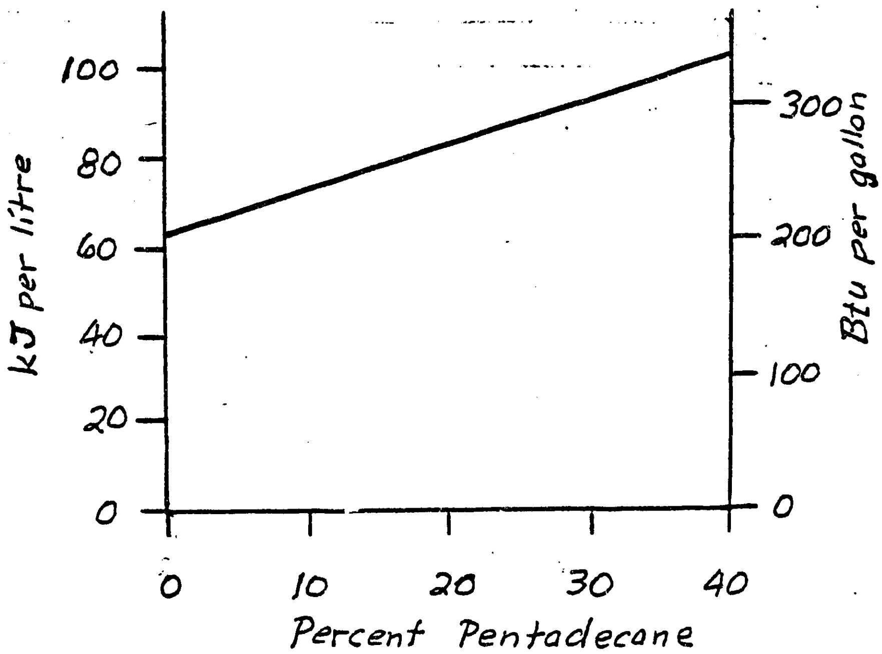

Fig. 5 Volumetric heat capacity of water/pentadecane slurries (ΔT=15°C, 27 F)

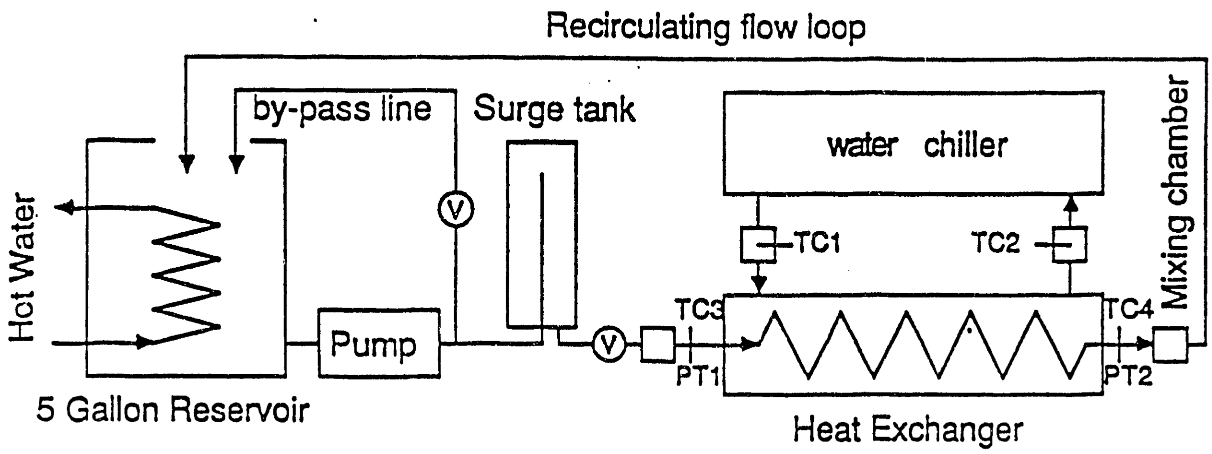

Fig. 6 Schematic diagram of a recirculating flow loop

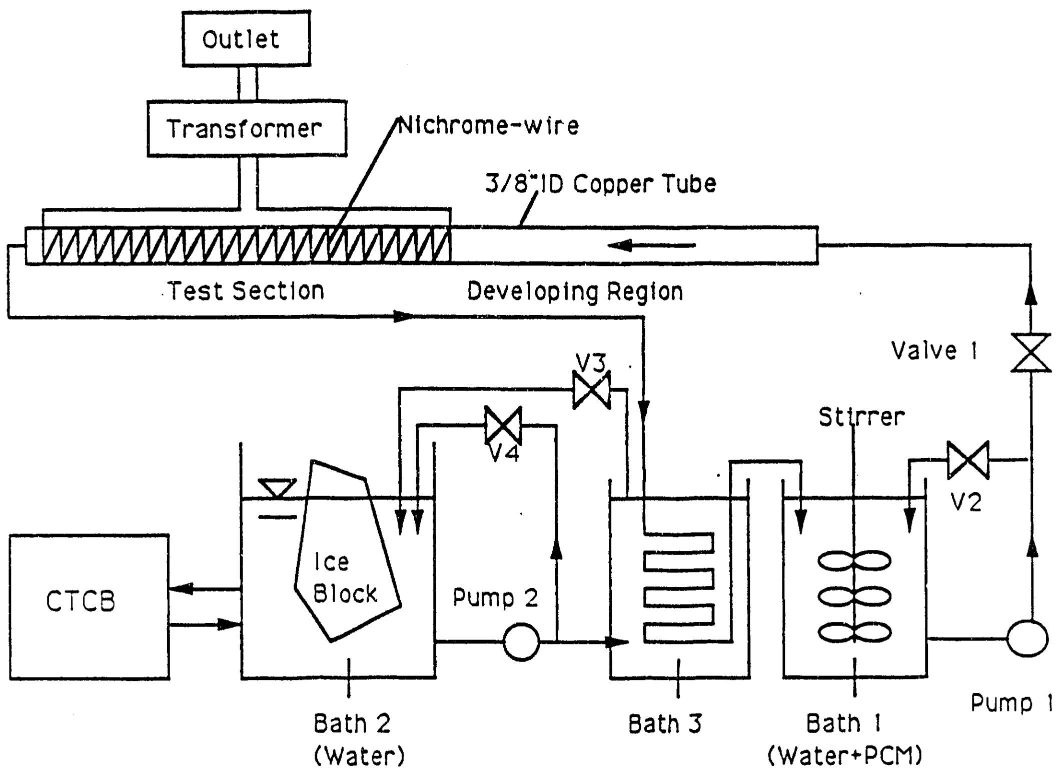

Fig. 7 Schematic diagram of the improved flow loop

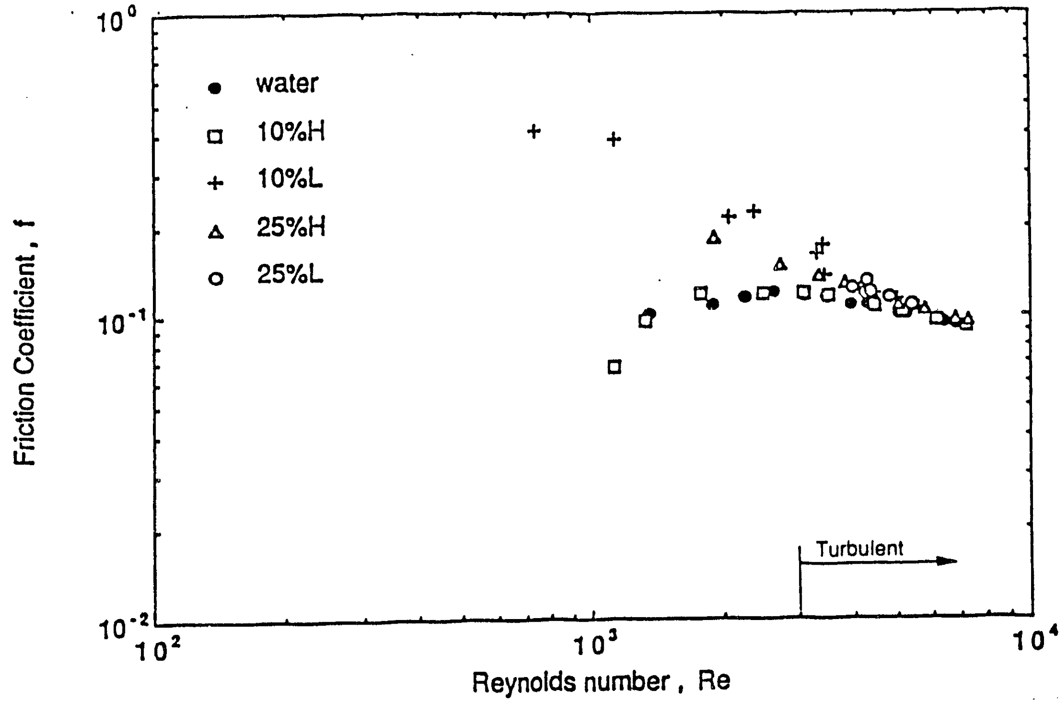

Fig. 8 Fanning friction coefficient vs. Reynolds number. 10% and 25% represent the amounts of pentadecane mixed in distilled water.

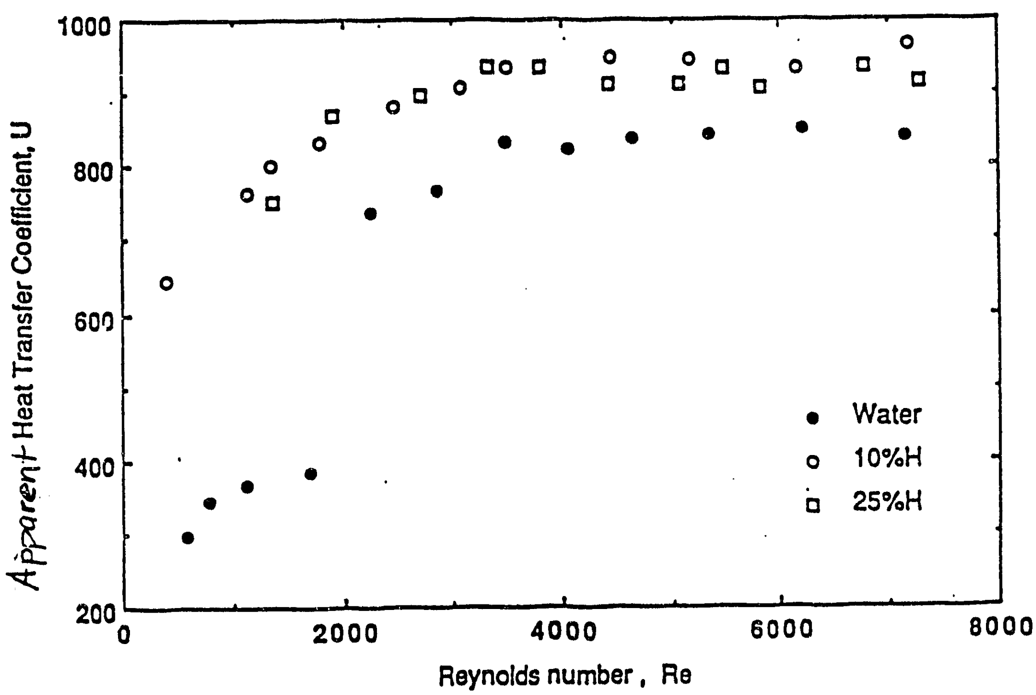

Fig. 9 Apparent heat transfer coefficient U vs. Reynolds number for the high inlet temperature case.

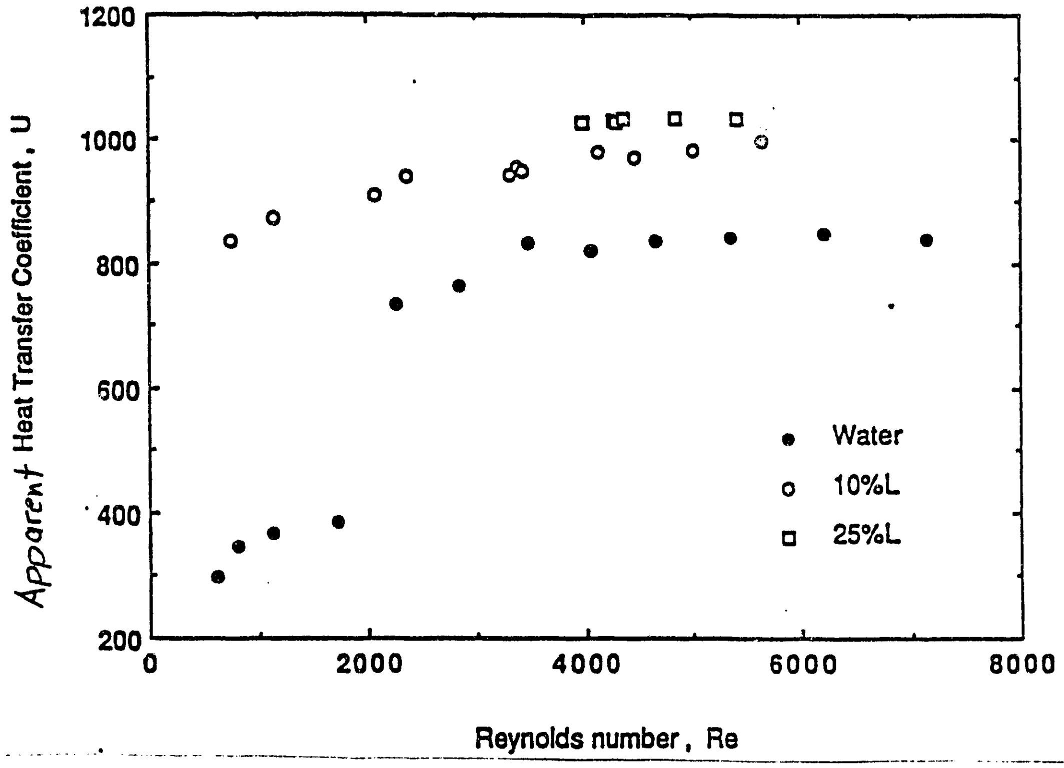

Fig. 10 Apparent heat transfer coefficient U vs. Reynolds number for the low inlet temperature case

FEDERAL ASSISTANCE PROGRAM
PROJECT DE-FG01-90CE2660
STATUS REPORT AS OF DECEMBER 31, 1990

Contractor: Drexel University
Philadelphia, PA 19104

Project Title: Development of Advanced Low-Temperature Heat Transfer Fluids for District Heating and Cooling

A number of phase change materials (PCM) were identified that, when mixed with water, could potentially increase the thermal capacity and the heat transfer rate over that of water alone. They are the tetradecane, pentadecane, and hexadecane waxes and their mixtures.

Tests were performed on tetradecane/pentadecane and tetradecane/hexadecane mixtures to determine melting points and latent heats of fusion. These tests showed tetradecane to be the most promising of the candidate materials, possible mixed with one or two of the others mentioned above. Its melting point is approximately 60C, and its heat of fusion is approximately 207 J/g, compared to 335 J/g for the water/ice transition.

The tests were performed on a differential scanning calorimeter (DSC) coupled with a differential thermal analyzer (DTA). Most of the tests were carried out on laboratory grade (99+% pure) material, and they will be repeated on commercial grade material. It is, of course, the commercial grade material that would be used in a district cooling application.

A bench-type flow loop was built consisting of a storage tank, a test cooling section, and a heating section, with an associated pumping system. Successful tests proved the basic feasibility of the concept. The PCM component of the PCM/water mixture could be repeatedly frozen and thawed without interfering with the fluid transport.

A larger flow loop was then built in order to measure thermal and hydraulic properties of the PCM/water mixture. For 10% and 25% PCM/water mixtures, the increase in the apparent heat transfer coefficient was measured to be 18 and 30 percent, respectively. No pumping penalty due to an increase in pressure required to move the PCM/water mixture was encountered in the turbulent region for additions of up to 25 percent of PCM to the water.

The larger flow loop was rebuilt and improved in order to permit more accurate observation and data-taking in addition to longer run times. From the operation of this loop it became apparent that the design of the cooling reservoir is highly important. It appears that freezing of the PCM should preferably take place inside, not outside, the cooling coil. Under certain conditions, the piping became clogged with frozen PCM particles. This difficulty was overcome by the addition of an extremely small amount of detergent, which resulted in a well/mixed PCM/water slurry that did not clog.

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