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

Final Report

Young I. Cho and Harold G. Lorsch
Drexel University
Philadelphia, Pennsylvania 19104

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ABSTRACT

The feasibility of adding phase change materials (PCMs) and surfactants to the heat transfer fluids in district cooling systems was investigated. It increases the thermal capacity of the heat transfer fluid and therefore decreases the volume that needs to be pumped. It also increases the heat transfer rate, resulting in smaller heat exchangers. The thermal behavior of two potential PCMs, hexadecane and tetradecane paraffin wax, was experimentally evaluated. 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.

While test results for laboratory grade materials showed good agreement with data in the literature, both melting point and heat of fusion for commercial grade hexadecane were found to be considerably lower than literature values. PCM/water mixtures were tested in a laboratory-scale test loop to determine heat transfer and flow resistance properties. For 10% and 25% PCM/water slurries, the heat transfer enhancement was found to be approximately 18 and 30 percent above the value for water, respectively. Within the turbulent region, there is only a minor pumping penalty from the addition of up to 25% PCM to the water.

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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INTRODUCTION

As a part of the development of advanced low-temperature heat transfer fluids to be used in district cooling systems, the use of phase change material (PCM) particles suspended in water was investigated. 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. It is proposed to add to the carrier fluid a quantity of a PCM whose phase change (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. The use of viscosity-reducing surfactants was also investigated; this would further reduce the required pumping power.

In order to optimize the saving in pumping energy that can be obtained from using a water/phase change material (PCM) mixture as the primary heat transfer medium in a district cooling system, the phase change 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

Figure 1. Molecular structure of paraffin waxes

Table 1. Thermophysical Properties of Selected Paraffin Waxes
(from Ref. 2)

Material	No of C atoms	Melting Point		Heat of Fusion		
		°C	°F	J/g	cal/g	Btu/lb
Tetradecane	14	5.8	42.4	227	54.3	97.4
Pentadecane	15	9.9	49.8	206	49.2	88.4
Hexadecane	16	18.1	64.6	236	56.3	101.3

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 CH₂ units with a CH₃ unit at each end (Figure 1). The exact phase transition temperature depends on the length of the chain, increasing with increasing chain length. These substances are hydrophobic, and they have similar molecular structures. Their significant thermophysical properties are shown in Table 1.

The desirable phase change temperature for a district cooling system probably lies somewhere between the phase change temperatures of tetradecane and pentadecane. Thus, a mixture of these two waxes may constitute a technically suitable PCM. On account of the high cost of pentadecane, mixtures of hexadecane with tetradecane may constitute a more economical mixture. This is why the bulk of the work was concentrated on evaluating mixtures of tetradecane and hexadecane as the PCM to be used in the advanced heat transfer fluids.

PRELIMINARY DETERMINATION OF THE MELTING POINT OF HEXADECANE

A simple method was devised to determine the approximate melting point of commercial grade hexadecane. The apparatus is shown schematically in Fig. 2. A Constant Temperature Circulating Bath (CTCB) was used to control the temperature of the water in the beaker. The temperature of the water in the CTCB was measured by a thermocouple and a mercury thermometer whose maximum temperature was 120°F. For additional details, see Appendix D. A small amount of hexadecane was floated on the surface of the water in the beaker. It spread uniformly across the water surface and had a shiny appearance when the water temperature was above the melting point of the wax. As the temperature was decreased by circulation from the CTCB, the shape of the surface of the hexadecane was observed. At a certain temperature, cracks formed on the surface of the hexadecane, indicating its freezing point. Since these paraffin waxes do not supercool, melting and freezing points are identical.

Figure 2. Test apparatus for measuring melting point of hexadecane

The melting point of this commercial grade hexadecane was observed to be 15.3-15.5°C by the thermocouple and 60.2-60.4°F (15.7-15.8°C) by the mercury thermometer. These values are between the values for laboratory grade and commercial grade material obtained by the more accurate DSC method (see Table 2 below). The preliminary method was not deemed to be sufficiently accurate for our purposes, and it was not used thereafter. Instead, melting points and latent heats of fusion of potential phase change materials were experimentally determined by Differential Thermal Analysis (DTA).

Table 1. Melting temperatures and fusion energies of selected *n*-alkanes

Alkane	Melting Temperature (°C)			Heat of Fusion (J/g)		
	Ref.2	Measured lab.	Measured comm.	Ref.2	Measured lab.	Measured comm.
Tetradecane	5.8	5.4	1.66	227	234	206
Pentadecane	9.9			206		
Hexadecane	18.1	17.2	12.1	236	224	185

* lab. : laboratory grade

* comm.: commercial grade

Figure 3. Schematic diagram of differential thermal calorimeter (from Ref.3)

EXPERIMENTAL DETERMINATION OF THERMOPHYSICAL DATA FOR PARAFFIN WAXES

Differential Thermal Analysis

The instruments used in this test method are a du Pont 910 Differential Scanning Calorimeter (DSC) (Fig. 3) connected to a du Pont 1090 Differential Thermal Analysis (DTA) System. The DSC consists of a thermally guarded cell in which identical samples of two different materials are simultaneously heated or cooled. One of the materials is the one to be tested, the other one is a reference material whose properties are accurately known. The primary heat transfer element, a constantan disc, is enclosed by a silver heating block capped with a silver venting lid. The two samples are placed on raised platforms of the constantan disc ensuring equal heat transfer to both of them. A chromel wafer is attached under each sample platform to which an alumel/chromel thermocouple is attached. A solid base and a bell jar lid that can be attached to it enclose the test cell. A purge gas entering the heating block through the cell base is preheated to the block temperature by circulation and, in turn, brings the entire assembly to that temperature before leaving through a vent hole in the lid.

Figure 4. Typical DSC output diagram

The output of the DSC system is a curve showing heat flow rate vs. temperature (Fig. 4). Starting with the frozen substance, the sample is heated at a constant rate represented by the straight line in the left part of the diagram whose slope is the specific heat of the frozen sample. The first deviation from that line indicates a phase change. The phase change temperature is obtained by drawing a tangent to the curve beyond the phase change and intersecting it with the original linear portion of the curve, as illustrated in the figure. The heat of fusion is obtained by integrating the area between the actual curve and the original straight line. The DTA is programmed to integrate that area and print the result directly on the chart.

In order to determine the accuracy of the DSC, nine samples with different amounts of distilled water were prepared and tested. Although the samples were of different sizes, the measured values of melting temperature were consistent; their standard deviation was 0.54°C . The standard deviation of the measured values of fusion energy was 15.3 J/g , which is 4.6% of the mean value.

Figure 5. DSC output diagram for pure laboratory grade hexadecane

Figure 6. DSC output diagram for a 60% tetradecane, 40% hexadecane laboratory grade mixture

These accuracies were deemed to be sufficient for the purposes of this project. Additional details are given in the section on *Description and Calibration of Test Apparatus* of Appendix B.

Test Results

Laboratory grade (99% pure) hexadecane and tetradecane were used to prepare twenty samples with various fractions of these waxes. A pure hexadecane sample exhibits only a single phase change (Fig. 5). When a small amount of tetradecane is added, the onset temperature of phase change, i.e., the melting point, is slightly lowered. When larger amounts are added, the shape of the curves changes (Fig. 6) indicating that molecular rearrangement occurs at more than one temperature. This phenomenon of "phase transition" increases as the fraction of tetradecane increases, resulting in a lowering of the first (primary) melting point and its heat of fusion. Additional details are given in Appendix B.

When varying amounts of hexadecane are added to tetradecane, the effects are similar. It can thus be concluded that, in the range of 10% to 90% of either substance, multiple phase changes take place. In the range of 40% to 80%, a tertiary phase change occurs, but at a considerably lower temperature. In the ranges of 0 to 10% and 90 to 100%, the primary phase change is dominant, and the mixture can be treated as a homogeneous solution. It is these latter ranges that are of interest to the district cooling application.

The phase change temperatures for all tested samples are plotted in Fig. 7. The horizontal coordinate indicates the concentration of tetradecane by weight. Thus, 0% tetradecane (left margin) indicates pure hexadecane, and the right margin indicates pure tetradecane. Table 2 indicates that the measured values for the pure substances show good agreement with published data (Ref. 2).

The fusion energy of the phase change for the 0 to 10% mixtures is somewhat reduced below that of either of the pure substances, as is illustrated in Figure 8. Here, too, the measured data agree well with Ref. 2 data shown in Table 2.

Melting Point Depression

In the ranges of 0 to 10% and 90% to 100%, the melting temperature depression is predictable according to the melting point depression theory (Ref. 4). These ranges are important because they cover a fairly wide spread of melting temperatures: from 10°C to 18°C and from 0°C to 5°C, respectively. The melting or freezing points of mixtures are always below those of their constituent pure substances. They are material properties, i.e., unique values for each material. The melting temperature depression constants, K_f , were determined by a regression analysis using all the tests data obtained. The results are

$$K_f = -8.74 \text{ }^{\circ}\text{C/mole} \quad \text{for hexadecane and}$$

$$K_f = -15.33^{\circ}\text{C/mole} \quad \text{for tetradecane.}$$

Details of this analysis are presented in the *Melting Point Depression* section of Appendix B.

Commercial Grade Material

If the proposed improvement in heat transfer fluids is to be viable for district cooling applications, commercial grade PCMs must be used instead of laboratory grade chemicals because of the high cost of laboratory grade materials. Therefore, exploratory tests were also performed on commercial grade PCMs. Test results for melting temperatures and heats of fusion for various mixtures of a commercial grade hexadecane and a laboratory grade tetradecane are shown in Figs. 9

Figure 7. Measured melting temperatures for mixtures of laboratory grade hexadecane and tetradecane

Figure 8. Measured heats of fusion for mixtures of laboratory grade hexadecane and tetradecane

Figure 9. Measured melting temperatures for mixtures of laboratory grade tetradecane with laboratory grade and commercial grade hexadecane

Figure 10. Measured heats of fusion for mixtures of laboratory grade tetradecane with laboratory grade and commercial grade hexadecane

Figure 11. Schematic diagram of a recirculating flow loop

and 10 together with tests on laboratory grades of both materials. The difference in melting in heat of temperatures between laboratory grade and commercial grade materials is almost 5°C. The reduction fusion of the commercial grade materials from laboratory grade materials is 10% for tetradecane and more than 20% for hexadecane.

HEAT TRANSFER AND PRESSURE TESTS

Experimental Setup

In order to determine whether slurries containing high fractions 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 11 shows a schematic diagram of the test flow loop used for that purpose. 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 was cooled below the freezing temperature of the wax in the main test section, and it was reheated above that temperature in the 5-gallon reservoir by using building service hot water. The test section consisted of a coiled glass tube, while the remainder of the loop was made of tygon tubing. The behavior of the test slurry could therefore be observed through the entire test loop. To minimize the effect of pumping-head fluctuation on pressure drop measurements, a surge tank was installed between the pump and the test section.

The temperature of the well-mixed test fluid was measured by four thermocouples (TC1 through TC4), and pressures were measured at two pressure taps (PT1 and PT2) located at the inlet and outlet of the coiled tube as shown in Fig. 11. The test fluid inside the reservoir was stirred by a mixer. Flow rates were controlled by two valves V, one located on a by-pass line and the other on the recirculating flow loop. The reservoir, the test section, and all tygon tubing were insulated with a one-inch-thick fiberglass blanket to minimize parasitic heat losses.

Test runs with pure water and 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 Figs. 12 through 14), 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 to 5°C, respectively. Additional details are given in the *Bench-Type Flow Loop* section of Appendix C.

Figure 12. Fanning friction coefficient vs. Reynolds number. 10% and 25% represent the amounts of pentadecane mixed in distilled water

Figure 13. Apparent heat transfer coefficient U vs. Reynolds number for the high inlet temperature case

Figure 14. Apparent heat transfer coefficient U vs. Reynolds number for the low inlet temperature case

Analytical Evaluation

The overall heat transfer coefficient U was obtained from

$$U = Q / (A \Delta T_m) \quad (1)$$

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

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

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

$$Q = \dot{m} C_p \Delta T \quad (3)$$

where $\dot{m}$ is the mass flow rate of the constant-temperature bath (measured and kept constant by means of a circulator pump not shown), C_p is the specific heat of water, and ΔT is the temperature difference between TC2 and TC1.

The use of the log mean temperature difference, ΔT_m , in a situation 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 it was felt no useful purpose would be accomplished by defining a different parameter. One should be 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 project 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), it is believed 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 exchanger with different heat transfer fluids.

The Fanning friction coefficient f was calculated from

$$f = \tau_w / (0.5 \rho u^2) \quad (4)$$

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 $\dot{m}$:

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

Results

Figure 12 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 13 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 between 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. This may, however, lead to clogging of the tube after an extended run time (see the section on *Tube Clogging* below).

Figure 14 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.

A quantitative indication of the increase in volumetric heat capacity of a district cooling heat transfer fluid through the addition of a PCM is given in Fig. 15. Assuming a difference of 15°C between send-out and return temperatures and using pentadecane test values for the PCM, the

Figure 15. Volumetric heat capacity of water/pentadecane slurries ($\Delta T=15^{\circ}\text{C}, 27^{\circ}\text{F}$)

increase in volumetric heat capacity is approximately 1.6 percent for each percent of PCM added. If the temperature difference between send-out and return is smaller, the capacity increase becomes even larger.

ENLARGED FLOW LOOP

In preparation for full-scale testing, a somewhat larger flow loop than the bench-size loop of Fig. 10 was constructed. It will incorporate a refrigeration-type water chiller with a capacity between 5 and 10 kW (1 to 3 tons). However, procurement of that chiller had to be postponed, since no funds for its acquisition had been provided within the contract scope. These funds have now been incorporated in a follow-on contract. The current configuration of that loop without the chiller is depicted and described in detail in the section on *Enlarged Flow Loop* of Appendix C.

TUBE CLOGGING

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 of the Fig. 10 test loop, 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-diameter tygon tubing and the glass coil never became clogged, and full flow was maintained throughout the test.

On the other hand, when hexadecane was introduced in a larger flow loop consisting of copper piping (Fig. 7, Appendix C), the PCM solidified inside the copper tubing. This had not occurred, when the PCM was mixed with water by hand outside the flow loop piping. The addition of a surfactant ("Tide" detergent) remedied the situation in the flow loop, and no clogging occurred. However, after several hours running time, solid PCM particles could no longer be observed even at temperatures well below the freezing point of hexadecane. At this time, no good explanation is apparent for this behavior. It will be investigated in greater detail during the follow-on contract.

During these preliminary experiments it was also observed that pentadecane does not readily dissolve in water. Most of it floats 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.

CONCLUSIONS

- * Experimental results for melting temperature and fusion energy for laboratory grade hexadecane and tetradecane show good agreement with handbook data. Mixtures of these two substances in the ranges of 10% to 90% by weight exhibit multiple phase changes.

- * In the ranges from 0 to 10% and from 90% to 100%, a single phase change is dominant. It is these ranges that are of interest for district cooling. The fusion energy of the mixtures in these ranges is slightly lower than that of the pure substances.

- * Melting temperature and heat of fusion of commercial grade hexadecane were found to be considerably below values given in the literature and measured for laboratory grade materials.

- * 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 $\Delta t=15^{\circ}\text{C}$ 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 did not clog either tygon or glass tubing in the turbulent flow region for mixtures containing as much as 25% PCM.

- * Hexadecane clogged copper tubing. The clogging did not occur when a small amount of surfactant was added to the water/PCM mixture.

- * Tests will be performed during the follow-on contract to determine whether the clogging depends on the PCM or the tube wall material.

- * 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 installing a chiller in an enlarged flow loop. This will permit testing at increased heat transfer rates. Different ratios of tetradecane/hexadecane combinations will be investigated 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. In addition to reducing friction, surfactants are expected to improve the uniformity of PCM particle size and distribution throughout the PCM/water mixture. The clogging phenomenon observed during some of the tests will be further investigated.

PUBLICATIONS RESULTING FROM THIS PROJECT

Appendix A: "Use of Advanced Low-Temperature Heat Transfer Fluids for District Cooling Systems." Y.I. Cho, B.-C. Pak, and H.G. Lorsch. Presented at the International District Heating and Cooling Conference, Virginia Beach, VA, June 1989.

Appendix B: "Thermal Analysis of n-Alkane Phase Change Material Mixtures," Y.I. Cho, E. Choi, and H.G. Lorsch. To be presented at the International District Heating and Cooling Conference, Copenhagen, Denmark, April 1991.

Appendix C: "A Novel Concept for Heat Transfer Fluids Used in District Cooling Systems," Y.I. Cho, E. Choi, and H.G. Lorsch. To be presented at the Annual Meeting of the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE), Indianapolis, IN, June 1991. To be published in ASHRAE Transactions, vol. 97, part 2.

Appendix D: Description of Flow Loop and Data Acquisition System

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APPENDIX D

Description of Flow Loop and Data Acquisition System

A conceptual schematic of a flow loop for measuring heat transfer fluid properties is shown in Figure 1. Heat Q_H is added at one point and Q_L is withdrawn at another point. If a steady state is reached,

$$Q_H = Q_L$$

Figure 2 is a schematic diagram of the initial flow loop constructed. The inner diameter of the test section is 3/8", the temperature difference between the inlet and the outlet is 5°C, the length of the flow loop is 5 m, the flow velocity in the test section is 3.255 m/s, and the total pressure loss is 10.72 psi. In order to obtain a Reynolds number of 20,000, a heat input and a cooling output of approximately 5 kW was required. While a 5-kW heater was readily available, no equivalent cooling unit could be procured immediately. An ice block was therefore used in lieu of a cooling unit. A one-cubic-foot block of ice can be used to produce 5 kW of cooling for 30 minutes.

Referring to Fig. 2, pump 1 is the main pump, valve 1 is the main supply valve, and valve 2 is a bypass valve. Through the use of valves 1 and 2, the flow rate in the test section can be controlled while maintaining a constant flow in the main pump. The mixture of PCM and water is in bath 1 from where it is circulated through the test loop. In bath 1, there are a heat exchanger and a stirrer. The heat exchanger is designed to have provide 5 kW of cooling. It is constructed of four parallel tubes in order to reduce the pressure loss. Bath 2 contains water and some ice blocks, and its cold water is circulated through the heat exchanger of bath 1 by the pump 2. Valves 3 and 4 are used to control the flow rate of the cooling water. The CTCB (constant temperature circulating bath) is used to assist the ice blocks in chilling the water in bath 2, thus extending the potential run time of the experiment.

Figure 1. Simplified diagram of flow loop test

Figure 3. Schematic diagram of the heating test section

The heater is made of 0.65-mm-diameter nichrome wire wrapped around the 3/8-inch diameter test section tubing. A transformer with a maximum capacity is 140 volts and 40 amperes is used to control the current in the nichrome wire. Details of the heater are shown in Fig. 3. Three parallel nichrome wires with a total resistance of 3.0 ohms are used. Four 10-ampere fuses are connected in parallel for safety. The space between two adjacent nichrome wires is 1 cm. The nichrome wires are insulated from the copper tube by a thin layer of epoxy. Even though the layer is thin, the insulation is sufficient.

A block diagram of the data acquisition system is shown in Fig. 4. Two pressure collars are located at the inlet and the outlet of the test section to measure the static pressure of the flow. They are connected to a Validyne DP15TL pressure transducer, which is activated by a Validyne CD15 amplifier. The signal from the pressure transducer is shifted and amplified in that amplifier. Two thermocouple sensors are located at the inlet and outlet points of the copper tube. They are connected to the thermocouple terminal box which contains a thermistor for measuring the reference temperature. In order to calculate the heater power output, at least two quantities among current, voltage, and resistance must be measured. Since the resistance of the heater is known, only current or voltage need be measured. A hook gage type AC current transducer Fluke Y8101 is used to measure the heater current. The voltage is also measured as a check. The voltage is too high for the data acquisition interface, so a voltage attenuator was used.

All signals from the pressure transducer, the thermocouple, the current transducer, and the voltage attenuator are connected to the Omega WB-AAI-B16 data acquisition interface. The interface resolution is 16 bits, and the sampling frequency is above 225 Hz, which is ample for a steady state experiment. The interface is connected to an IBM-AT, so all the data are immediately fed to the computer. In order to measure the temperature of the nichrome wire, another thermocouple is installed on the nichrome wire, and its temperature is displayed by a portable temperature indicator. Due to the small temperature difference, the system is somewhat sensitive to noise. In order to decrease the noise in the thermocouple signal, a floating ground line was therefore installed. The thermocouple noise was checked by an oscilloscope that gave satisfactory readings.

Figure 4. Block diagram of the data acquisition system