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SOLAR HEATING AND COOLING SYSTEMS

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RATING AND CERTIFICATION OF DOMESTIC WATER HEATING SYSTEMS

Thermosyphon System:

Testing of the Solahart 180 JK thermosyphon system using the CSU Solar Simulator has been completed. The system was also modeled using TRNSYS 13.1 and a version of the TRNSYS thermosyphon subsystem as modified by Graham Morrison of the University of New South Wales, Australia (hereafter TRNSYS GM) to see how well the models predicted the observed behavior. Many of the results were presented in the last project status report (June 1994). Several of the final conclusions are presented below. Please refer to the Master's thesis, <u>Short Term Performance Comparison Between A Solar Thermosyphon Water Heater and Two Numerical Simulations</u>, by Carl Bickford, for more information.

Experimental Results:

- The storage tank has good stratification during solar-only operation (up to 10°C).
- Use of the auxiliary heater brings the upper tank to the set point, but leaves the lower half of the tank stratified such that solar heat may still added to the lower half of the tank.
- Tank mixing during load draws is minimal (as defined by the Collector and System Testing Group 1989 Procedure).

Modeling Results:

- TRNSYS GM predicts closed loop collector flow better than TRNSYS 13.1 and is about 8% lower than the observed value.
- TRNSYS GM predicts higher collector temperature rises than observed, but when combined with the lower predicted flow rates yields collector output energies that are very close to the experimental values.
- TRNSYS GM predicts much higher storage tank stratification than observed during solar-only operation resulting in exaggerated initial draw temperatures and energies. The discrepancy is so large that it suggests that an influential destratification mechanism is being ignored.
- TRNSYS GM accurately models storage tank stratification with auxiliary heat use.
- Average tank temperature prediction is consistently high, but can be matched by increasing tank and pipe heat losses.
- Both models predict total system energy delivery within 17%, but TRNSYS GM parallels experimental flow and temperature data much better. The error can be reduced (to essentially zero) by adjusting the loss coefficient for the tank and piping.

Recommendations:

- The basic TRNSYS 13.1 thermosyphon model should not be used to model thermosyphon systems which incorporate heat exchangers.
- Since the auxiliary heater heats the upper half of the tank, the total solar gain of the system is limited when it is on. This suggests, that for many installations, such as in climates that have cold winters, the system may perform better as a solar-only preheater, with the auxiliary element used only for freeze protection. This would allow use of the entire tank for solar storage.
- A thorough experimental determination of storage tank and piping heat loss coefficients should be performed to improve modeling results.

Current Work:

The TRNSYS GM decks are being modified slightly to give yearly performance results for the suggested use as a solar-only system. These will be compared to the yearly results with full auxiliary.

A new self-pumping system is being sent to SEAL and will be tested using the solar simulator.

UNIQUE SOLAR SYSTEM COMPONENTS

INTEGRATED TANK/HEAT EXCHANGER MODELING/EXPERIMENTS

The following experiments have been completed on the 80 gallon Rheem wrap-around heat exchanger tank:

A 24 hour tank heat loss test was conducted to determine the effective overall heat loss coefficient of this tank. The effective heat loss coefficient is determined from the experimental data and the results are compared to a TRNSYS simulation.

Seven constant temperature heat input tests were conducted to characterize the heat transfer performance of the tank and heat exchanger. The experimental heat transfer performance is determined and compared to the results of TRNSYS simulations using the wrap-around heat exchanger tank model previously developed at CSU (Miller et al., 1993). Model normalization has not yet been considered.

Three auxiliary heater thermostat were tested to determine the upper set point and control deadbands of typical thermostats insitu. These experiments were conducted at the request of SRCC and FSEC to help establish correct parameter inputs for TRNSYS modeling. Results of these test are summarized.

Experimental Facility and System Instrumentation

Figure 1 is a schematic of the Rheem wrap-around heat exchanger solar storage tank. The tank is a typical 80 gallon electric hot water heater tank with the bottom electric resistance heating element replaced by a 120 feet long coil of 5/8 inch copper tubing wrapped around outside of the lower half of the tank. The top third of the tank, heated by the remaining electric element, functions as an auxiliary tank. The bottom two-thirds is a solar pre-heat tank heated by hot antifreeze circulating from the solar collector through the external copper coil. This design provides a compact single tank system with an integral double-walled heat exchanger. No hot collector fluid flows through the tank to directly disturb the tank stratification.

Figure 2 shows the installation of the tank on the test stand. On the collector side of the tank/heat exchanger, heat gain is simulated with a computer controlled circulation heater. Two cold water supply tanks allow the solar storage tank to be preconditioned at the start of a run and allow the temperature of the make-up water during domestic hot water draws to be controlled.

Eight special limit, T-type thermocouples (T8 - T15) are located vertically in the tank as shown in Figures 1 and 2. They divide the tank into eight equal segments. The thermocouples are inserted through the 3/4" NPT hot water outlet fitting in individual 1/8 inch brass sheaths. The open area of the outlet is reduced by approximately 25%. The thermocouple beads are electrically insulated from the sheath and from the water with thermocouple epoxy. Temperatures at the inlet and outlet of the circulation heater (T22 and T23), the heat exchanger (T16 and T17) and the solar storage tank (T6 and T7) are measured with prefabricated, 2252 ohm thermistors in 1/8" stainless steel sheaths. All remaining temperatures (T0 - T5) are measured with T-type thermocouples in brass sheaths. The tank environment temperature is measured by a radiation shielded thermocouple at the mid height of the tank.

For all of the experiments reported here, water is the heat transfer fluid on the collector side of the heat exchanger. Turbine flow meters measure the collector loop mass flow rate and the draw mass flow rate. The electrical power drawn by the circulation heater and the solar storage tank auxiliary heater is measured with watt transducers.

Tank Heat Loss Test

A tank heat loss test was performed to determine the apparent overall heat loss coefficient of the Rheem wrap-around heat exchanger tank. The tank was heated to a uniform temperature of 47°C and allowed to cool over a 24 hour period. This was accomplished by enabling the electric resistance heaters in both of the cold water preparation tanks and the auxiliary heater in the Rheem tank. The cold water circulation pump thoroughly mixed the water in the three tanks. The target starting temperature was 50°C, however, the temperature controllers on the cold water preparation tanks were set to their lowest possible setting 110°F (43.33°C) and the maximum temperature reached was only 47°C. Once the main storage tank had reached a uniform 47°C, the three heaters and the circulation pump were disabled and the data acquisition system was started. The isolation valves on the tank were left in their normally open positions, therefore, some natural convection circulation in the entrance and exit piping was possible. The tank was allowed to cool over a 24 hour period. No precautions were taken to control the tank environment temperature.

The water temperatures were recorded in the cold water supply system (T0 - T6), in the main storage tank (T8 - T15), and in the outlet of the main storage tank (T7). The tank environment temperature (Tamb) was also recorded. The data channels were sampled at a rate of one sample every 5 seconds. The data was averaged and recorded every 30 seconds.

Figure 3 shows traces of the tank temperatures (T8 (top) to T15 (bottom)) over the 24 hours of the test. All of the data records are plotted. The average tank temperature is computed and plotted with a bolder line. Also shown is the tank environment temperature. The initial average temperature of the tank is 47.01°C, and the final average tank temperature is 41.53°C. Note, the tank environment temperature varied considerably during the test. The maximum environment temperature was 25.12°C; the minimum was 23.22°C. The mean (over the length of the test) tank environment temperature was 24.10°C with a standard deviation of 0.54°C. Ideally, the tank environment temperature should be held constant within ±0.1°C. See analysis below.

Determination of the Tank Heat Loss Coefficient, U_{solar}

An energy balance on the main storage tank requires that the rate of change of energy in the tank must be equal to the rate of loss of energy from the tank.

$$(MC_p)_{tank} \frac{d\overline{T}_{tank}}{dt} = - (UA)_{tank} (\overline{T}_{tank} - T_{env}) \tag{1}$$

where, the mean temperature of the tank is

$$\overline{T}_{tank} = \sum_{i=8}^{15} \frac{T_i}{8} \tag{2}$$

A the lumped capacitance model of the tank where the capacitance, $(MC_p)_{tank}$, is essentially constant (not a function of temperature), and, the energy of the tank is characterized by the mean temperature of the tank is assumed. The overall loss coefficient, U_{tank}, of the tank is assumed

to be uniform over the surface of the tank. Note, the tank capacitance, $(MC_p)_{tank}$, is the apparent capacitance of the tank and its contents.

For constant tank environment temperature, T_{env}, Equation (1) can be rewritten as

$$(MC_p)_{tank} \frac{d\Delta T}{dt} = - (UA)_{tank} \Delta T \tag{3}$$

where $\Delta T = (\overline{T}_{tank} - T_{env})$, and $d\Delta T/dt = d\overline{T}_{tank}/dt$.

Rearrangement of Eqn. 3 gives,

$$\frac{d\Delta T}{\Delta T} = - \left(\frac{UA}{MC_p} \right)_{tank} dt$$

which integrates to,

$$\ln \Delta T = - \left(\frac{UA}{MC_p} \right)_{tank} t + \ln C \tag{4}$$

where $\ln C$ is a constant of integration. At t=0, $\Delta T = \Delta T_0 = (\overline{T}_{tank}(t = 0) - T_{env})$, therefore,

$$\ln \left(\frac{\Delta T}{\Delta T_0} \right) = - \left(\frac{UA}{MC_p} \right)_{tank} t \tag{5}$$

and,

$$(UA)_{tank} = - \frac{(MC_p)_{tank}}{t} \ln \left(\frac{\Delta T}{\Delta T_0} \right) \tag{6}$$

The tank heat loss coefficient (UA) can be determined from Eqn. 5, or 6. Using Eqn. 5, the tank heat loss coefficient (UA) is obtained by plotting the quantity $\ln(\Delta T/\Delta T_0)$ against time, t. The slope of a line fitted through the data is $-(UA/MC_p)$. The capacitance of the tank (and contents) is estimated as the capacitance of the volume of water in the tank (neglecting the capacitance of the tank itself). The European community dynamic test procedure (Aranovitch et al., 1989) recommends using Eqn. 6 to determine the tank heat loss coefficient. ΔT_0 is the temperature difference between the initial average temperature of the tank (Eqn. 2) and the environment temperature averaged over the duration (t) of the run. ΔT is this temperature difference at the end of the run.

The capacitance of the volume of water in the tank can be estimated as

$$MC_p = \rho V C_p = 1.0 \left(\frac{kg}{l} \right) \left(80(gal) \cdot 3.7854 \left(\frac{l}{gal} \right) \right) 4187 \left(\frac{J}{kg\ K} \right) = 1.26796 \times 10^6 \left(\frac{J}{K} \right) \tag{7}$$

Figure 4 shows a plot of $\ln(\Delta T/\Delta T_0)$ vs. t (as in Eqn. 5). The tank environment temperature is assumed constant in the development of Eqns. 5 and 6. As shown in Figure 3, the environment temperature varies by as much as 1°C about a mean of 24.1°C. In the first trace in Figure 4, the measured value of T_{env} is used to calculate ΔT. In the second trace, the run average value of T_{env} (24.1°C) is used. A linear least squares fit to each of the traces is shown. The resulting values for UA_{tank} are, respectively,

$$UA_{tank} = -(-3.213 \times 10^{-6} s^{-1}) \cdot MC_p = 4.07 \ (W/K)$$

$$UA_{tank} = -(-3.220 \times 10^{-6} s^{-1}) \cdot MC_p = 4.08 \ (W/K) \tag{8}$$

Using Eqn 6,

$$UA_{tank} = -\frac{1.26796 \times 10^6 \ J/K}{86400 \ sec} \ln\left(\frac{(41.53-24.1) \ C}{(47.01-24.1) \ C}\right) = 4.01 \ (W/K) \tag{9}$$

TRNSYS Comparisons

The effective loss coefficient, U_{solar} (with units of kJ/hr-m^2-K), is a required parameter input for the TRNSYS Type 4 Stratified Storage Tank Model (and for the Type 67 Wrap-Around Heat Exchanger Tank Model). Other unit systems are possible, but are not commonly used. Note the kJ energy units and per hour time units. The model then calculates the overall effective conductance (for a single node tank) by multiplying the effective loss coefficient by the surface area of the tank, or

$$UA_{tank} = U_{solar}\left[\pi D_{tank} h_{tank} + 2\left(\frac{\pi D_{tank}^2}{4}\right)\right]$$

where D_{tank} is the diameter of the tank and h_{tank} is the height of the tank. D_{tank} is calculated from the parameter inputs, V_{tank}, the volume of the tank and h_{tank}.

$$D_{tank} = \sqrt{\frac{4V_{tank}}{\pi h_{tank}}}$$

For this tank,

$$V_{tank} = 80 gal \cdot \frac{3.78541}{1 gal} \cdot \frac{1 m^3}{1000 l} = 3.028 \times 10^{-1} m^3$$

$$h_{tank} = 58.625 in \cdot \frac{2.54 \times 10^{-2} m}{1 in} = 1.489 m$$

$$D_{tank} = \sqrt{\frac{4 \cdot 3.028 \times 10^{-1} m}{\pi \cdot 1.489 m}} = 0.509 m \quad (= 20.0 in)$$

and

$$A_{tank} = \pi \cdot 0.509m \cdot 1.489m + 2\left(\frac{\pi(0.509m)^2}{4}\right) = 2.788m$$

The nominal value loss coefficient that has been used for the 80 gallon Rheem wrap-around heat exchanger tank is calculated from the nominal R-value of 2 inches of polyurethane foam insulation surrounding the tank. The R-value is 16.7 (hr-ft^2-F/Btu).

$$R_{solar} = 16.7\frac{hr-ft^2-F}{Btu} \cdot \frac{1\frac{Btu}{hr-ft^2-F}}{5.68\frac{W}{m^2K}} \cdot \frac{1000J}{1kJ} \cdot \frac{1hr}{3600s} = 0.817\frac{hr-m^2-K}{kJ}$$

Following SRCC/SMUD convention, the calculated R-value has been divided by a factor of 2 for use in TRNSYS simulations. The loss coefficient specified to the Type 4 tank model is then,

$$U_{solar} = \frac{2}{R_{solar}} = \frac{2}{0.817\frac{hr-m^2-K}{kJ}} = 2.45\frac{kJ}{hr-m^2-K}$$

and the apparent overall conductance of the tank is

$$UA_{tank} = 2.45\frac{kJ}{hr-m^2-K} \cdot 2.788m^2 = 6.826\frac{kJ}{hr-K} \quad \left(\underline{=1.90\frac{W}{K}}\right)$$

This value of the tank UA is less than the half of the measured value of 4.0 W/K (see calculations above).

$$4.0\frac{W}{K} = 14.4\frac{kJ}{hr-K}$$

The value of R_{solar} witch matches the experiment results is

$$U_{solar} = \frac{UA_{tank}}{A_{tank}} = \frac{14.4\frac{kJ}{hr-K}}{2.788m^2} = 5.17\frac{kJ}{hr-m^2-K}$$

$$R_{solar} = \frac{2}{U_{solar}} = 0.387\frac{hr-m^2-K}{Btu} = 7.916\frac{hr-ft^2-F}{Btu}$$

Figure 5 shows the tank temperatures predicted by a TRNSYS simulation. Eight tank nodes were specified and the measured variation in the tank environment temperature was input to the simulation. U_{solar} was set equal to 5.17 kJ/hr-m^2-K to match the experiment. The TRNSYS time step was fixed at 5 minutes.

The simulation results show good agreement (as expected) with the experimental results of Figure 3 . The final average tank temperature predicted by TRNSYS, 41.55°C, agrees with the

measured value of 41.53°C. The experimental temperatures show a larger variation in temperature from the top of the tank to the bottom than do the predicted temperatures. Conduction between tank nodes is not modeled by this TRNSYS tank model, so there is no heat transfer path from the upper tank nodes to the cooler bottom node.

Table 1 summarizes the predicted final average tank temperature for several TRNSYS simulations. In runs 1 and 2, the original value of U_{solar} = 2.45 kJ/hr-m^2-K is used. The number of tank nodes is 1 and 8 respectively and the average value of T_{env} = 24.1°C is specified. The predicted final average tank temperature is nearly 3°C higher in each case. In runs 3 through 6, the experimental value of U_{solar} = 5.17 kJ/hr-m^2-K is used. The number of tank nodes is varied as shown. In runs 3 and 4, the average tank environment temperature is specified. In runs 5 and 6, the experimental variation in tank environment temperature is specified. Agreement is good regardless of number of tank nodes or specification of tank environment temperature.

Table 1. Results of TRNSYS simulations of heat lost test.

Run	U_{solar} kJ/hr-m^2-K	Number of Tank Nodes	T_{env} °C	$\overline{T}_{tank}$ °C
1	2.45	1	24.1	44.23
2	2.45	8	24.1	44.23
3	5.17	1	24.1	41.55
4	5.17	8	24.1	41.55
5	5.17	1	Experimental Variation	41.53
6	5.17	8	Experimental Variation	41.55

Note, the tank volume used to calculate the capacitance of the tank and its contents, $(MC_p)_{tank}$ in Eqn. 7 above has not been determined experimentally. In fact, the dimensions shown in Figure 1 suggest that the actual tank volume is 73 gallons. The volume used to calculate $(MC_p)_{tank}$ has a clear affect on the value of U_{solar} calculated in Eqns. 8 and 9. However, the when the same volume is specified in the TRNSYS simulations, the outcome of the comparisons is not affected. Therefore, the volume of the tank must be verified independent of this experiment and calculation, and, when determined, used to recalculate U_{solar}, from the results of this experiment. A more realistic value for the volume of the tank has not been determined at this time.

Constant Heat Input Tests

When the heat input on the collector side of the heat exchanger is held constant with no draw flow and no auxiliary heat addition, the energy balance on the solar portion of the tank becomes

$$MC_{p_r} \frac{dT}{dt} = UA_{hx}\Delta T_{lm} = Q_{input} = \text{cons}\tan t \tag{10}$$

For all cases of interest, tank losses are small compared to the heat gain across the heat exchanger. The tank temperature will increase linearly in time and for a fixed collector-side flow rate, the heat exchanger overall heat transfer coefficient and log mean temperature difference will

remain nearly constant over the length of the test. Increasing Q_{input} (from run to run) while holding the collector-side flow rate constant, increases the ΔT_{lm} and the tank-side free convection heat transfer coefficient independent of the coil-side. Holding Q_{input} constant (from run to run) while increasing the collector-side flow rate increases the coil-side forced convection heat transfer coefficient relatively independent of the tank-side heat transfer. The temperature difference driving the free convection flow is, however, dependent on the ratio of h_{coil}, the tube-side forced convection coefficient, and h_{tank}, the tank-side free convection coefficient. A constant heat input test is, therefore, an ideal experiment to test the accuracy of the heat exchanger portion of the tank model.

Figure 6 shows the history of the tank and heat exchanger temperatures for a typical constant heat input test (Test #2 below). Data is sampled at 10 second intervals. The average heat input for this test was 1872 watts. The collector loop flow rate was 3.78 lpm (1.0 gpm). At the beginning of the test, the auxiliary heater element is enabled to allow the top of the tank to reach normal operating condition. The temperatures at thermocouples T8 and T9, located above the heater element rise together. Note that the bulk water temperature above the heater does not initially reach the thermostat set point of 54 °C (130 °F). However, approximately three hours into the test, the auxiliary heater thermostat turns the heater back on for a short time and the bulk water temperature does reach the thermostat set point. This response is typical of many electric hot water heaters. The thermostat measures the wall temperature of the tank just above the heater element, not the temperature of the water. On initial heat-up from a cold start, the wall temperature above the base of the element rises more quickly than the bulk fluid temperature and the thermostat appears to shut off the heater early. As the wall temperature cools to the bulk fluid temperature, the heater turns back on. The operation of the auxiliary heater does not seem to influence the heat transfer across the heat exchanger.

The temperature at T10, just below and to the side of the heater element, is strongly influenced by the auxiliary heater. The temperature rises more slowly, however, and decays quickly when the heater shuts off due to conduction losses to the colder bottom of the tank.

At one hour into the test, the circulation pump and heater are enabled. As heat is added, the water temperature in the bottom half of the tank (T12 - T15) rises above the temperature of the water just above (T10 -T11). This temperature instability gives rise to free convection circulation which mixes the bottom and middle nodes of the tank. The portion of the tank heated by the auxiliary heater is not affected until the temperature of the lower portion of the tank rises above the auxiliary set point. At six hours, the circulation heater and pump are disabled. Through out the test, the bottom most thermocouple, T15, lags behind due to increased losses through the bottom surface of the tank.

Figure 7 plots the heat transfer across the circulation heater and heat exchanger calculated from the mass flow rate and temperature differences (as shown) and the measured electrical power drawn by the circulation heater. The electrical power input to the heater is preset before the start of the experiment and remains nearly constant through out the test. Oscillations are visible in the calculated quantities at the start of the test. They are result of the finite time required for fluid to circulate from the circulation heater through the heat exchanger and return and also of the initial temperature distribution in the collector loop. For historical reasons, the circulation heater is located two stories (approximately 30 ft) above the solar storage tank. At a flow rate of one gallon per minute, the fluid takes on the order of three minutes to make a round trip. Initially, the temperature of the water in the 120 feet of heat exchanger tubing is at the tank starting temperature of 22 °C. The temperature of the water in the piping leading to and from the

circulation heater maybe two to three degrees warmer (in the summer) due to heat exchange with the ambient. Constant heat input to the collector loop at the circulation heater causes a fixed temperature rise across the heater. As cooler water from the heat exchanger reaches the inlet of the heater, the outlet temperature drops slightly and then rises again with rising inlet temperature. These oscillations may persist for as long as 30 minutes before a steady increase in outlet temperature is reached. A decrease in heat input to the heat exchanger is due to piping losses as the fluid heats up during the test is also apparent. For this test, the average value of the heat input to the heat exchanger is 1872 watts.

When overall log-mean temperature difference across the heat exchanger is defined as

$$\Delta T_{lm} = \frac{(T16 - T12) - (T17 - T12)}{\ln\left(\frac{(T16 - T12)}{(T17 - T12)}\right)} \tag{11}$$

the effective heat exchanger conductance is,

$$UA_{hx} = \frac{Q_{input}}{\Delta T_{lm}} \tag{12}$$

These quantities are plotted in Figure 8 for this test. The trace of ΔT_{lm} shows some of the oscillations visible in the plot of Q_{input}. The effective heat exchanger conductance, UA_{hx}, does become nearly constant early on in the test.

TRNSYS Comparison

A TRNSYS deck was prepared to simulate the experiment show in Figure 6. The average heat input to the collector loop measured in the experiment (1872 W) was specified as an input to the TRNSYS simulation. Likewise, the average value of collector loop flow rate (3.87 lpm) and tank environment temperature (23.9 °C) from the experiment were specified. The volume of the solar tank was set at the nominal value of 303 l (80 gal.) even though the tank measurements in Figure 1 suggest that the actual tank volume is more like 276 l (73 gal.). The height of the tank is 1.47 m (58 in.). The height of the heat exchanger is 0.76 m (30 in). The appropriate heat loss coefficient of the tank was determined in a separate heat loss experiment to be 5.164 (kJ/hr-m^2-K). Eight equal sized tank nodes were specified for the simulation to match the eight thermocouple measurements in the tank. The location of the auxiliary heater and thermostat was specified as the second (from the top) tank node (T9 in Figure 1). The reference value for the auxiliary heater power is 3375 W; 4500 W de-rated for the 208 VAC service available on this test stand. The thermostat set point and deadband were determined form experiment to be 54 °C (130 °F) and 3 °C (5 °F).

Figure 9 shows the development of the tank and heat exchanger temperatures for a TRNSYS simulation of the experiment shown in Figures 6 through 8. Unlike the experiment, the simulation responds immediately to heat input to the collector loop. The collector loop heat input and flow rate are experimental values, so the temperature difference from the inlet to the outlet of the heat exchanger (T16 - T17) must be equal to the experimental value. The temperature difference from the heat exchanger to the to the tank is, however, much smaller than observed. Figure 10 shows a plot of the log-mean temperature difference, ΔT_{lm}, and the overall heat exchanger conductance, UA_{hx}, calculated from the simulation results (reference Eqns 11

and 12). The predicted value of UA_{hx} is approximately 2.5 time larger than the measured value; consequently, ΔT_{lm}, is 40% lower.

The slope of the temperature rise in the experimental tank is slightly larger than the TRNSYS simulation. This indicates that the actual volume of the tank is, indeed, slightly less than the nominal volume.

Table 2 compares the average log-mean temperature difference, ΔT_{lm}, and effective overall heat exchanger conductance, UA_{hx}, determined from experiment to those predicted by TRNSYS simulation for seven combinations of Q_{input} and $\dot{V}_{coll}$. In the first series, the heat input was varied from approximately 1000 W to 4000 W with the collector loop flow rate fixed at approximately 3.78 l (1 gpm). In the second series, the heat input was fixed at approximately 2000 W and the collector loop flow rate was varied. Note, results of Test 6 are merely an echoed of Test 2. In Test 5, the heat input to the collector loop dropped well below the target value (2000W) for this series.

Table 2. Comparison of measured and predicted values of log-mean temperature difference and effective heat exchanger conductance.

				Experiment		TRNSYS	
Test	Q_{input} W	Flow Rate lpm	Flow Rate gpm	ΔT_{lm} K	UA_{hx} kW/K	ΔT_{lm} K	UA_{hx} kW/K
1	992	3.98	1.0	5.5	0.153	2.5	0.388
2	1872	3.78	1.0	10.3	0.166	4.4	0.431
3	2474	3.73	1.0	15.7	0.147	5.5	0.449
4	3472	3.70	1.0	21.7	0.148	7.2	0.481
5	1171	1.03	0.27	7.0	0.108	4.4	0.266
6	1872	3.78	1.0	10.3	0.166	4.4	0.431
7	1799	7.61	2.0	10.5	0.163	3.7	0.486
8	1729	11.78	3.1	10.5	0.159	3.3	0.517

The experimental values for UA_{hx} do not change greatly with either increasing ΔT_{lm} in the first series or with increasing collector flow rate in the second. The largest change in UA_{hx} occurs for the case of low flow rate and low heat input (ΔT_{lm}) in Test 5. The TRNSYS model consistently over predicts the effective heat exchanger conductance by a factor of 2.5 to 3.0. The effect of increasing ΔT_{lm} and increasing flow rate is only a little more pronounced for the TRNSYS results.

That the TRNSYS model described above predicts a larger heat exchanger UA_{hx} product than is observed is not surprising. On the coil side of the heat exchanger, the Nusselt number correlation employed applies to uniform heat transfer around their circumference of the tube,

while, the heat transfer occurs primarily on one side of the tube in this case. In addition, one would expect, based an observation of the joint between the coil and the tank, that the overall surface efficiency, $\eta_{0,coil}$ would indeed be less than one. The conduction heat transfer resistance through the bond and tank wall may indeed be significant. Perhaps most important, a Nusselt number correlation for free convection on a flat vertical surface in an infinite medium was used to estimate the heat transfer coefficient on the tank wall. The presence of the enclosure and the effect of the (slightly) stratified medium both inhibit the natural convection circulation in the tank and reduce the heat transfer coefficient.

Recommendation for Additional Experiments and Analysis

The current model of this unique combination of tank and heat exchanger clearly over-estimates the effective heat transfer conductance of this heat exchanger by a factor of 2.5 or more. A means of correcting the model and/or normalizing the model to the experimental data must be identified. These comparisons suggest that the model does in fact over estimate the free convection heat transfer on the tank side of the heat exchanger.

In any case, a more complete data set should be acquired to insure that the resulting model is applicable over the widest possible range of operating conditions. Of particular interest, is the performance of the heat exchanger with low collector flow rates.

In order to make comparison of the simulation results with measured values, the experiment should be modified to reduce the heat loss in the piping between the heater and the tank and also to reduce the oscillation at the beginning of the test. In further testing, a test with constant temperature at the inlet of the heat exchanger should be considered. Such an experiment would varies the delta T across the heat exchanger continuously while the flow rate is held fixed.

Once model modifications are adopted, a simulated solar day test should be conducted to insure that the model works under the real world conditions.

Auxiliary Heater Thermostat Tests:

At the request of Jim Huggins of the Florida Solar Energy Center (FSEC), the upper set point and control deadband of three electric auxiliary heater thermostats were determined insitu on our Rheem wrap-around heat exchanger solar storage tank. Two thermostats were supplied by FSEC. They are arbitrarily labeled FSEC 1 and 2. The thermostat supplied with our tank was also tested. It is labeled as CSU 1. The thermostats were all Therm-o-disc model 4000-59T. All of the thermostats were nominally set at 130 °F.

Each thermostat was installed on the Rheem wrap-around heat exchanger tank in turn. The tank was preconditioned to 22 °C, and the auxiliary heater was enabled and allowed to operate for a 24 hour period. The auxiliary heater cycled on and off several times during each test. The eight tank temperatures, the tank environment temperature, and the auxiliary heater power were recorded at 5 second intervals. The tank temperature histories are plotted and a summary lists the tank conditions whenever the heater cycled.

Note, the auxiliary heater element is rated at 4500 W at 240 VAC. As installed, it is derated to 3375 W for our 208 VAC service.

The temperature histories for tests of the FSEC 1, FSEC 2 and CSU 1 thermostats are shown in Figures 11, 12, and 13 respectively. The highest temperatures shown on the plots are the

traces for the top two thermocouples (T8 and T9). The two traces coincide. The trace for temperature T10 lies below T8 and T9. Thermocouple T10 lies just below and slightly to the side of the auxiliary heater element. The temperature of the water at T10 is strongly influenced by the operation of the auxiliary heater but does not reach the temperature of the upper portion of the tank. The remainder of the tank is heated slowly as energy is conducted downward (through the water and walls) during the 24 hour period of the test. The temperature at thermocouple T9 is noted on the plots whenever the auxiliary heater cycles. This number is an instantaneous value. Note that in each test, the apparent set point is of the auxiliary heater thermostat after the initial heat-up of the tank is approximately 2°C lower than for each of the remaining cycles.

The data sheet accompanying each of the Figures summarizes the conditions during each test. The timing summary shows when the auxiliary heater cycled on and off. The average initial tank temperature and average tank environment temperature are listed. The average auxiliary heater power is given and the total energy supplied to the tank is calculated for each cycle. Tank temperature profiles are listed at each instant when the heater cycled.

Table 3 summarizes the results of the tests. T_{off} is the value of thermocouple T9 at the instant that the heater shuts off. T_{on} is the value of T9 when the heater comes back on. The apparent set point temperature is calculate by averaging the values of T_{off} excluding the initial shut-off point. ΔT is the difference between T_{off} and the next T_{on}. Note that both the value of T_{off} and T_{on} seem to be decreasing over time and that the difference, ΔT, is increasing. This is probably related to the conduction of heat to the colder bottom of the tank.

Table 3. Summary of experimental results. (Temperatures in °C)

		1	2	3	4	5	Average
FSEC #1	T_{off}	55.2	57.1	57.2	57.1	56.8	57.1*
	T_{on}	53.7	53.6	53.0	52.4	N/A	53.2
	DT	1.5	3.5	4.2	4.7	N/A	4.1*
FSEC #2	T_{off}	57.1	58.7	58.4	58.2	57.9	58.3*
	T_{on}	54.9	54.8	54.0	53.4	N/A	54.3
	DT	2.2	3.9	4.4	4.8	N/A	4.4*
CSU #1	T_{off}	55.9	57.3	57.1	56.5	N/A	57.0*
	T_{on}	53.3	52.6	51.7	N/A	N/A	52.5
	DT	2.6	4.7	5.4	N/A	N/A	5.1*

* excluding the first value

These results suggest that the actual values of T_{set} and ΔT_{db} are slightly higher than the SRCC values of 55°C and 2.78°C, but not significantly higher (i.e.. 10°C). The experiments also show some variation in both T_{set} and ΔT_{db} from thermostat to thermostat. In fact, the CSU 1 thermostat seems to have a larger deadband.

Figure 14 shows results of an earlier test using our original thermostat (CSU #1) at the factory setting of 120°F. Figures 15 and 16 show the results of TRNSYS simulations run for comparison. In Figure 14, the auxiliary parameters were set to nominal values of $T_{set}=120°F$,

$\Delta T_{db}=5°F$, and $Q_{aux}=3375$ W (4500 W derated for 208 VAC service). The loss coefficient of the tank was set to a value of 14.4 kJ/hr-K as determined in a separate heat loss experiment (see discussion above). The environment temperature is specified at 22°C. In Figure 16, experimentally determined values of all of the parameters are used (as shown).

The measured total energy used by the auxiliary heater was $1.45x10^4$ kJ. For the first TRNSYS run (Figure 15), the predicted value is $1.03x10^4$ kJ; 71.3% of the measured value. For the second TRNSYS run (Figure 16), the predicted value is $1.23x10^4$ kJ; 85.1% of the measured value. Comparison of the temperature profiles from the experiment , Figure 14, with the simulation, Figure 16, suggests the source of the remaining difference in energy used. The portion of the actual tank that is heated by the auxiliary heater looses energy to the remainder of the tank through conduction (in water and tank walls) as well as to the environment. The TRNSYS tank model used here does not account for this conduction within the tank and, therefore, predicts a slower loss rate for the upper part of the tank and a lower auxiliary energy requirement. Time permitting, I plan to add the conduction term to the tank model.

References:

Aranovitch, E., D. Gilliaert, W.B. Gillett and J.E. Bates, editors, "Non Nuclear Energies European Solar Collector and Systems Testing Group Recommendations for performance and Durability Tests of Solar Collectors and Water Heating Systems," Published by Commission of the European Communities EUR 11606 EN May 1989

Miller, J. A., and D. C. Hittle, "Yearly Simulation of a PV Pumped, Wrap-Around Heat Exchanger, Solar Domestic Hot Water System," *Solar Engineering 1993, ASME/ASES/ISES Solar Energy Conference*, Washington, D.C., April 4-9, 1993.
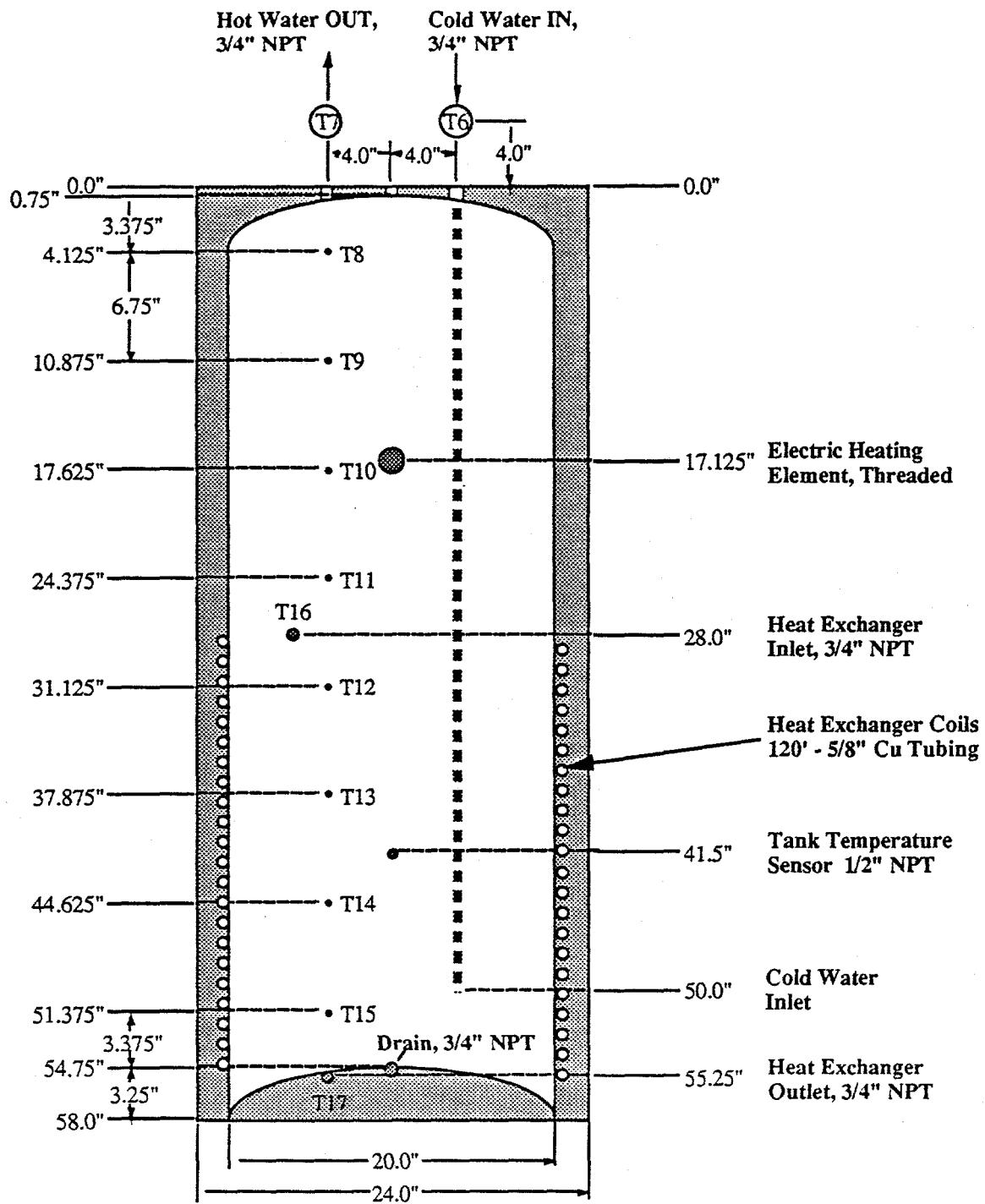
Figure 1. Schematic of the wrap-around heat exchanger tank. Locations of thermocouple tree sensors are shown. All measurements are from the top of the outer shell of the tank. Internal geometry and dimensions are inferred from external measurements. All dimensions to 1/8".
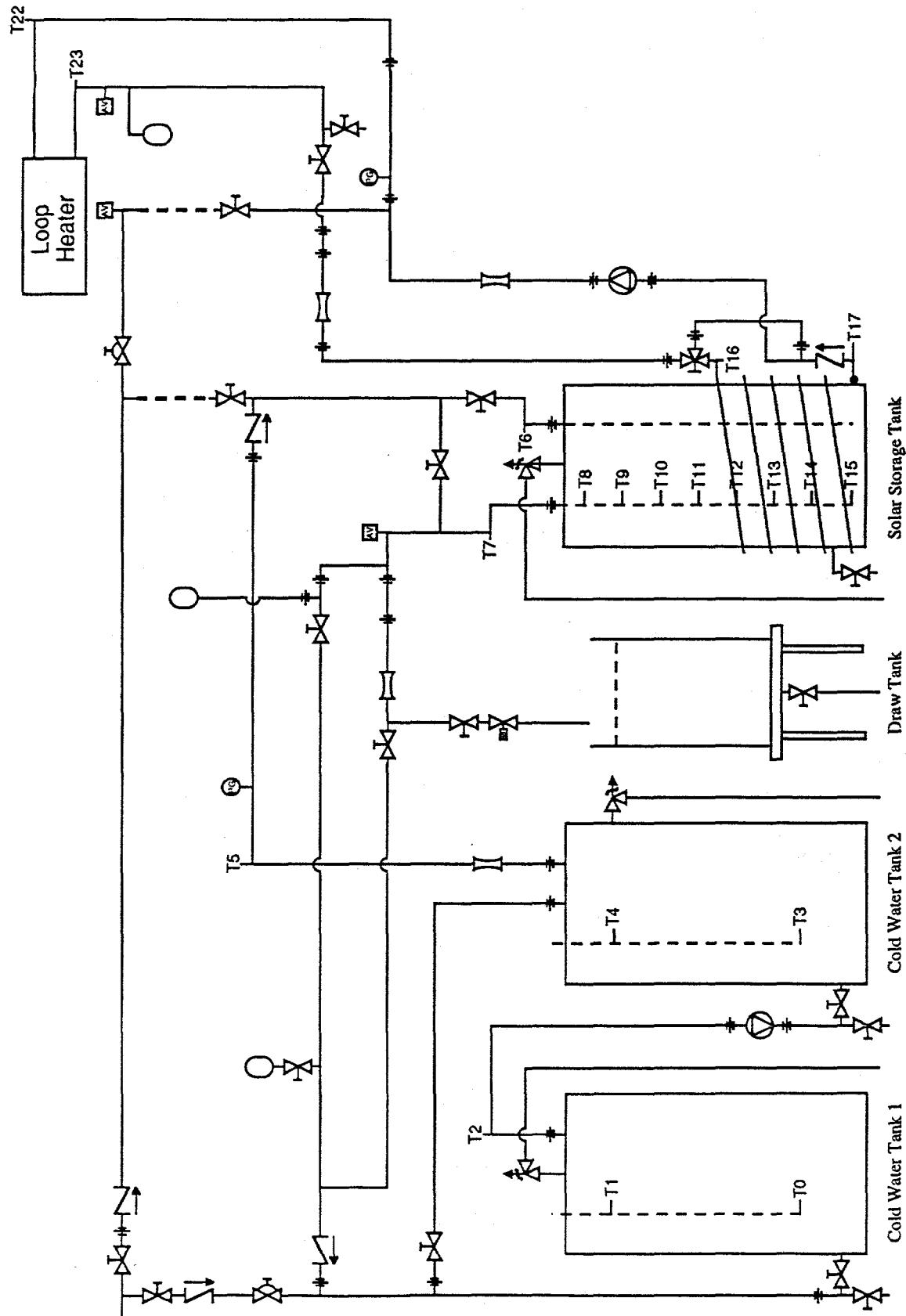
Figure 2. Schematic of the Colorado State Universities solar storage tank test stand with the wrap-around heat excahnger installed. Piping sysmbols follow ASHRAE standards.
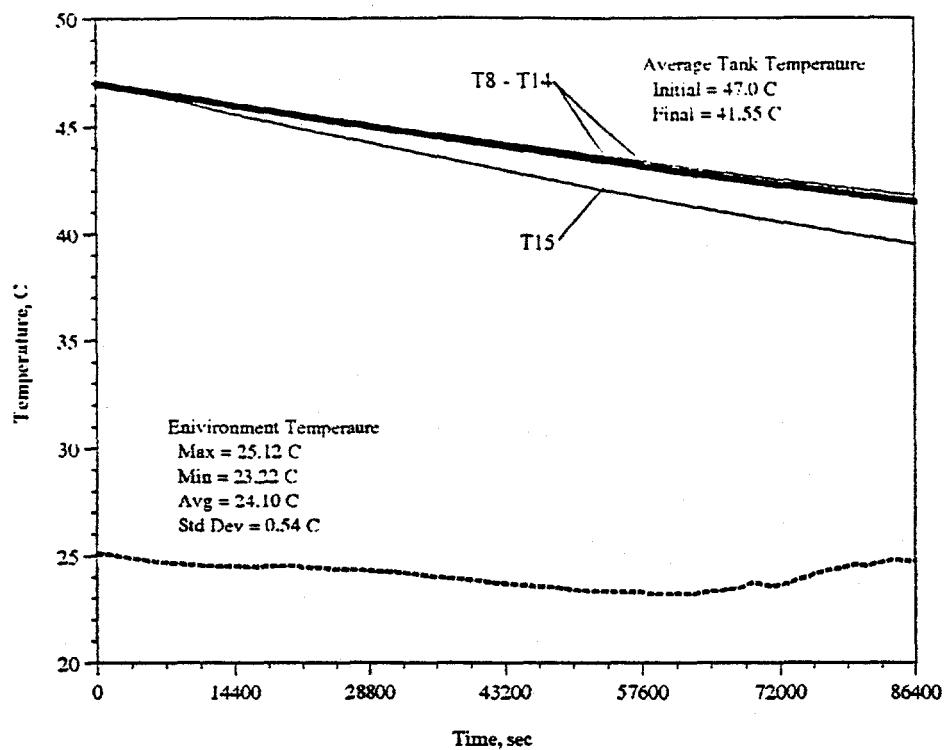
Figure 3. Development of tank temperatures in the Rheem wrap-around heat exchanger tank during a 24 hour tank heat loss test.
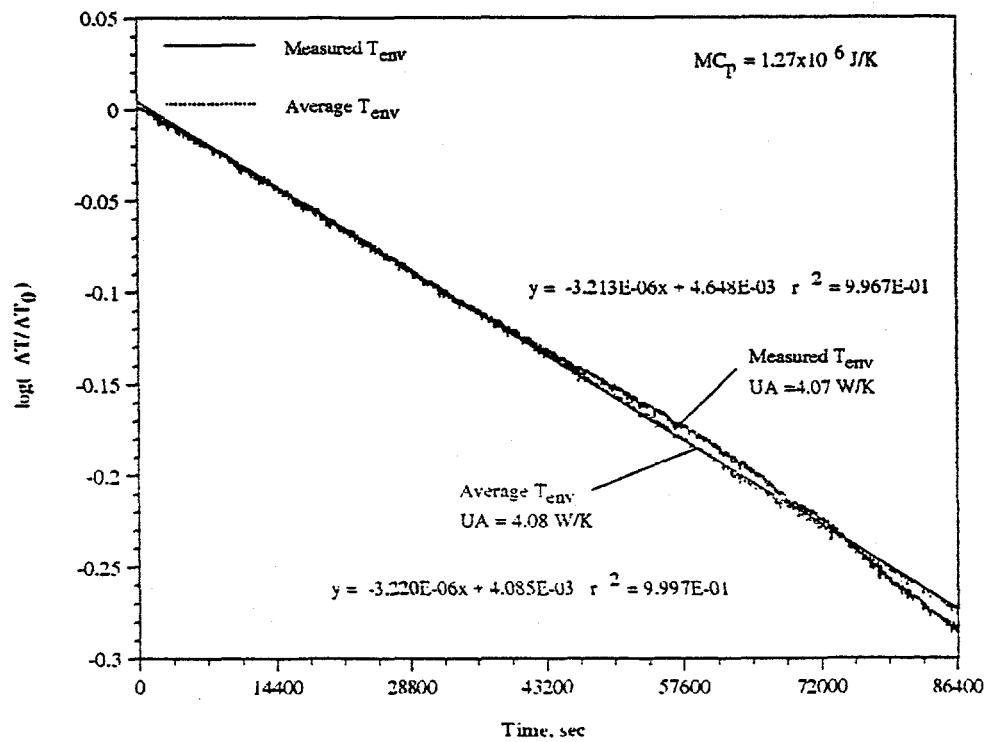
Figure 4. Calculation of the effective overall heat loss coefficient of the Rheem wrap-around heat exchanger tank. (Reference Eqn. 5).
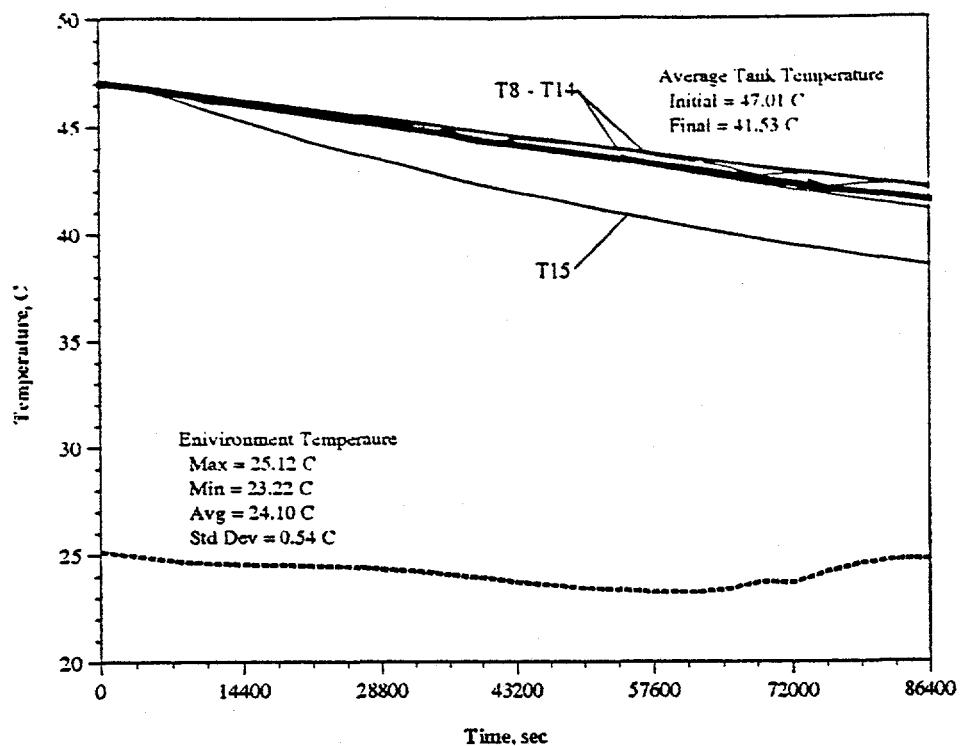
Figure 5. Tank temperatures predicted by TRNSYS for the 24 hour heat loss test. Eight tank nodes and the experimental variation in the tank environment temperature were specified.
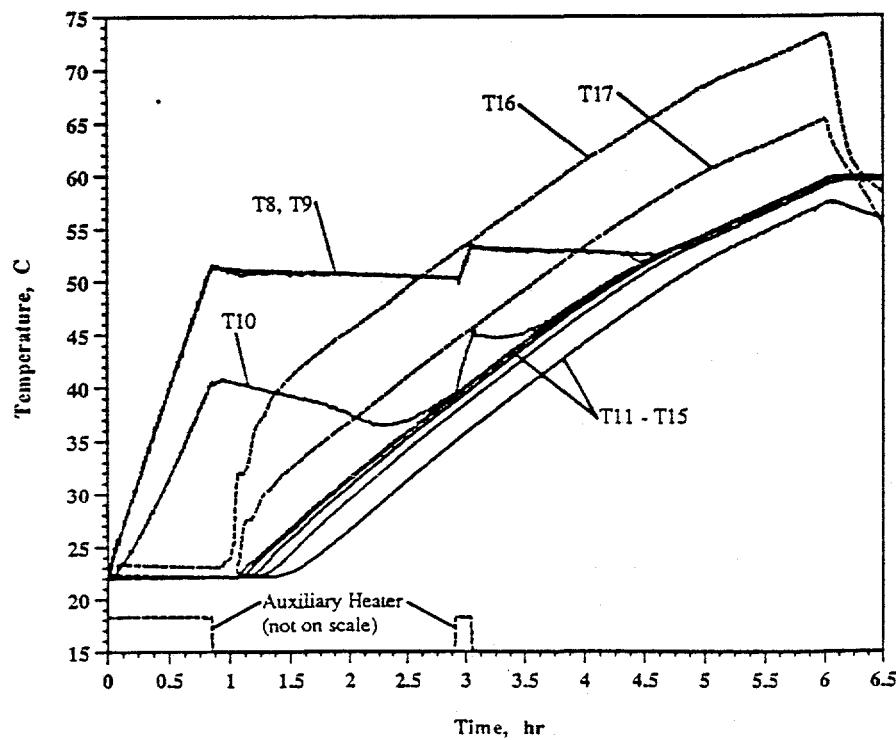
Figure 6. Development of tank and heat exchanger temperatures for a typical constant heat input test. Test 2, Q_{input} = 1872 W and $\dot{V}_{coll}$ = 3.78 lpm (1 gpm).
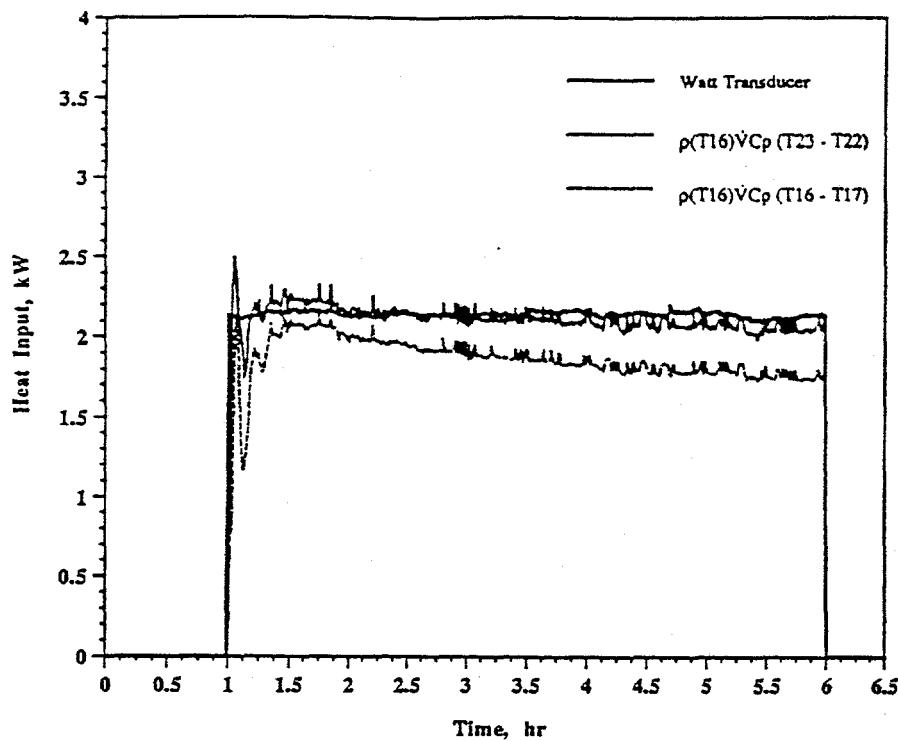
Figure 7. Comparison of the measured electrical power drawn by the circulation heater with the calculated heat transfer across the circulation heater and across the heat exchanger. Test 2, Q_{input} = 1872 W and $\dot{V}_{coll}$ = 3.78 lpm (1 gpm).
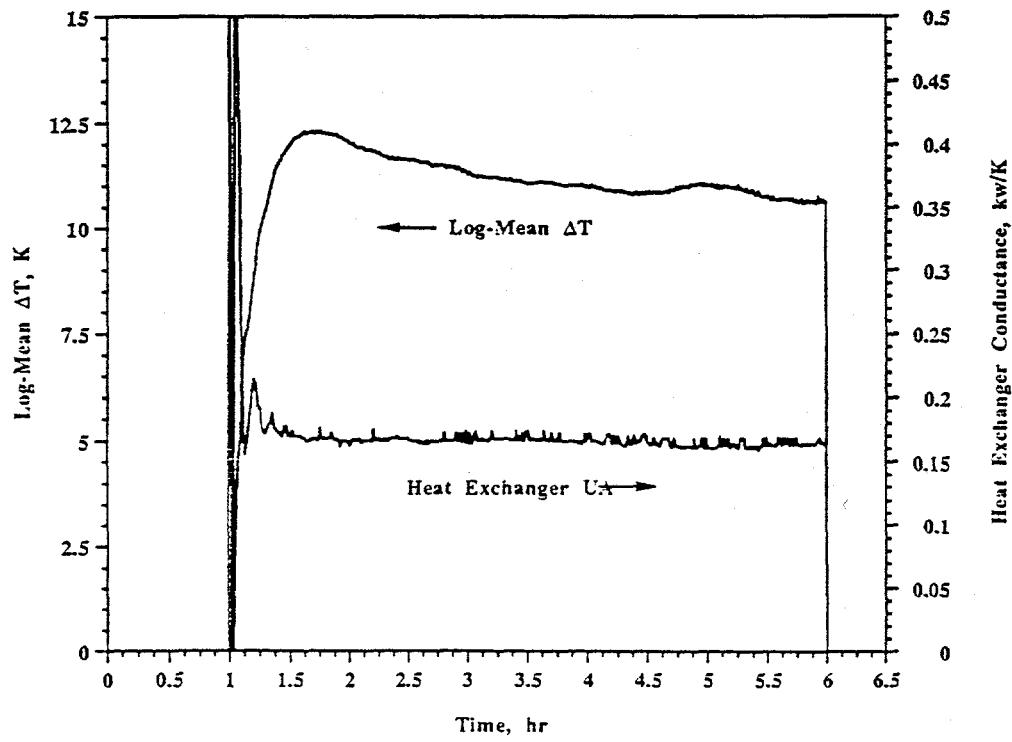
Figure 8. Overall log-mean temperature difference, ΔT_{lm}, and effective overall heat exchanger conductance, UA_{hx}, calculated from the experimental data. Test 2, Q_{input} = 1872 W and $\dot{V}_{coll}$ = 3.78 lpm (1 gpm).
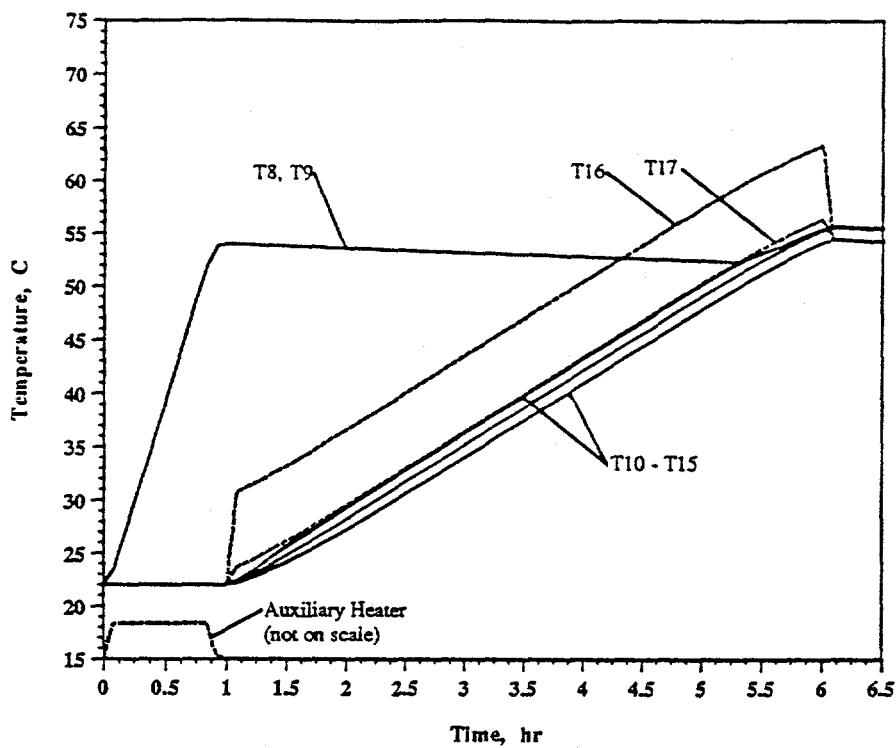
Figure 9. Tank and heat exchanger temperatures from TRNSYS simulation of constant heat input test. Test 2, Q_{input} = 1872 W and $\dot{V}_{coil}$ = 3.78 lpm (1 gpm).
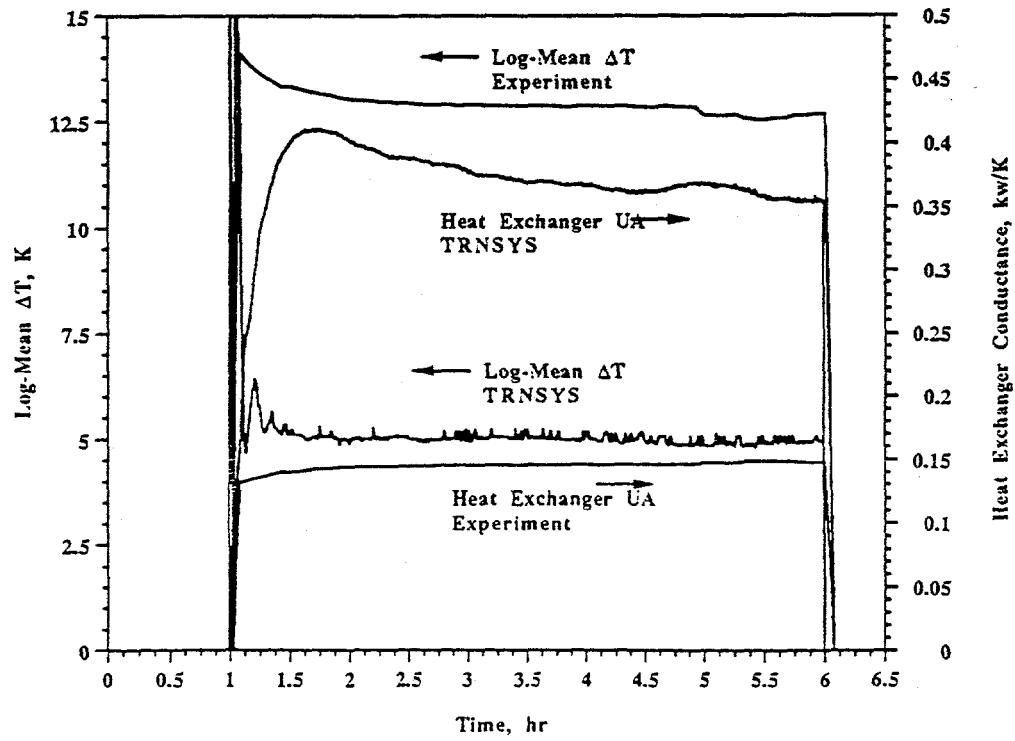
Figure 10. Comparison of overall log-mean temperature difference, ΔT_{lm}, and effective overall heat exchanger conductance, UA_{hx}, calculated from TRNSYS results and from the experimental data. Test 2, Q_{input} = 1872 W and $\dot{V}_{coil}$ = 3.78 lpm (1 gpm).
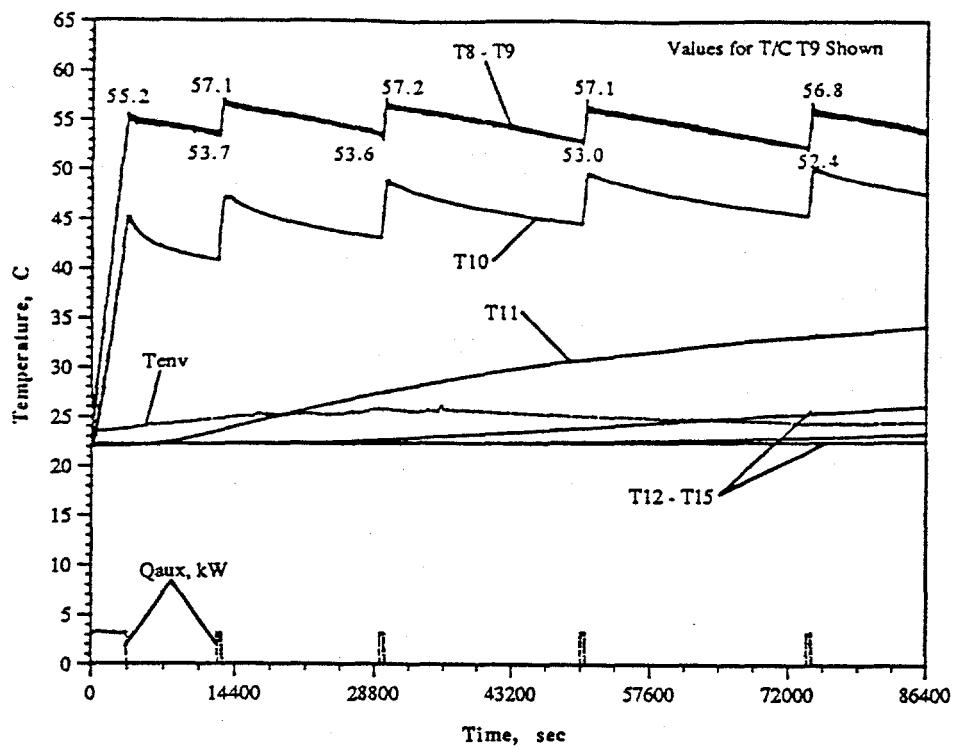
Figure 11. Tank temperature history from test of FSEC 1 thermostat. Thermostat upper set point was 130 °F.
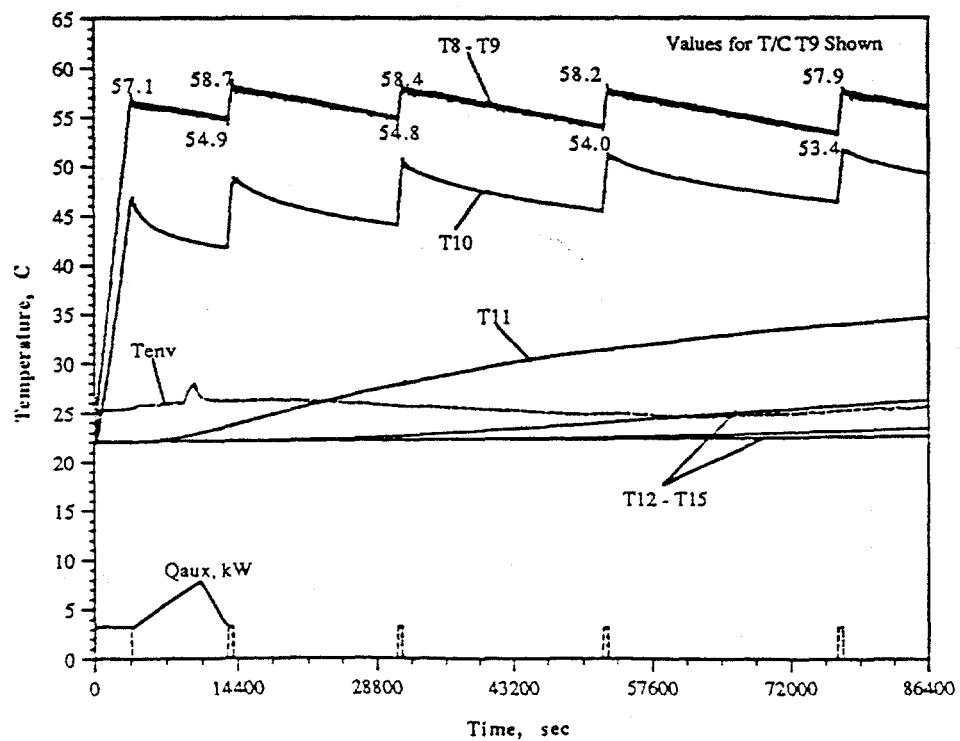
Figure 12. Tank temperature history from test of FSEC 2 thermostat. Thermostat upper set point was 130 °F.
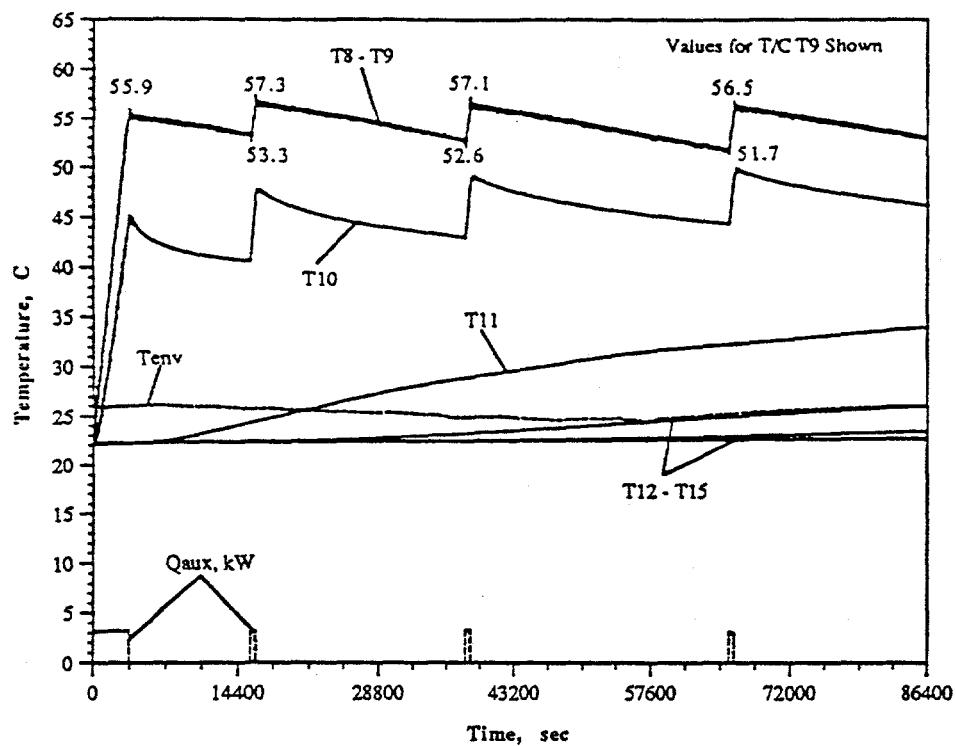
Figure 13. Tank temperature history from test of CSU 1 thermostat. Thermostat upper set point was 130 °F.
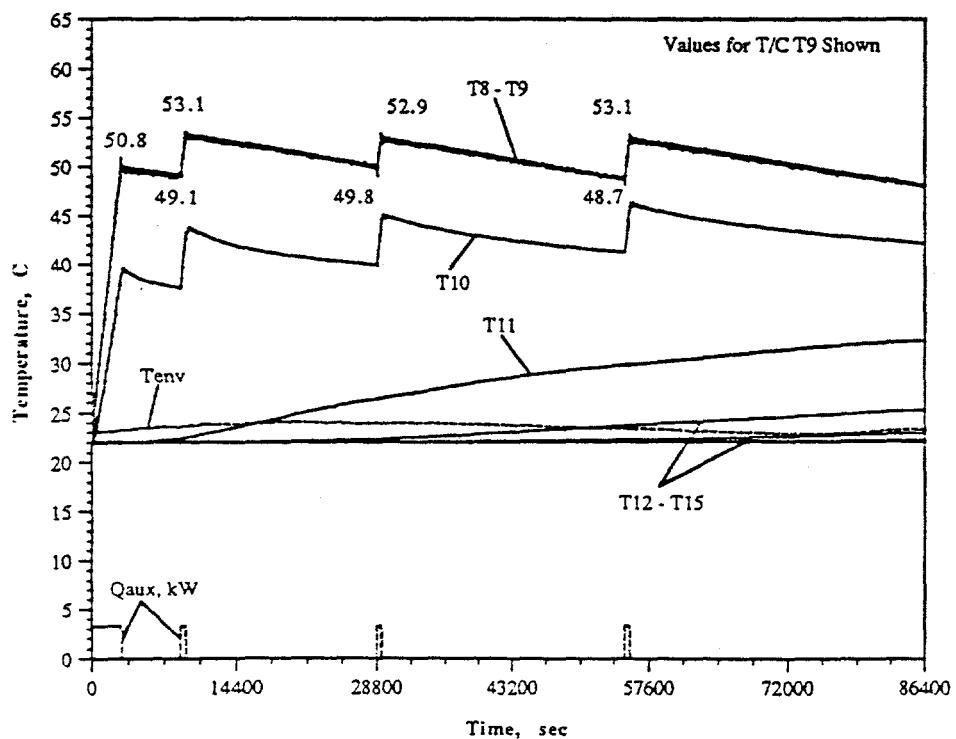
Figure 14. Tank temperature history from test of CSU 1 thermostat. Thermostat upper set point was 120 °F.
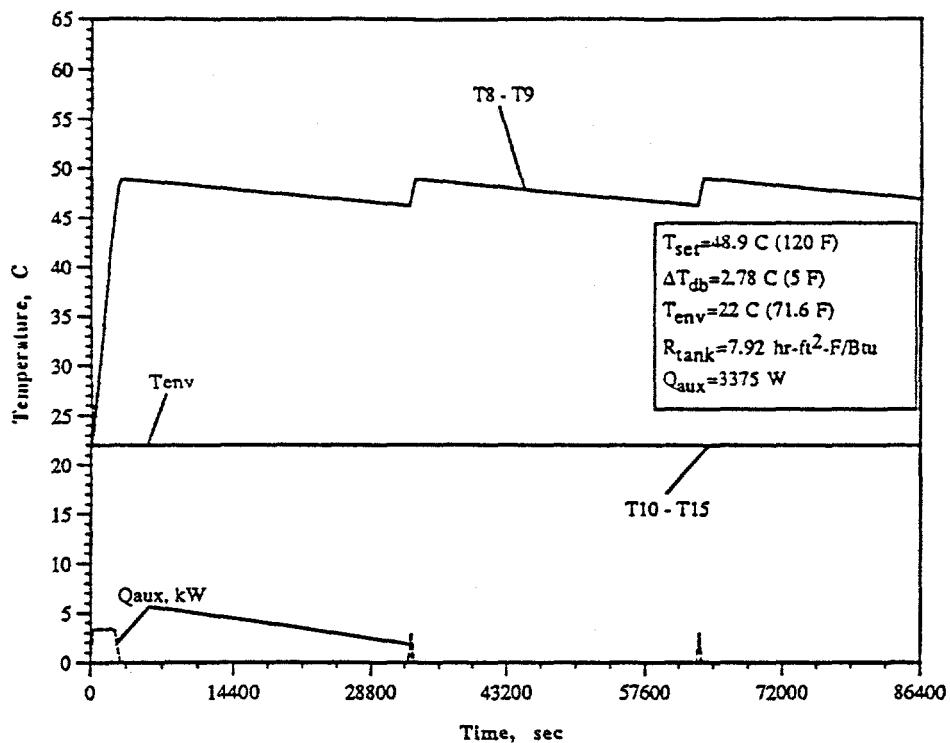
Figure 15. Tank temperatures predicted by TRNSYS for comparison with test of CSU 1 thermostat. Simulation conditions as shown.
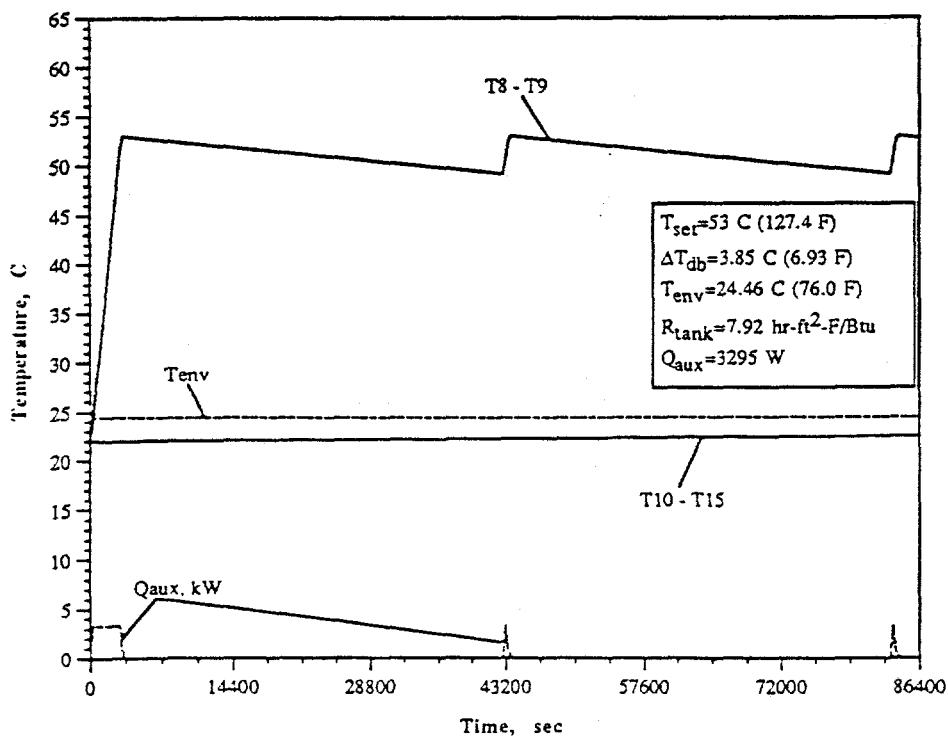
Figure 16. Tank temperatures predicted by TRNSYS for comparison with test of CSU 1 thermostat. Simulation conditions as shown.

ADVANCED RESIDENTIAL SOLAR DOMESTIC HOT WATER (SDHW) SYSTEMS

This report is for June and July, 1994. Experimental tests are being conducted on three side by side systems: ASN, NEG and Thermodynamics.

Over the course of these two months several experimental test runs were completed and analyzed. The first of these tests was run from June 1 to June 5; the systems were operated normally with three draws per day taken, no auxiliary heaters were used. Upon analysis of this data set several operational problems were recognized and fixed. These problems were:

(1) The pump on the NEG system worked erratically. The pump motor would turn on when needed, but the water was not always actually pumped. The cause was evaporation of fluid in the collectors in conjunction with the mains supply valve being kept closed; this caused vapor lock in the pump. Originally, mains supply was kept closed (except during a draw) because of severe water hammer. A surge suppresser was installed and the mains supply valve is now kept open.

(2) The thermosyphon loop on the thermodynamics system is prone to vapor lock if the system sits inactive for any period of time. An air vent was installed on this loop to aid in expelling the air.

(3) Since ASN is a drainback system, the pump line becomes void of water if the system sits for too long, hence the pump would have to be manually cycled before working. A hose fill line was added to the pump line to aid in this initial start up of the system.

(4) The ASN pressure difference data appeared to be erroneous. Upon further inspection it was discovered that certain valves that should have been open were actually closed.

(5) The 5 minute averages of the data are not sufficient to perform energy balances during a draw. The FORTRAN code needed to be modified to extract out the 8 second data for the time during draws.

Tank heat loss tests were performed on June 7-8 and June 9-10. For the June 7-8 test the valves on each system were in the "closed" position, for the June 9-10 test, the valves were kept in the "open" position (normal operation). Also during this time a collector heat loss test (the system pump was operated at night), and a test where the pumps were continuously operated during the day was run. Both of these tests will have to be repeated because of the above mentioned problems.

Finally, a test was run from July 25-29. The systems were operated normally with three draws per day executed. Graphs of the system operations are included for this time period.

Initial work was begun with TRNSYS modeling of the systems.

# Outdoor Weather Conditions

TEST 2, July 24-29, 1994
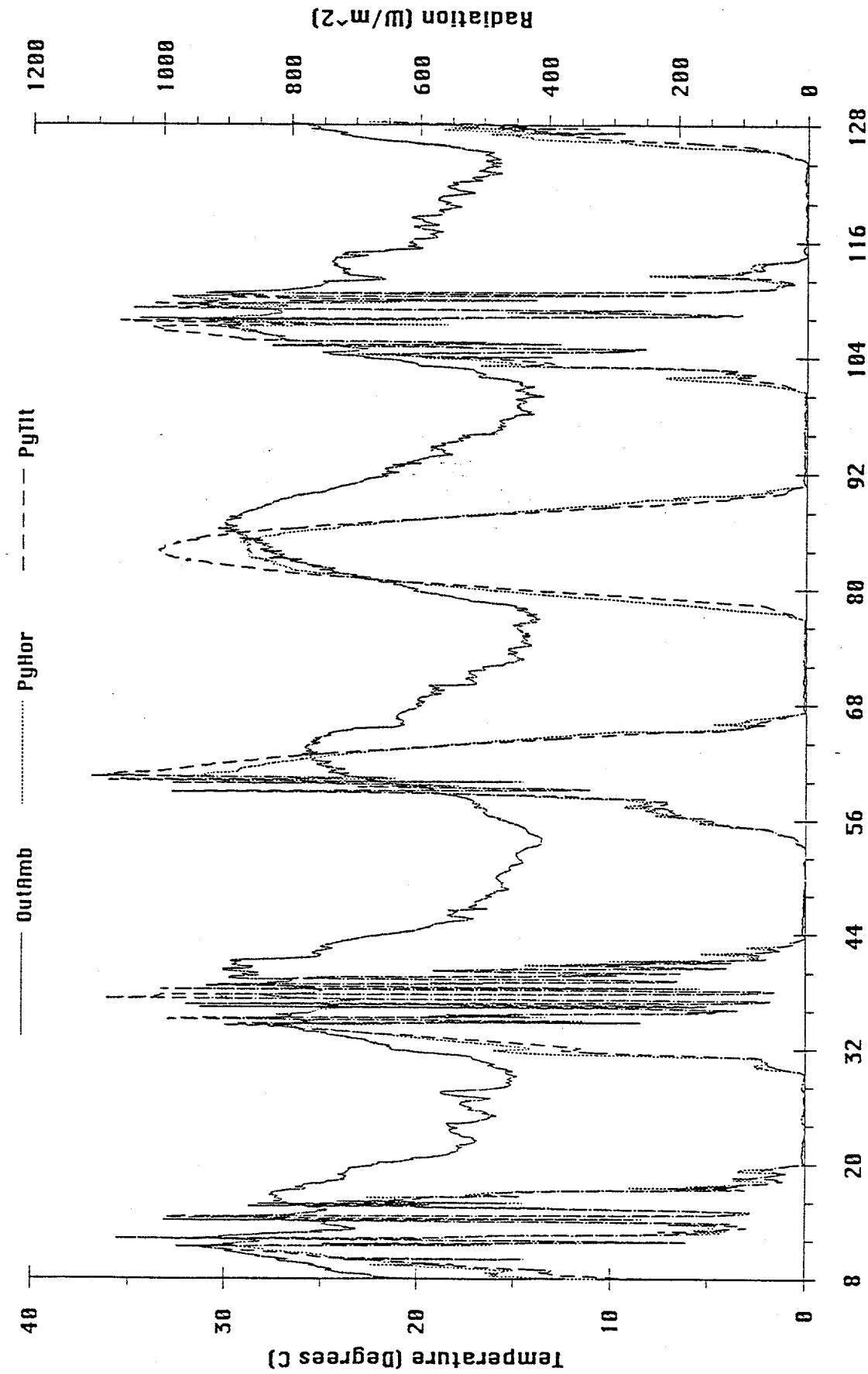
----------- OutAmb   .... PyHor   — · — · — PyTlt

Radiation (W/m^2)

Temperature (Degrees C)

Time, hours starting at 8 a.m.

Indoor Temperatures

IndrAmb ——— SupplyT — — — DrainT ········

Time, hours starting at 8 a.m.

Temperature (degrees C)

ASN System Temperatures

Legend: InAux ——— O/SBKCl — — — O/SToCl ·········

X-axis: Time, hours starting at 8 a.m. (8, 20, 32, 44, 56, 68, 80, 92, 104, 116, 128)

Y-axis: Temperature (Degrees C) (0, 10, 20, 30, 40, 50, 60, 70, 80, 90)

ASN Main Tank Temperatures

Temperature (degrees C) vs. Time, hours starting at 8 a.m.

Legend: MainBot, Main7, Main6, Main5, Main4, Main3, Main2, MainTop
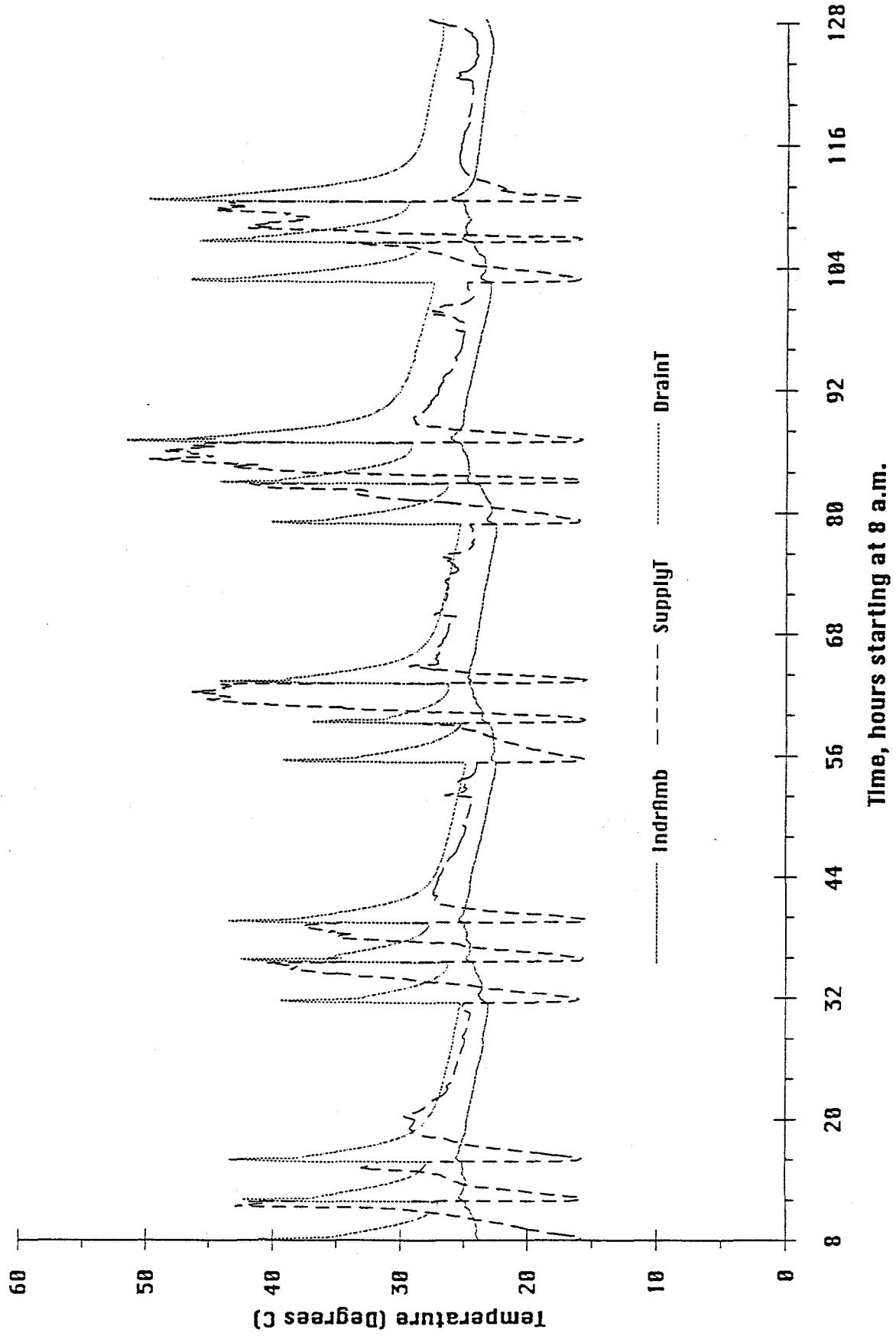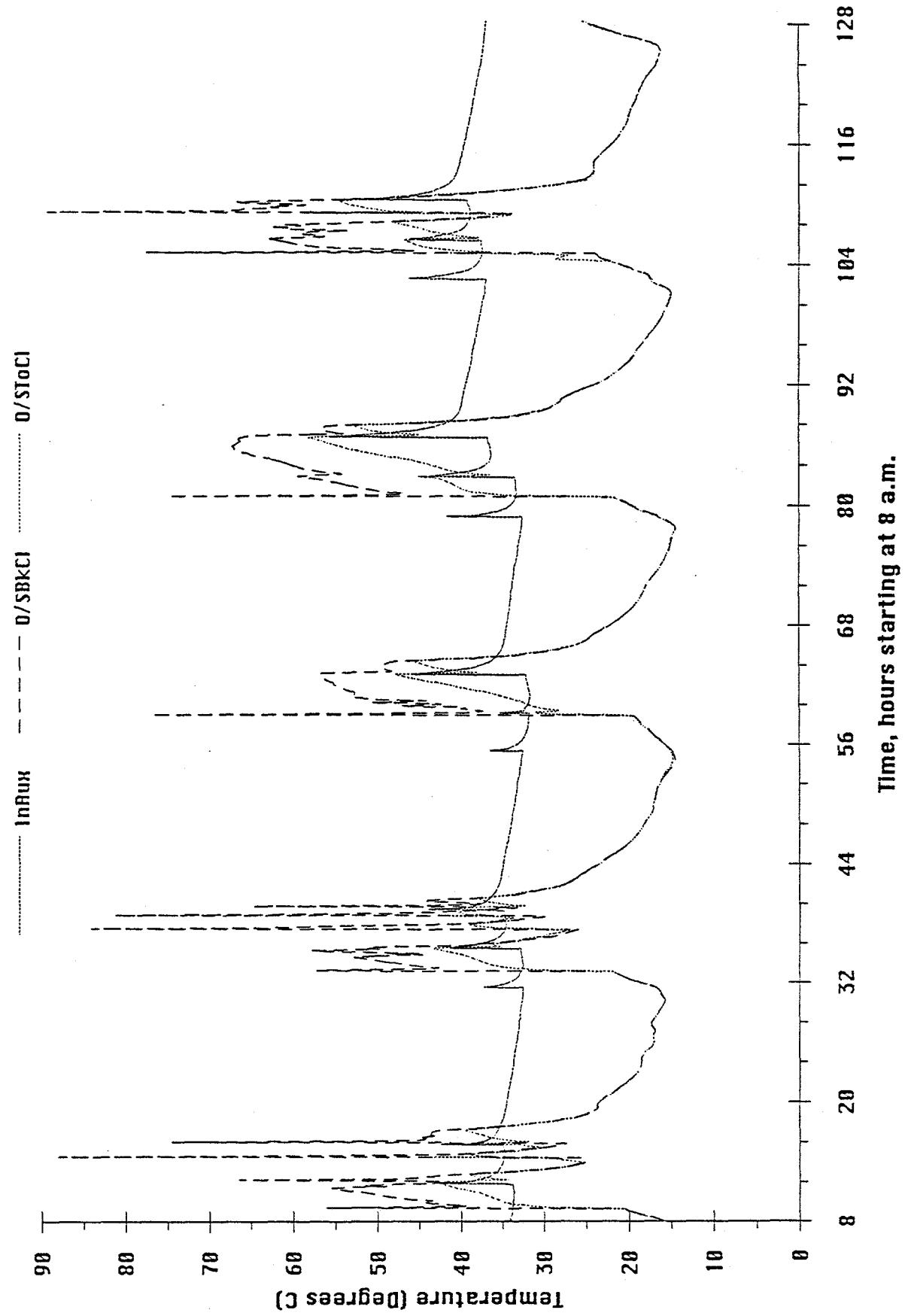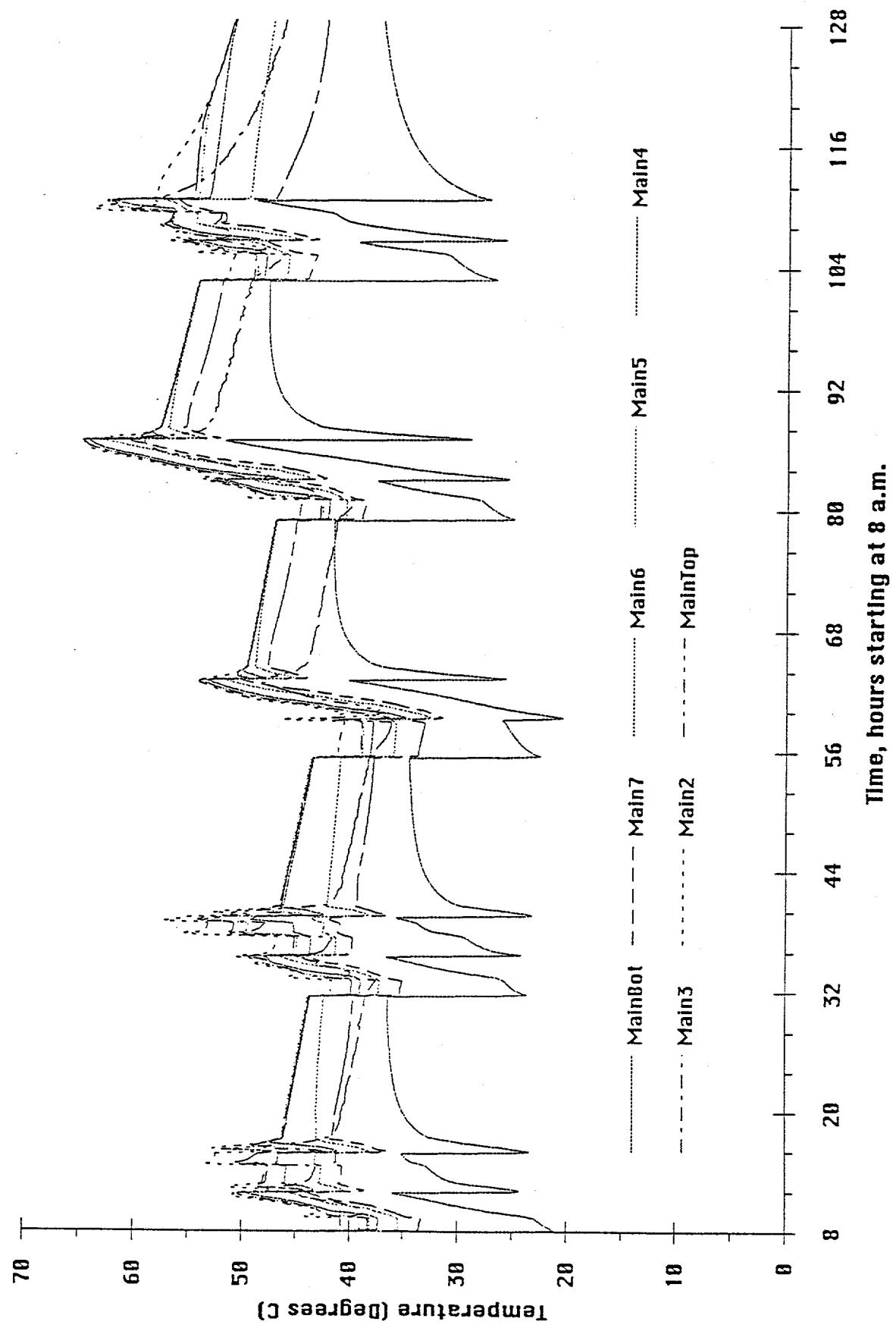
ASN Auxiliary Tank Temperatures

Temperature (degrees C)

Time, hours starting at 8 a.m.

AuxBot ———   Aux7 – – –   Aux6 ·········   Aux5 ———   Aux4 ———

Aux3 –·–·–   Aux2 ·········   AuxTop – – –

ASN Flow and Pump Power

Flow ——— PmpX – – –

NEG System Temperatures

Temperature (Degrees C)

Time, hours starting at 8 a.m.

InTnk ——— BkTnk – – – – InPmp ········· BkPmp ——— O/SToCl ——— O/SBkCl —·—·—

NEG Tank Temperatures

Temperature (Degrees C)

Time, hours starting at 8 a.m.

Legend: TankBot — Tank7 — Tank6 — Tank5 — Tank4 — Tank3 — Tank2 — TankTop
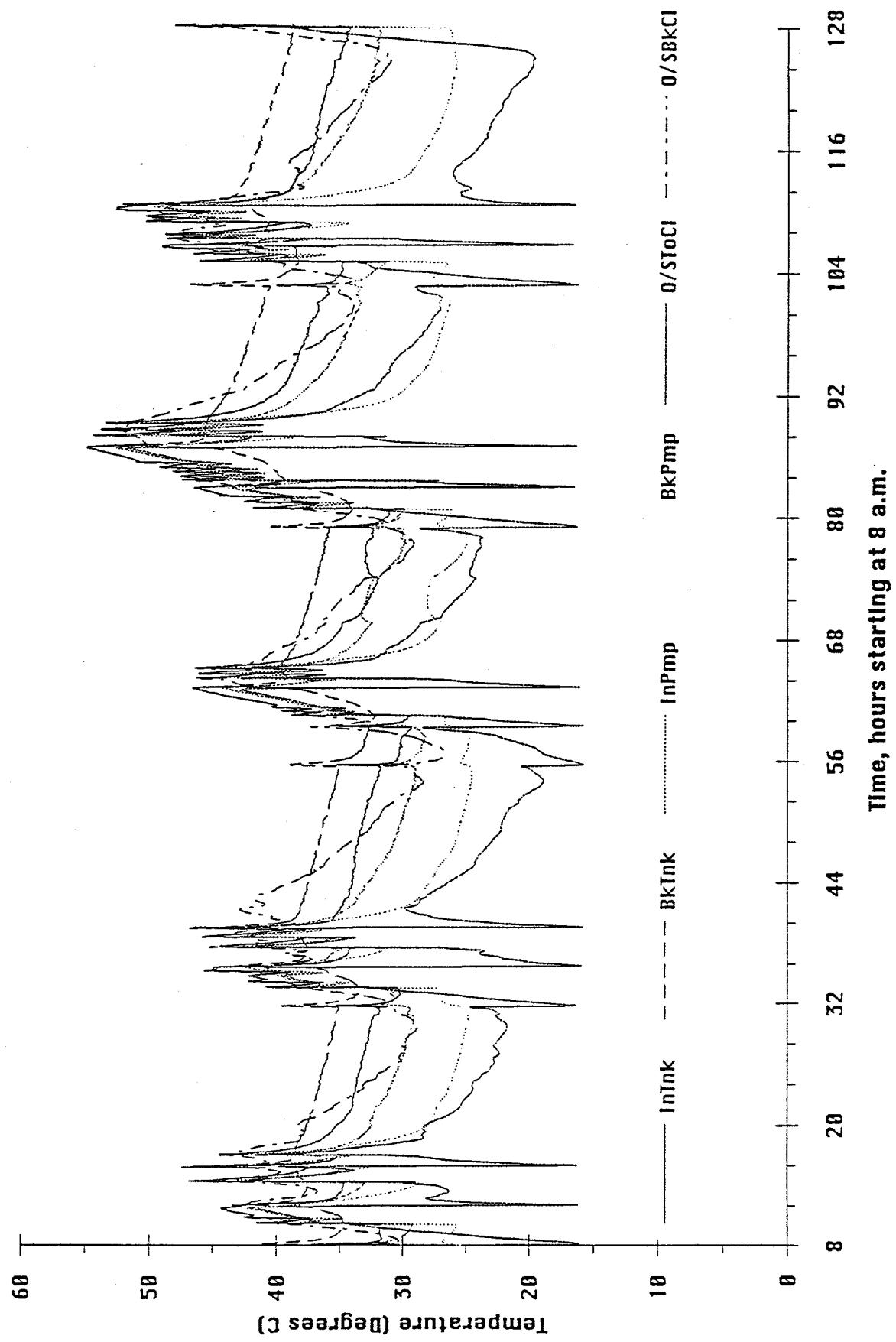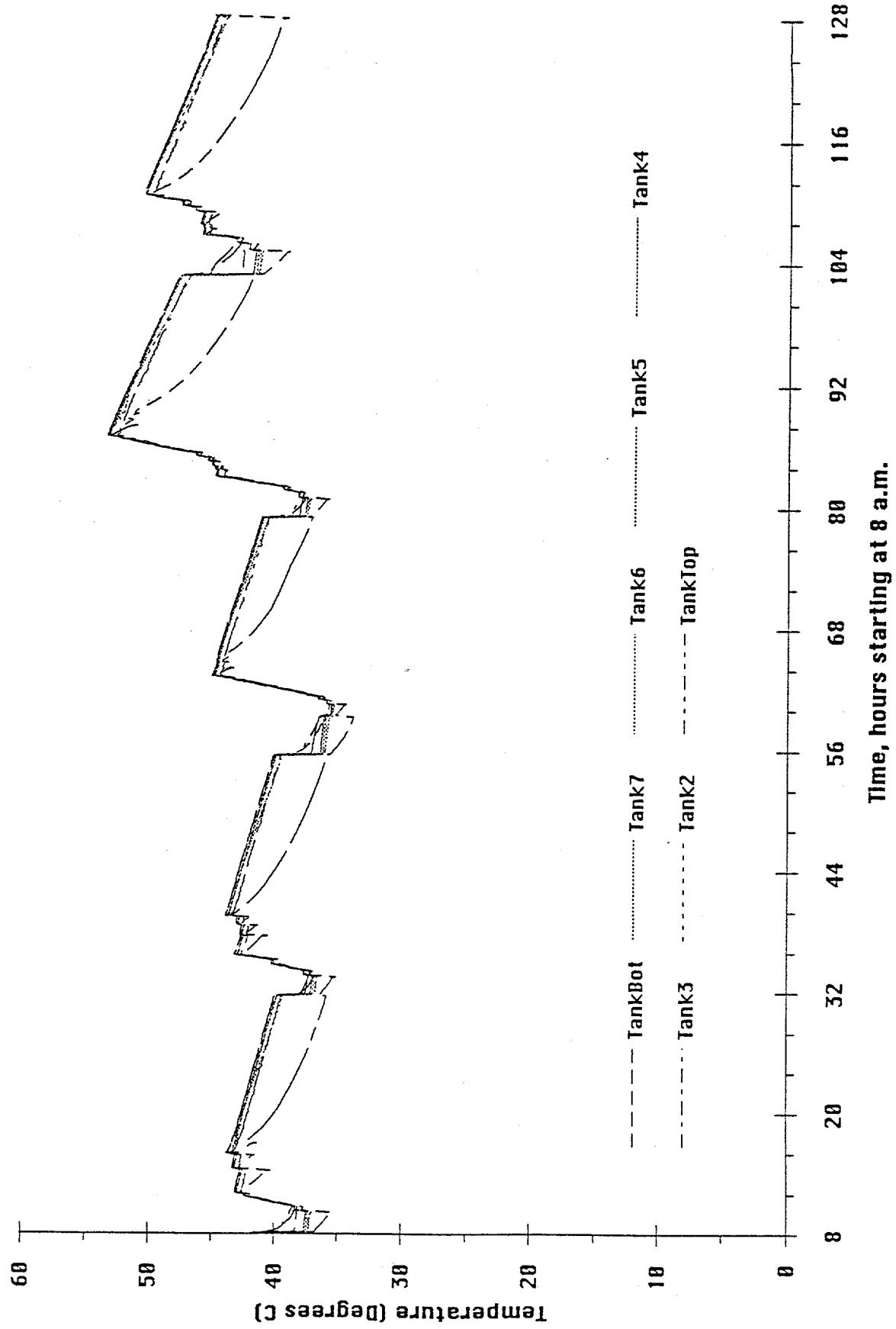
NEG Flow and Pump Power

Thermodynamics System Temperatures

InMTnk — — — BkMTnk ········· InATnk ········· O/SToCl ——— O/SBkCl

Temperature (degrees C)
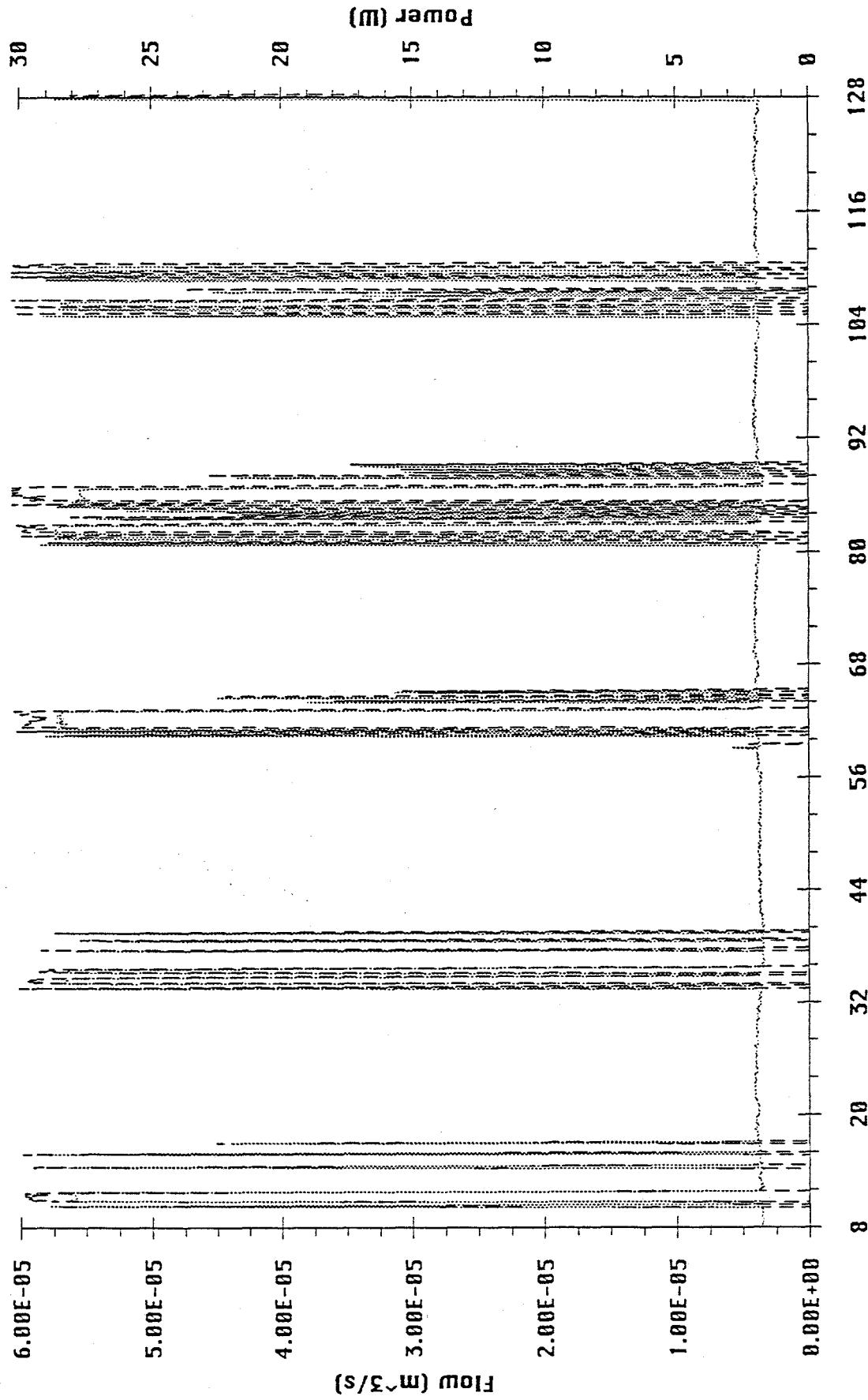
Time, hours starting at 8 a.m.
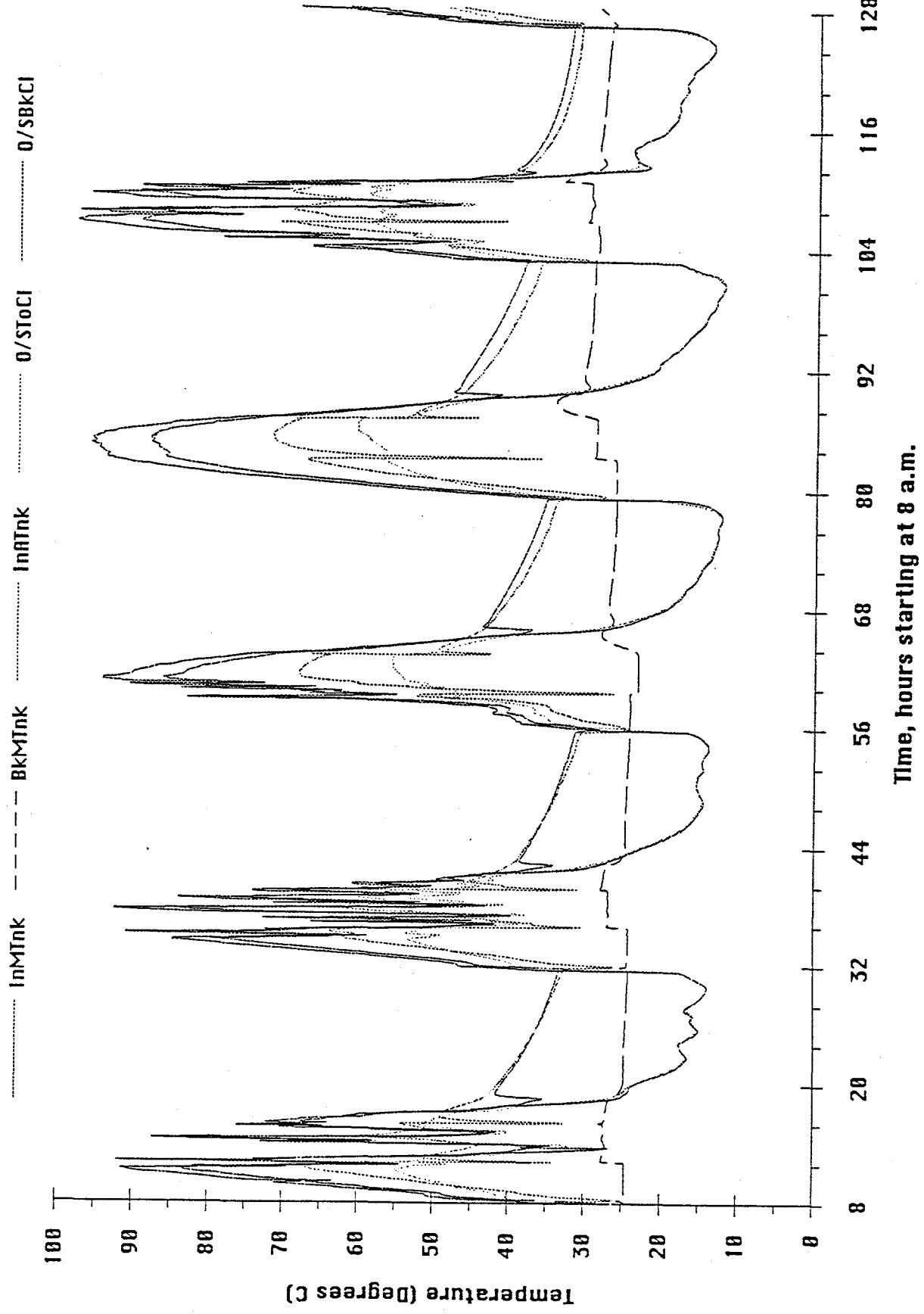
Thermodynamics Main Tank Temperatures

Temperature (degrees C)

Time, hours starting at 8 a.m.

MainBot ——— Main7 ——— Main2 ——— Main3

Main6 ——— MainTop

Main5

Main4

Thermodynamics Auxiliary Tank Temperatures

Temperature (degrees C)

Time, hours starting at 8 a.m.

AuxBot —— Aux7 ——— Aux6 ······· Aux4 ——
Aux3 ·—·—· Aux2 ·—·—· Aux5 ······· AuxTop ·—·—·

Thermodynamics Flows

MicroMotion Flow (Kg/s)

0.009 0.008 0.007 0.006 0.005 0.004 0.003 0.002 0.001 0

FluMic ------

FluFTI ———

zero @ -0.004

Time, hours starting at 8 a.m.

8 28 32 44 56 68 80 92 104 116 128

Collector Flow (m^3/s)

2.00E-05 1.50E-05 1.00E-05 5.00E-06 0.00E+00
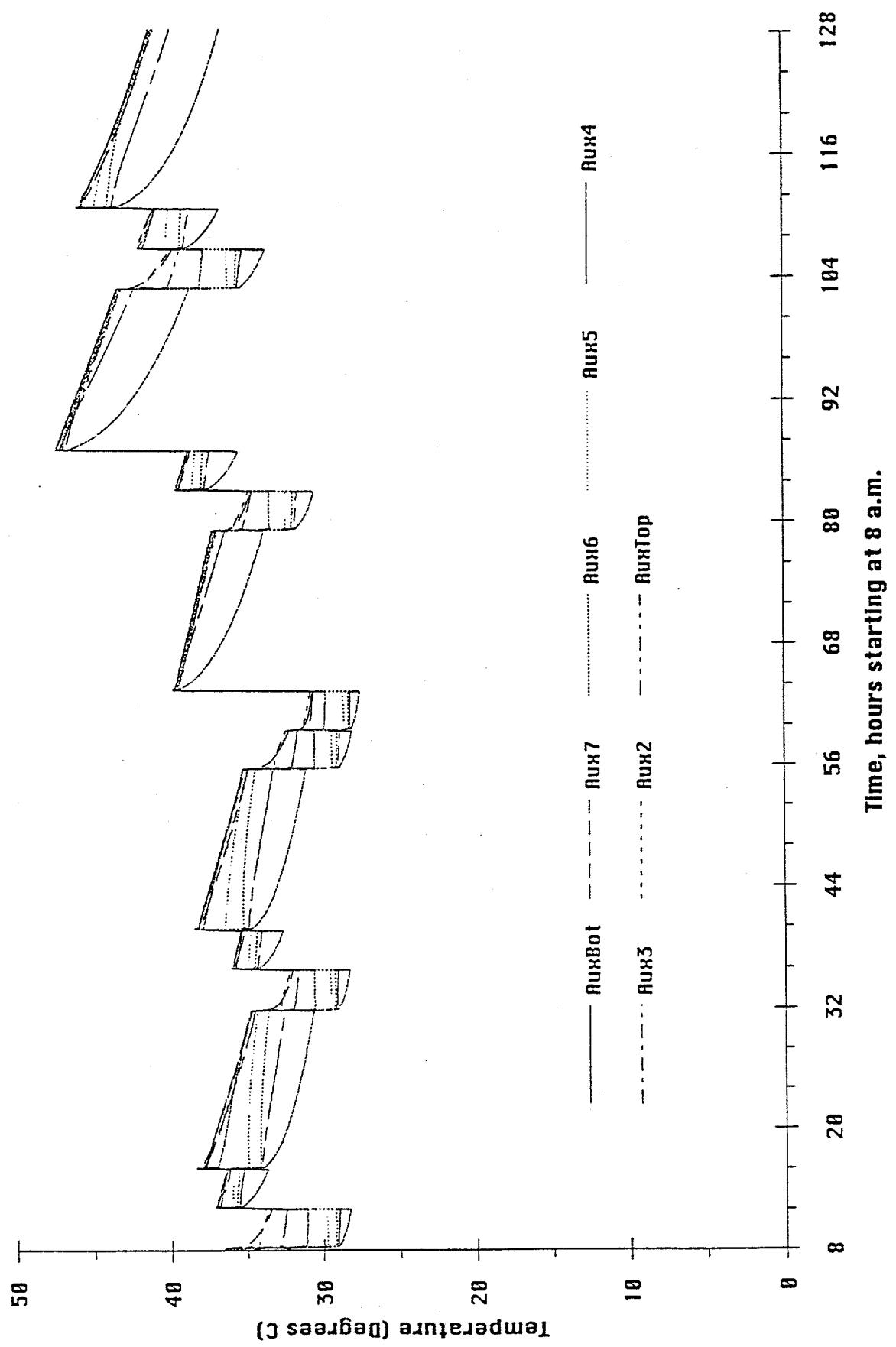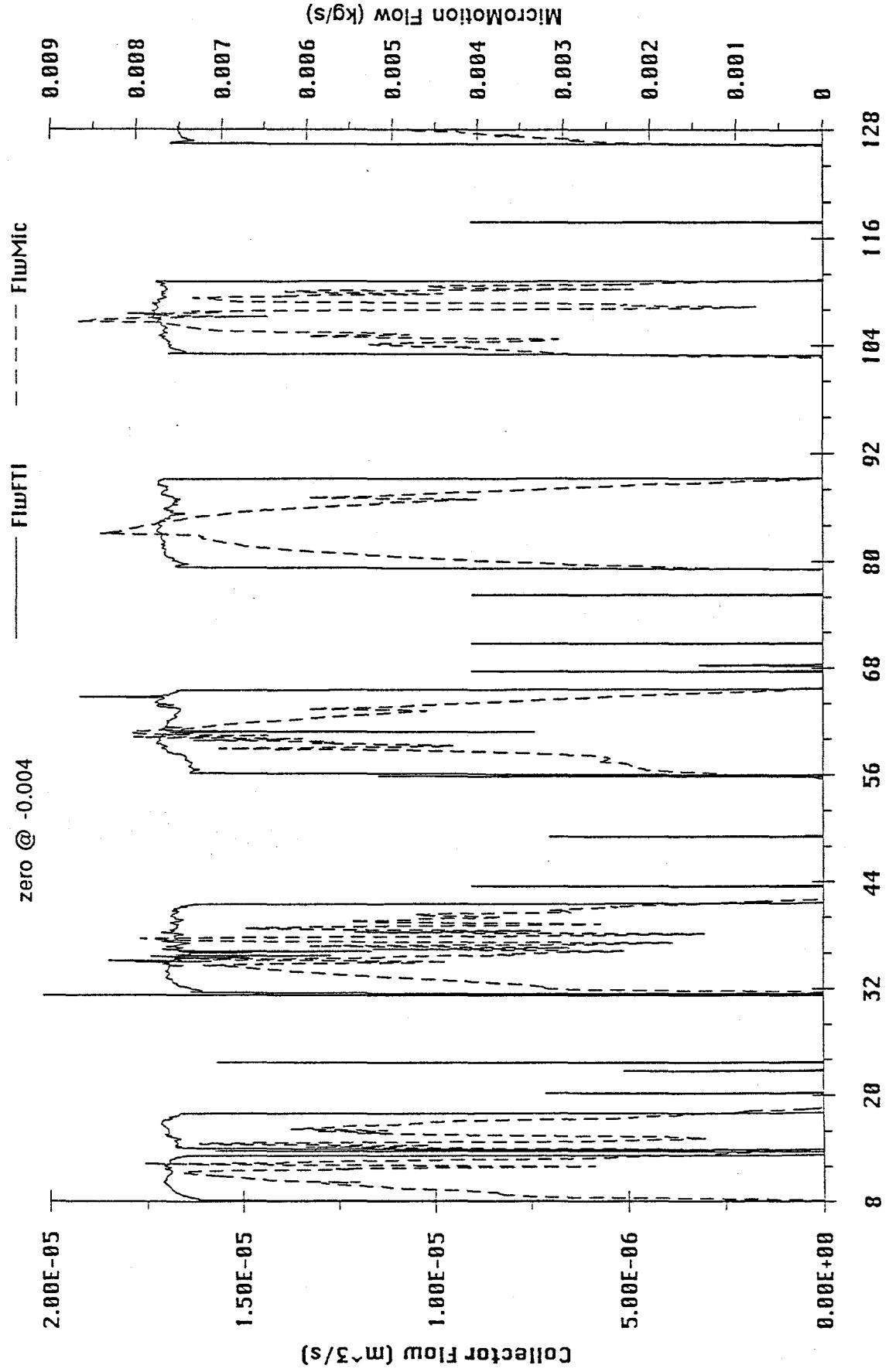
Thermodynamics Pump Pressure Drop and Power
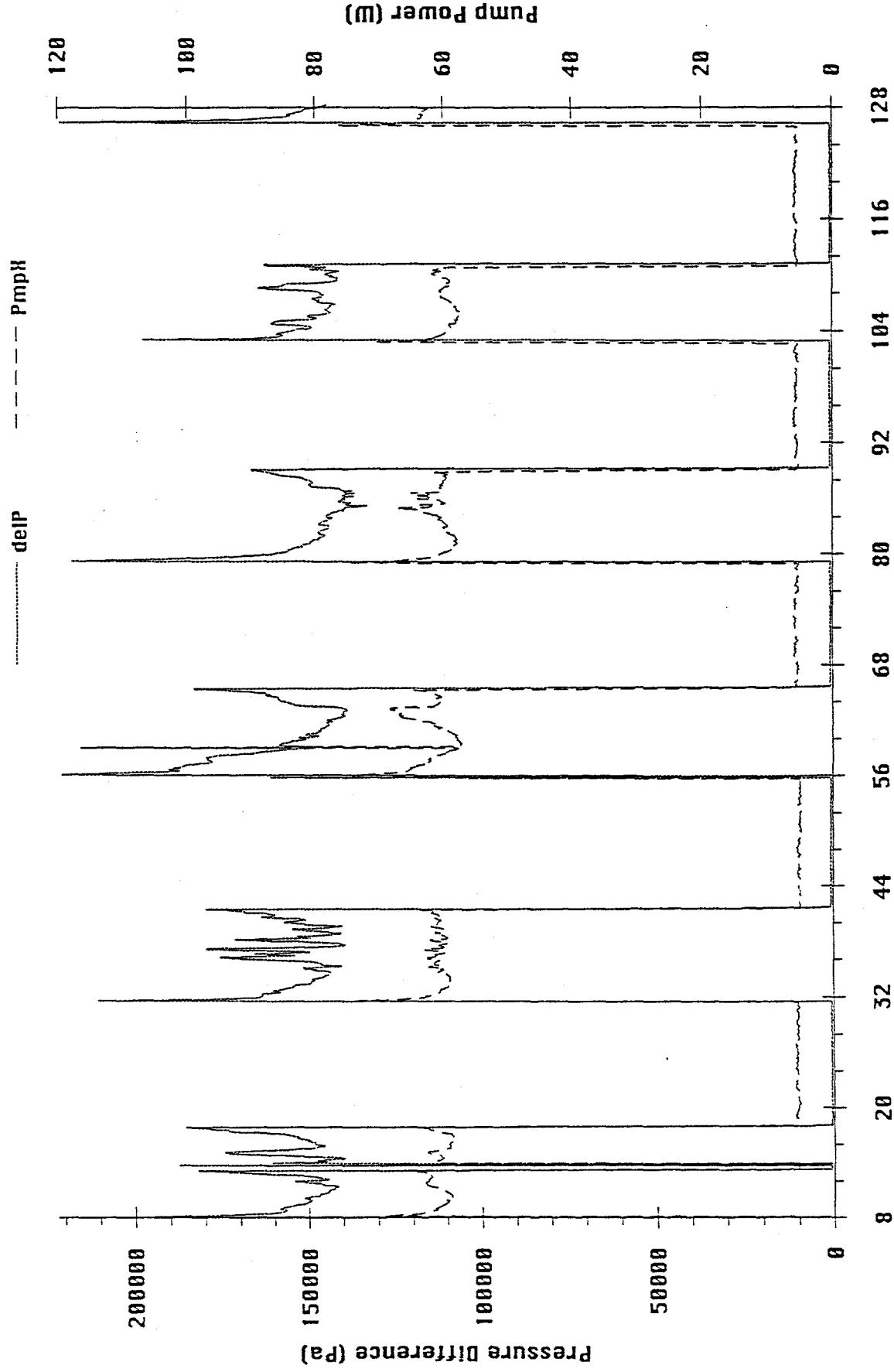
MANAGEMENT AND COORDINATION
OF COLORADO STATE/DOE PROGRAM

Director Doug Hittle participated in conference calls regarding various aspects of the SEAL/DOE research.

Coordination of research activities continued on the four technical research tasks under the DOE grant, and accounts were maintained and updated. Financial and technical reports were submitted as required.