MULTI-SOURCE HYDRONIC HEAT PUMP SYSTEM PERFORMANCE TEST BED

MILTON MECKLER, P.E.
President
The Meckler Group
Encino, California

ABSTRACT

An extensive independent evaluation recently was completed of the Multi-Source Hydronic Heat Pump (MSHHP) system, a proprietary heating, ventilating and air conditioning (HVAC) system developed by Meckler Systems Group. The MSHHP tests were conducted on a unique test bed designed and constructed by National Technical Systems (NTS) through a research and development grant program funded by Southern California Edison Company. This paper outlines testing methods and results, including evaluations of peak power and energy savings allowed by the innovative system. The main difference between the MSHHP and a conventional HVAC system is use of a chilled water "diversity" cooling loop interconnecting air to water coils (located at each water source heat pump unit) with a central chilled water storage tank. The MSHHP system uses significantly less energy than a conventional HVAC system, and lowers peak demand by shifting required electrical energy consumption to lower-cost, off-peak and mid-peak rates. Lower heat pump capacities are a main feature of the MSHHP. This is accomplished by pre-cooling return air from the zone space, a process that also allows the heat pump to operate at a higher Coefficient of Performance (COP), thereby contributing to further energy savings.

TESTING-GENERAL

The tests provided actual MSHHP system operating experience and valuable performance data necessary for design. One zone from a representative nominal 30,000 ft three-story multi-zone building was simulated by connecting full-size HVAC components in their actual configuration. Some of the diversity inherent in actual multi-zone commercial office buildings also was simulated by sizing the chilled water storage tank to accommodate a proportionate number of additional (i.e. interactive) commercial office temperature control zones.

TEST BED

A novel test bed capable of accommodating various HVAC systems was constructed at the NTS facility at Saugus, California (see Figure 1). The hardware design of the MSHHP system integrated within it included the following:

- -Water-to-air heat pump
- -Upstream (pre-cooling) air water coil
- -Downstream (re-cooling) air water coil
- -Chilled water storage tank
- -Associated piping, pumps, controls and other apparatus

The test bed included calibrated air and water flow systems, electrical power, and instrumentation to evaluate the MSHHP system for energy savings when loaded by prescribed latent heat, sensible heat and chilled air sources.

SHIFT TO MID- AND OFF-PEAK COOLING AND THERMAL STORAGE

Significant electricity rate increases are prompting a closer look at methods of reducing peak power demands. These demands, created partly by the high demand for air conditioning in the summer, cause electrical utilities to charge higher rates during these periods. This is because full use of all the utilities' generating equipment, including its less efficient spinning reserve, is required. These extra costs are passed on to customers in anticipation of the higher cost of providing additional new peak power generation capacity.

Most conventional commercial HVAC systems consume most of their energy when electric rates are at their highest. Therefore, it is in the utility's and the customer's best interest to reduce building air conditioning peak demand. A further benefit of installing the MSHHP system is in significantly lower first costs, which result from smaller HVAC equipment and associated electrical power distribution costs. High peak (thermal) demands of conventional HVAC systems present a problem because installed HVAC equipment must be oversized, often only for a short duration. With reduced cycling of unit compressors, lower overall energy consumption results due to significantly improved apparatus efficiency.

The MSHHP system allows for major daily peak (electrical) load reductions and significantly lower energy consumption (KWH) during summer conditions by allowing installation of a smaller water-source heat pump (WSHP) unit. This WSHP still meets peak building load by producing and storing chilled water during mid- and off-peak daily periods through sequenced operation of the downstream (re-cooling water coil, and by circulating this stored chilled water through the upstream (pre-cooling) coil to pre-cool air entering the WSHP during its subsequent zone peak cooling demand period.

SYSTEM DESIGN

In an effort to achieve overall system simplicity, the chilled water storage tank initially was designed for one-directional flow. It also included baffles to stratify tank temperatures. A control scheme was developed using a simulated variable air volume (VAV) system employing a ducted thermostatically-controlled air bypass damper control during off-peak cooling periods. In this way, improved heat transfer rates result in a lower re-cooling coil exit water temperature.

Dual hourly (zone) thermal loads were imposed on the MSHHP system by means of a separate environment-producing air system programmed to simulate actual imposed zone space loads. Associated electrical MSHHP system power needs were measured and compared with conventionally sized WSHP systems tested in the same test bed under identical load conditions.

Time-varying chilled storage tank performance using alternate regimes was monitored along with other tests to evaluate MSHHP's transient response and component performance.

TEST SET-UP

The MSHHP system was evaluated in a full-scale test bed to:

- Avoid scaling errors.
- Study system dynamics.
- Avoid uncomfortable swings in discharge temperature.
- Obtain component design data to achieve specified electrical power savings over comparable conventional HVAC systems.

The test bed was supplied with air which varied as a function of hour of the summer (and winter) design day in response to prevailing latent and sensible heat loads.

The following data were monitored for the MSHHP energy balance and system performance evaluation:

- Temperature differences and air flow across the WSHP.
- Temperature differences and water flow across the two air coils (pre-cooling and re-cooling) adjacent to the WSHP.
- Temperature difference and water flow across the desuperheater exchanger capable of producing (or supplementing) net building domestic hot water demands.
- Electrical energy to the test WSHP producing chilled water for charging the chilled water storage tank.

Temperatures along the length of the 3,200-gallon storage tank also were monitored to show the extent of mixing and spreading of return water into the tank in three (3) separate configurations:

- With baffling; one-directional flow
- With baffling; two-directional flow
- Without baffling; two-directional flow

The two-directional flow was undertaken to evaluate potentials for, and net system benefits resulting from, separating the chilled tank into two zones - one warmer at approximately 68°F average temperature and another cooler at approximately 61°F average temperature.

INSTRUMENTATION

Instrumentation included:

- Thermocouples (air, water)
- Pressure drop (air)
- Flow rate (air, water)
- Energy (electrical)
- Recorders (manual at sensor and panel; automatic at panel)

Temperature accuracy was measured at +/- 1°F using J-type thermocouples. Temperature differences across the hydronic coils were measured at +/- 0.1°F with four-junction J-type thermopiles.

Pressure drops were found to be of the magnitude of about 0.5 inches (water gauge), and were displayed on inclined Dwyer manometers. Air flow within the duct work was measured by recording the manometer pressure difference between the nozzle throat and the stagnation region upstream of the nozzle. Calibration of the manometer was accomplished by means of a heater and imposed temperature difference.

Water flow was measured by means of a Ryan Herco Paddlewheel flowsensor. Paddlewheel sensor (Model 5931-511), pipe fittings PVC 5911-007 (0-18 gpm) and PVC 5911-010 (0-30 gpm) were used. The initial sensor generated its own electric signal. With an error of less than 1 percent (i.e. within a rather wide range), the signal can travel approximately 200 ft to the data logger without need for amplification.

Electrical energy to the heaters, water pumps, WSHP supply fan and compressor was measured by Ducan, Class 100 or 200, 208 VAC single phase watt/hour meter and Class 100, 480 VAC three phase watt/hour meter.

HYDRONIC HEAT PUMP

The Carrier Model 50HQ-48-3, 4-ton WSHP was used to condition the air. Although it typically employs R-22 as its refrigerant, for the case of the MSHHP test, the WSHP was charged with R-502 (to reduce its net cooling capacity to approximately 3 tons). It also was used for the baseline configuration employing the conventionally sized 4-ton capacity (R-22) WSHP unit.

DESUPERHEATER

The desuperheater accepts the hot pressurized (300 psig) Freon from the WSHP passing it along to the reversing valve through a heat exchanger. For the purposes of the test bed the heat exchanger was used to preheat make up (i.e. city) water at 66°F, heating it to 120°F for use with the building domestic hot water (DHW) system, for example.

The desuperheater was selected with a 1-ton heat capacity and at 120°F, 23.7 GPH can be accommodated. Tests were performed to determine the maximum steady state water flow rate for both the 110°F and 130°F outlet water temperature case and with the WSHP operating at full load.

TEST RESULTS

Test results were excellent, with the MSHHP system demonstrating a 23 percent reduction in peak electrical capacity demand versus a conventional WSHP system operating at the same cooling load.

The storage tank worked well in all configurations tested, with the two-direction baffled tank exhibiting excellent separation of warmer and cooler stored water. Tank sizing appeared to match specified loads rather well, and contributed significantly to providing the necessary MSHHP system supplementary peak cooling capacity.

Pre-cooling and re-cooling air coils worked rather effectively, transferring up to 1.5 tons of net cooling capacity to the return (pre-cooling or upstream) air coil. Tests clearly showed the opportunity to reduce the re-cooling coil face area to about one-half the original specified area, further reducing construction cost and space needs, without significantly affecting MSHHP system performance.

The test bed MSHHPS control system was implemented completely for testing because design needs changed as system behavior was clarified. However, data obtained will be used to validate actual proposed MSHHP control strategies.

SUMMARY

The tests conducted at the NTS facility in Saugus demonstrated the MSHHP system:

- Reduced peak electrical demand for equivalent cooling by 23 percent, as compared to the conventional baseline WSHP system.
- Effectively stored chilled water in the fully-instrumented 3,200-gallon tank provided.
- Transferred up to 1.5 tons of energy through the WSHP pre-cooling and re-cooling coils.
- Exhibited smooth transient behavior on programmed state change.
- Worked well with baffled and non-baffled chilled water storage tanks, and with either one- or two-directional flow.
- Produced DHW in the desuperheater, while only losing 13 to 20 percent of its heat transfer capacity.
- Resulted in a higher overall apparatus EER than conventional cycle compressors at comparable loads. This was accomplished by a one--directional flow fully-baffled 3,200- gallon chilled water storage tank, inner-connected to re- and pre-cooling coils, which respectively charge or withdraw stored electrical energy in the form of stored chilled water. This results in a 40 percent savings on installed MSHHP equipment (and associated elecrical) capital costs.

The major MSHHP system control problems identified were in the test operator control of simulated temperatures, which resulted in some over cooling of the chilled water storage tank. Also, the thermostatic zone bypass duct damper (used when charging the chilled storage tank) was not explicitly included, but was effected by allowing the mixing chamber temperature to drop during part-load conditions. This simulated proper operation of the zone bypass duct damper, but did not test the actual hardware now proposed for actual implementation.

HOURLY DEMAND

The hourly demands for the estimated summer design load on day 233 is graphically displayed in Figure 2. The loads were determined for consecutive summer days. The average hourly power needs for both MSHHP and conventional WSHP systems were compared. It should be appreciated that the higher power needed by the MSHHP system during the off-peak periods actually was used to refrigerate chilled water tank contents and store it in the 3,200-gallon chilled water storage tank for subsequent use.

PEAK ELECTRICAL DEMAND COMPARISON

Peak electrical demand for the MSHHP system was found to be 23 percent less than for the conventional baseline WSHP system case (after the R-22 refrigerant conversion factor was applied).

The peak unit electrical demand for the conventional WSHP system test bed on summer design days 233 and 234 was found to be 6.0 kw, while the peak demand for the MSHHP was found to be 4.6 kw. This 1.4 kw difference represents 23 percent of the conventional WSHP system's demand of 6.0 kw. On day 233, the conventional WSHP system's peak hour was the same as for the MSHHP - hour 15:00 (between 2 and 3 p.m.) On day 234, the conventional WSHP system peaked at hour 16:00, with a slightly lower peak at hour 12:00 (5.9 kw). The MSHHP peak for summer day 234 was at hour 18:00.

CHILLED WATER STORAGE

Tank Operations

The MSHHP system worked effectively with the 3,200-gallon chilled storage tank operating in each of the three following configurations: 1) baffles with one-directional flow; 2) baffles with two-directional flow and 3) two-directional flow without baffles. Tank temperature data for each of the three two-day tests are given in Tables 1 and 2.

Data in these tables suggest larger differences than actually resulted from changes in the designated operating mode. The most noticeable difference is that very low temperatures were obtained in the baffled one-directional flow test run. This difference is attributed to approximately twice the scheduled amount of required supplementary cooling delivered to the chilled storage tank during first (day 233) run than actually needed for thermal balance. In effect, this demonstrates the ability of the MSHHP system to reach lower than earlier designated test bed design temperatures.

Supplementary loads for the two-directional flow test case were equal to approximately 2.74 additional WSHP zoned systems, each provided with one-ton nominal capacity pre-cooling and re-cooling air coils of the type described earlier. Chilled water tank temperatures for these latter tests were found to rise approximately 6 to 8°F during the worst (i.e. summer days 233 and 234) test days. This suggests that the chilled water tank size, 3,200 gallons, was appropriate for the preselected nominal 3.74 ton zone load.

One-Directional Flow

Besides lower temperatures resulting mostly on the second consecutive (i.e. 234 day), this test run clearly demonstrated the effects of a "slug" warmer water as it moves through the baffled tank configuration (see Figure 3). On day 233, between hours 11:00 and 13:00, water about 68°F flowed into the tank inlet.

Refer to the time plot in Figure 3 of the third and sixth days to see this slug of warm fluid passing through points between 13 and 15, and between 15 and 17 hours, respectively. Somewhat cooled, the water slug finally exits the storage tank between hours 17:00 and 19:00. This slug maintained a fair degree of cohesiveness, yet took about six hours to traverse the storage tank. This is at a rate of about 9 gpm, which appears quite reasonable.

Two-Directional Flow

One of the major performance criterion for both one-directional and two-directional test cases relates to the condition of the chilled water storage tank on the second day of each set of two-day tests (day 234). Testing was designed with sequential test days to observe the effect of 233 day's operation and establish the efficiency of the proposed methodology by demonstrating that the chilled water tank could maintain design conditions during the second day as well.

The baffled chilled water storage tank was found to be effective at separating the relatively warmer and cooler waters, while no temperature separation or stratification was evident for the case of the unbaffled tank. Temperatures at both ends of the tank are almost equal at any point as can be seen from reference to Figure 3.

Baffles Versus No Baffles

The net rate at which heat can be transferred in or out of the chilled water storage tank is affected by placement of baffles. This assumes, of course, that the high and low temperature limits of the chilled water tank are fixed, allowing the total energy stored in the chilled water tank to remain the same.

If, however, the lower temperature water (61°F) is available when the MSHHP system demands pre-cooled return air, more heat can be absorbed by it than if the water were at the high limit (68°F) temperature.

The rate difference has definite design implications. One can assume two cases. In the first, a fixed heat transfer rate through the pre-cooling and re-cooling coil, and a fixed flow rate. Obviously, it is also desirable to minimize coil size and cost.

To transfer the most energy through the return air pre-cooling coil, a large delta "T" between entering water and air is required. Therefore, temperature stratification through the use of baffles in the tank is valuable. Without baffling to maintain lower temperatures, the size of the chilled water storage tank would need to be much larger.

Estimates from Figure 4 suggest that about 80 percent separation can occur, or that 80 percent of the chilled water tank can be assumed to be "useful" due to temperature stratification.

The advantage of baffling is that smaller re- and pre-cooling air coils can be used. If return air is 78°F, 61°F water has a 17°F temperature difference, while 68°F water has only a 10°F difference. For the same rated heat transfer, it could use a coil of about 60 percent the effectiveness. Because of the non-linear relationship between size and effectiveness, the respective pre- and re-cooling coils could be confidently designed with one-half the number of heat transfer units.

If the MSHHP system was designed to use a baffled two-directional flow chilled water storage tank, the tank could be sized based on 80 percent effective temperature stratification, and economies in reduced heat exchange areas would result. If a non-baffled (or uniformly mixed) chilled water storage tank is used, either the tank volume or coil area should be increased.

TRANSIENT REPONSE

Potential transient effects may result from a rapid change of supply air temperature to the maintained temperature control zone, particularly when flow to either the re- or pre-cooling coils is interrupted.

Since the MSHHP heat pump runs continuously, transients are not possible. Return air temperature changes due to operation of the pre-cooling coil.

Only a significant thermal response could effect such a change. And, even then it is more likely the thermal time constant of the WSHP would further dampen the transient. If the pre-cooling air coil is inoperative, and the zone temperature is brought down to 65°F by the WSHP alone, the re-cooling coil would most likely be activated.

The WSHP inlet condition does not change immediately, so its output to the re-cooling coil is constant. The re-cooling coil will warm the air as it cools the water within it, so the supply air to the zone will rise, but only as fast as the re-cooling coil can respond.

A worst case response was investigated and found to be relatively mild. The thermal response of air passing through the re-cooling air coil when water circulation through the coil was started was found to be smooth and reasonably slow.

The same WSHP provided with refrigerant R-502 (in lieu of R-22) has a larger heating capacity because all of the (less efficient) compressor work is included in the condenser heat output.

PRE- AND RE-COOLING COILS

Potentials for reducing the size of the re-cooling coil were evaluated. Face area reductions were made by inserting a galvanized steel sheet in front of the coil air-side face to reduce its (flow) effective area. Water flows were increased proportionally due to the reduction in area resulting in improved heat transfer.

Air and water temperatures were measured, (as was air flow pressure drop) across all coils. The air flow rate also was checked to determine if the earlier referenced flow restriction at the coil affected overall air circulation. The face area of the coil was reduced to 3/4, and then 1/2, the original area. Air flow stayed constant, satisfying the original assumptions.

Pressure drop and heat transfer effectiveness also were measured for all three sizes. The results generally validated the theoretical analysis.

Variations in the effectiveness data probably stem from air flow variations around the thermopile measuring the air. Otherwise, effectiveness for the 3/4 area would be between 1 and 1/2 area. The overall result is that effectiveness only dropped 20 percent for a 50 percent area reduction.

HARDWARE PERFORMANCE

The hardware investigated included:

- Schedule 80 PVC piping
- Pilot-operated solenoid valves from Ryan Herco
- Three-way diverter valves
- Desuperheater
- Pre-cooling and re-cooling water coils

Most of the equipment performed within reasonable bounds of design expectations. The major exception was the Ryan-Herco solenoid valves. These did not operate without significant line pressure to open the valve. They would not open at all under the static head of 6 ft deep water tank. Under line pressure, however, they operated satisfactorily.

Use of PVC piping proved to be reasonably successful. The Schedule 80 material did not cause condensation on its surfaces for chilled water temperatures above 48°F. Condensation did, however, result in water temperatures below 48°F. Since specific air humidity data was not taken, the dewpoint at which condensation actually occurred is not known.

DESCRIPTION OF MSHHP TEST BED

The test bed consisted of five sections: load, mixing chamber, air circuit, water circuit and instrumentation. The overall floor plan is depicted in Figure 5.

The mixing chamber and ducting occupy the indoor floor space. The (external) chiller and chilled water storage tank were located outside the test building. Most of the instrumentation was placed next to the WSHP component under study, but thermocouples were routed back to the instrumentation panel.

The MSHHP system consisted of:

- "Carrier" water-to-air heat pump with a desuperheater 3-ton cooling capacity.
- Pre-cooling and re-cooling water-to-air heat exchange coils connected to a chilled water storage tank (3,200 gallon)
- Control circuit

The WSHP cooled water from the chilled water storage tank whenever the water was circulated through the re-cooling coil and the colder air was not required in the zone, for example in an office building during the early morning hours.

As the solar gains entered the test case (simulated) office building later in the morning, and some further net cooling was required, the WSHP conditioned the office with excess capacity delivered to chilled water storage tank. Whenever the undersized WSHP attempted to condition office areas during the hottest (peak or design) outdoor conditions, water must be pumped (i.e. reclaimed) from the chilled storage tank flowing through the pre-cooling coil to lower the air temperature delivered to the WSHP. This allows it to satisfy peak zone loads.

The WSHP also required warmer water circulating through the pre-cooling coil when office areas experienced hot net demands for heating.

MIXING CHAMBER

The mixing chamber was designed to assure that the conditioned air from the load section was completely mixed with the conditioned air exiting from the MSHHP system heat pump.

Approximately two hundred concrete blocks placed within the chamber improved this mixing and also supplied thermal mass. This configuration was selected to simulate approximately 400 sq ft of typical office space filled with furniture.

Further mixing was achieved by reducing air velocities from 260 ft/minute in the 12 ft^2 cross-section of the load section to 51 ft/minute in the 64 ft^2 cross-section of the mixing chamber.

Adequate mixing was verified by placing dry bulb thermocouples into the dead air regions of opposite corners of the chamber, and in the active regions of the entrances of the two return ducts. When all thermocouple readings were approximately the same temperature, mixing was assumed to be complete.

LOAD SECTION

The load section consisted of a 48 X 37 $inch^2$ cross-section duct, passing 3,200 ft^3 of air/minute. This corresponded to a 12.3 ft^2 cross-section and a 260 ft/minute air velocity. Ducting was insulated and leak-checked to assure little energy was lost.

The environmental coil was designed to produce up to 8 tons of cooling to the air flowing past it at 260 ft/minute air velocity, and 3,200 CFM volumetric flow rate. The heat exchanger was four rows and had 10 fins/inch. At 3,200 CFM, it had a pressure drop of 0.14 inches of water.

The sensible heat history (simulating summer conditions) was applied to the mixing chamber during the summer by a Delta Flo, Model EH-15D-324, 15 kilowatt, three-phase heater controlled by a variac (variable transformer). Each test hour, the setting was adjusted to follow the prescribed energy input history.

The latent heat history (simulating summer conditions) applied to the mixing chamber during the summer was supplied by a steam supply system. It had a large number of jets across the entire duct cross-section. Air flow in the duct across the jets assured mixing of water vapor with air in the duct.

The steam flow was measured by a standard orfice plate section. Flow was adjusted each hour to match the prescribed history. A 3/4 hp motor with an 18-inch diameter fan recirculated 3,200 CFM of air through the load section.

The nozzle in the load section was used to calibrate air flow. The nozzle had a manometer across it. The calibration of the nozzle (static pressure versus stagnation pressure manometer) was achieved by using the coil and thermocouples, located upstream and downstream of the coil. The coil and thermocouples established the volumetric rate of air flow.

The manometer was marked for its pressure difference at this volumetric rate of air flow. On all test runs, the manometer was checked to determine if initial levels were maintained.

AIR FLOW CALIBRATIONS

The volumetric flow rate of 1,600 ft^3/minute was calibrated by using a coil and temperature difference across that coil to obtain air flow in the same manner as the environmental coil in the load section. This is explained mathematically by:

$$Q = \frac{q}{\rho\, C_p \Delta T} \quad (\mathrm{CFM})^{*}$$

The result was the volumetric rate of flow in ft^3/minute. This assumed the ducting was air-tight, had been operating for several minutes and had attained steady-state temperature.

WATER CIRCUIT

The water circuit was composed of polyvinyl chloride tubing (PVC), 3/4-inch diameter, schedule 80. These tubes could handle the expected four to 10 gallons/minute (gpm) water flow. The schedule 80 thick wall tubing not only handled pressure, but also supplied sufficient insulation to keep condensation from occurng on pipes down to 48 $^\circ$F water temperature, at mid-70s moderate relative humidity (RH) surrounding air.

The Teel ball bearing centrifugal pump (Model 1P796, 1/3 hp) was used in the water circuit. It could sustain 12 gpm at 40 ft in the pump head at 3,450 RPM, and supply 1 to 95 gpm.

Test bed valves were 3/4-inch and 1-inch bronze globe, gate and check valves (McMaster-Carr Co.), and the Ryan Herco two-way adjustable solenoid 3/4-inch PVC valves (Models 5638-007, 5638-010 and 5639-007).

The MSHHP three-way valves were Honeywell's three-way diverting valve (Model V5013C). The 3/4-inch valve handled a flow of less than 10 gpm, the 1-inch a flow less than 25 gpm. The valve linkage (Model 0618A1032) was the same for all valves. The actuator was either normally closed (Model M845A-1027) or normally opened (Model M845E-1007).

Water was routed via several circuits to:

(MSHHP SYSTEM:)
- Cool/heat the heat pump condensor/evaporator coil
- Cool/heat the coil upstream (pre-cooling) of heat pump
- Remove cool water from the coil downstream (re-cooling) of heat pump
- Remove heat from compressed superheated Freon gas

(TEST SYSTEM:)
- Heat/cool water in the chilled water tank
- Chill water in the winter environment coil

The water chiller system comprised a 500-gallon 10-ft tall steel water tank, an NTS 3-ton circulating mechanical water chiller and a supplementary liquid nitrogen delivery set up for bubbling boiling liquid nitrogen into the bottom of the tank. This cold water reservoir was connected directly to the environmental coil, via a heat exchanger to the 3,200-gallon chilled water storage tank. It included an inserted coil to cool city water before its delivery to the tank.

CONTROL SYSTEM

The MSHHP system in the simplest configuration tested had four basic sub-systems that need to be controlled. These were:

Sub-System	Control Functions
Heat pump	on-heat/on-cool/off
Pre-cooling coil	on-hot/on-cold/off
Re-cooling coil	on/off (cold only)
Air flow dampers	20 to 100 percent, continuous

With these control functions, the system could ventilate and temperature control the zone from peak heating to peak cooling capacity.

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* where: q = heat flow from air to coil (BTU/min)
ρ = density of air (lbm/ft^3)
C_p = specific heat of air (BTU/lb$^\circ$F)
ΔT = temperature differences across coil ($^\circ$F)

Hour	Days 233 and 234: Warm 1st Bay	2nd Bay	3rd Bay	4th Bay	5th Bay	6th Bay	7th Bay	8th Bay	Cool 9th Bay	
6	67.2	67.2	67.2	67.2	67.4	67.2	67.2	67.3		
7	66.8	67.0	67.0	67.2	67.2	67.0	65.6	57.8		
8	63.3	65.0	66.5	67.2	67.5	67.1	66.6	62.2		
9	65.4	65.1	65.4	66.7	67.3	67.1	67.1	66.0	58.3	
10	72.3	68.0	72.0	65.9	66.5	66.8	67.0	66.7	64.1	
11	73.1	73.2	73.0	72.7	69.8	66.5	66.7	66.6	66.5	
12	71.3	71.8	72.6	72.9	72.8	72.5	70.6	67.3	66.7	
13	73.3	72.6	71.8	72.0	72.6	72.6	72.7	72.4	66.7	
14	75.3	75.5	75.2	74.1	72.6	72.2	72.3	72.6	71.7	Day
15	75.2	75.3	75.0	74.0	72.3	72.0	72.2	72.4	72.5	233
16	74.7	75.3	75.1	74.0	72.4	72.0	72.2	72.5	71.6	
17	74.7	74.6	73.7	72.2	72.0	72.1	72.2	66.7	64.9	
18	74.6	73.5	72.2	71.9	72.0	72.0	69	64.7	64.6	
19	74.2	72.2	71.9	71.8	71.8	68.5	64.9	61.5	61.4	
20	74.0	71.9	71.7	71.6	69.1	64.9	62.2	61.5	61.4	
21	73.6	71.8	71.3	69.0	65.2	62.2	61.8	62.1	61.9	
22	73.5	71.6	71.0	68.3	64.8	61.8	61.7	62.2	62.4	
6	70.5	70.6	70.2	68.2	66.1	63.7	61.7	62.0	63.8	
7	69.9	69.0	67.9	65.8	63.4	61.7	61.7	59.0	60.2	
8	69.0	66.3	64.3	62.5	61.8	61.4	60.2	60.4	61.2	
9	68.5	63.1	62.0	61.7	61.0	60.4	60.7	60.7	61.2	
10	68.1	63.1	61.9	62.1	61.1	60.5	60.8	60.7	61.4	
11	68.9	67.8	63.2	62.6	62.6	60.7	60.6	60.5	61.0	
12	68.8	69.0	67.7	63.9	63.3	62.4	61.2	60.5	60.7	
13	68.7	68.8	67.5	63.7	63.2	62.1	60.9	60.9	64.5	
14	68.6	67.9	67.9	68.1	66.7	63.6	62.9	61.7	61.2	Day
15	70.7	70.6	69.2	67.9	68.0	67.5	65.3	63.2	62.1	234
16	69.6	70.1	70.4	70.1	68.5	67.9	67.8	66.2	63.8	
17	69.6	69.6	70.1	70.1	69.2	68.1	67.9	67.1	64.6	
18	69.6	69.6	70.1	70.1	69.2	68.2	68.0	67.2	64.8	
19	69.4	69.7	70.0	69.9	68.8	68.0	67.9	66.4	59.3	
20	69.8	69.9	69.6	68.6	68.2	67.7	66.3	60.7	61.4	
21	69.7	69.0	68.2	67.9	67.3	64.1	61.4	62.0	62.4	
22	69.5	67.4	67.2	65.3	61.9	61.8	61.9	61.9	61.6	

Table 1. STRATIFIED TANK TEMPERATURES (ONE DIRECTION FLOW)

Hour	Days 233 and 234: Water Inlet	2nd Bay	3rd Bay	4th Bay	5th Bay	6th Bay	7th Bay	Outlet
6	43.9	60.6	64.5	64.0	63.8	63.9	64.0	63.8
7	49.0	50.0	59.0	63.1	63.8	63.8	63.8	63.8
8	54.8	46.9	53.4	57.3	60.7	62.7	63.6	63.9
9	59.9	53.2	54.1	54.2	56.9	59.7	60.8	63.3
10	62.8	54.6	59.2	54.6	55.0	57.2	59.6	62.1
11	69.7	57.1	63.3	59.7	56.1	55.5	67.1	60.0
12	74.7	60.4	68.7	63.6	60.2	56.6	56.1	58.1
13	68.7	58.2	71.9	68.8	64.4	60.4	56.6	56.8
14	64.0	60.9	72.6	71.0	70.4	66.0	61.4	56.5
15	53.8	47.2*	72.2	72.1	70.8	69.8	65.8	57.9
16	54.9	40.1*	63.1	68.2	70.5	70.9	70.4	61.1
17	56.7	37.7*	57.8	62.5	66.7	69.2	70.2	66.0
18	54.9	37.3*	55.9	58.5	62.1	65.4	68.1	69.6
19	54.9	36.5*	55.4	56.5	58.7	61.4	64.6	67.9
20	54.3	36.4*	55.2	55.6	56.8	58.8	61.3	64.9
21	52.8	31.9*	53.9	55.1	55.8	56.8	58.3	61.2
22	56.5	32.7*	55.4	53.8	53.7	54.4	55.3	56.8
6	51.4	28.9*	54.9	53.8	53.9	54.7	55.1	56.3
7	52.2	25.5*	54.9	55.0	54.7	54.4	54.4	55.0
8	52.7	52.9	53.2	53.7	54.4	54.7	54.7	54.7
9	52.7	52.9	53.0	53.1	53.4	53.8	54.2	54.5
10	53.9	53.1	52.9	53.0	53.2	53.4	53.7	54.1
11	55.0	55.0	53.4	53.1	53.2	53.3	53.5	53.9
12	46.5	61.1	55.2	53.7	53.2	53.2	53.4	53.5
13	56.3	56.6	57.5	55.6	54.1	53.4	53.4	53.6
14	60.2	58.9	56.8	55.6	55.6	54.6	53.9	53.7
15	59.7	61.9	60.4	58.3	57.0	56.3	55.5	54.0
16	59.3	60.7	61.2	60.2	58.3	57.1	56.4	54.5
17	51.6	53.9	57.3	60.0	60.4	59.2	57.6	55.6
18	55.6	53.8	53.0	54.3	56.4	58.3	59.1	57.8
19	54.9	54.9	54.5	53.9	54.4	55.8	57.3	58.5
20	53.8	54.4	54.7	54.7	54.5	54.5	55.3	56.6
21	52.7	53.5	54.2	54.4	54.7	54.7	54.8	56.4
22	50.9	52.0	52.9	53.6	54.2	54.5	54.7	55.0

* Bad thermocouple - disregard data

Table 2. CHILLED WATER STORAGE TANK TEMPERATURES TANK WITH BAFFLES (TWO DIRECTION FLOW)

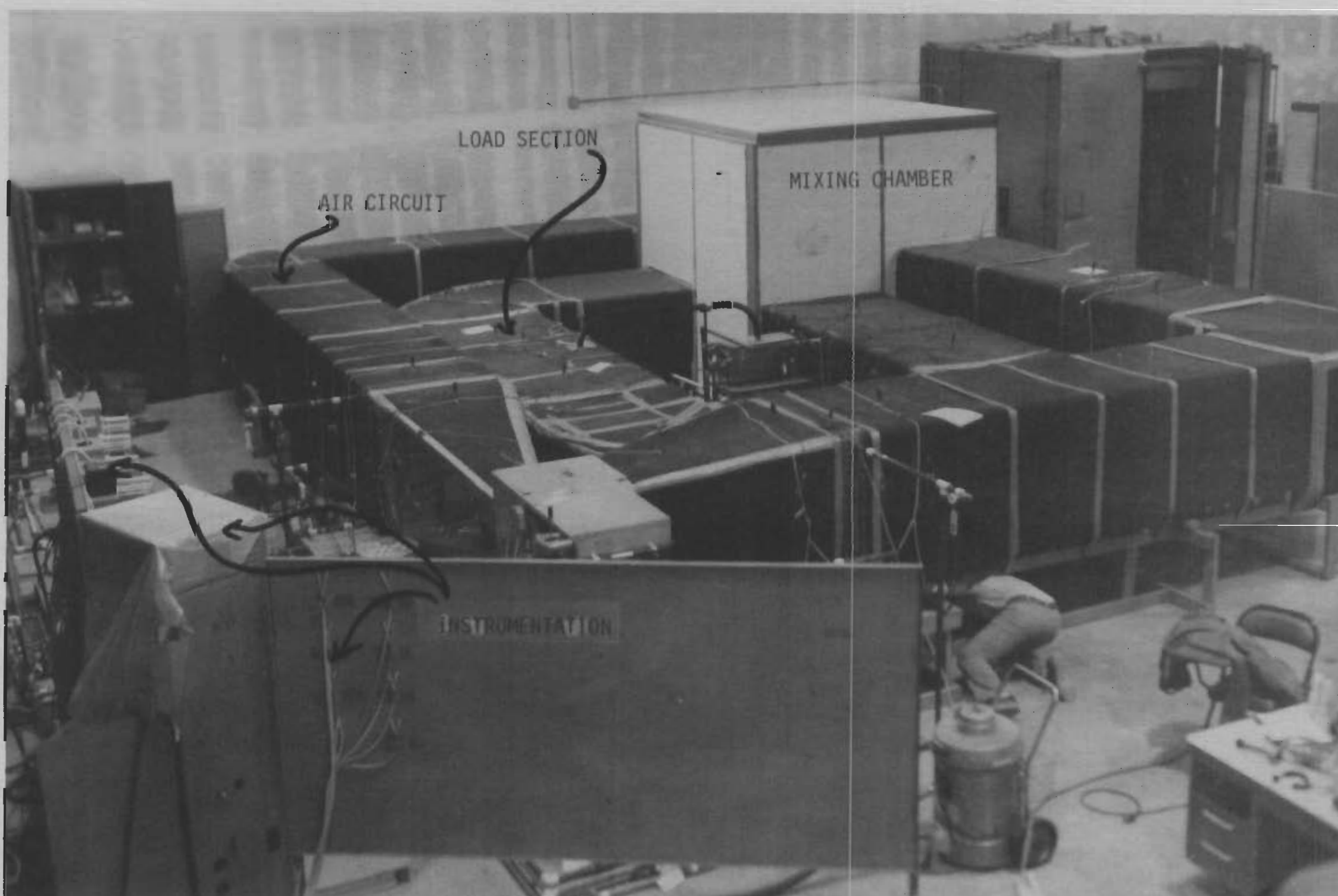

Figure 1. ACTUAL MSHHP TEST BED AT SAUGUS, CALIFORNIA

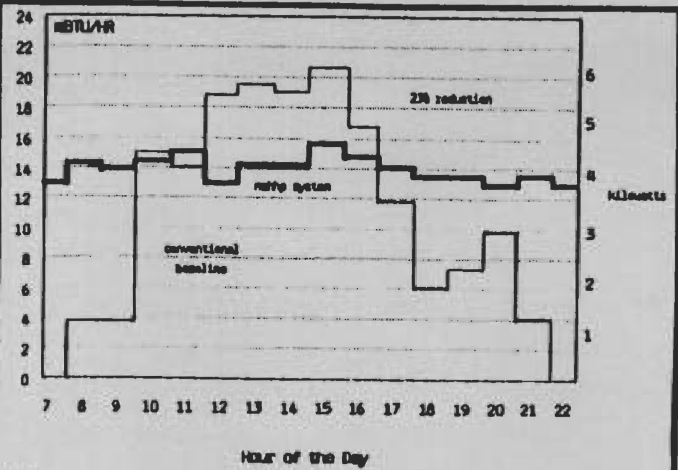

Figure 2. HOURLY POWER DEMAND
MSHHP VS CONVENTIONAL WSHP BASELINE
DAY 233

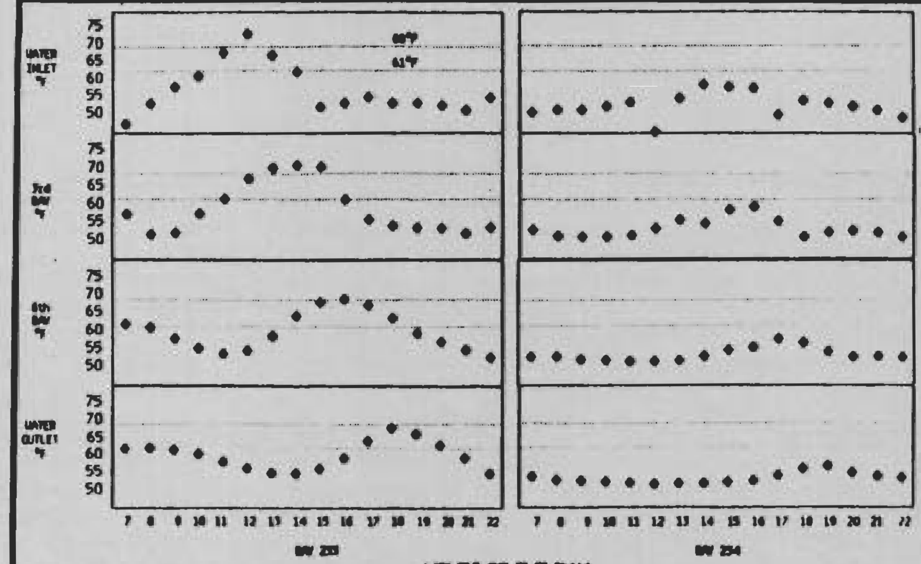

Figure 3. 3200-GALLON CHILLED WATER STORAGE TANK TEMPERATURES AT INLET, 3RD BAY, 6TH BAY AND OUTLET VS HOUR OF THE DAY, BOTH DAYS

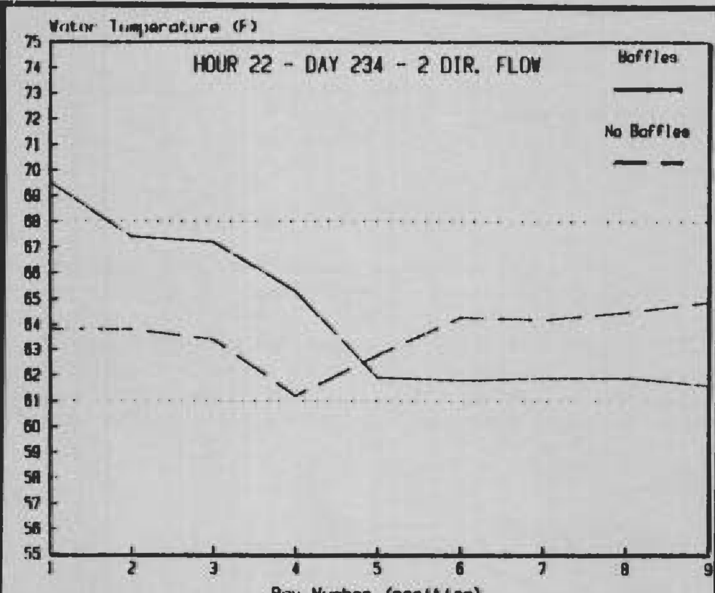

Figure 4. STRATIFICATION IN STORAGE TANK

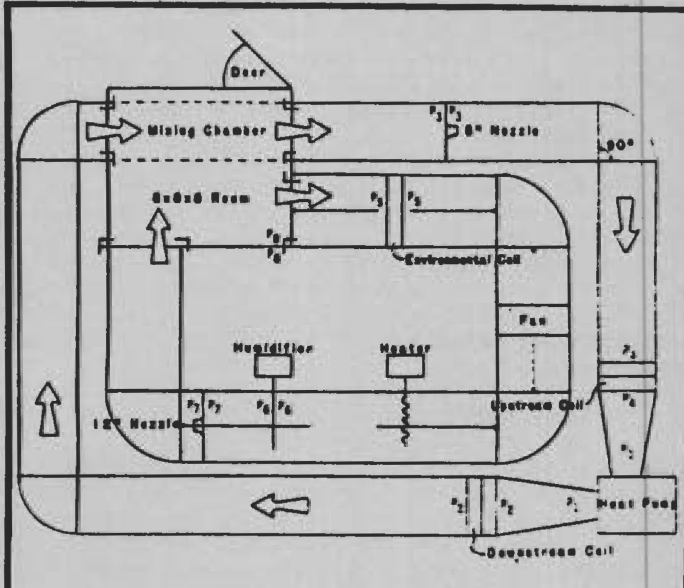

Figure 5. MSHHP TEST FACILITY