THERMAL ANALYSIS OF THE WEST ENGINEERED SAFEGUARDS ROOM OF THE PALISADES NUCLEAR POWER PLANT

BY

WESTINGHOUSE ELECTRIC CORPORATION

FOR

CONSUMERS POWER COMPANY

April 7, 1987

Revision No. 1

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REVISION 1

Consumers Power Company has reviewed the Westinghouse Analysis and concurs with the analytical methods and assumptions. Consumers participated in the reconciliation of differences between the original Westinghouse and Consumers analyses. The assumptions used in this revision of the analysis such as pipe lengths, fluid temperatures, motor horsepowers, etc. were reviewed with Westinghouse and accepted by Consumers.

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ACKNOWLEDGEMENTS:

The effort required to develop input data, perform and verify the analysis, document the work, and prepare this report included contributions from R. E. Schwirian, D. R. Forsyth, K. B. Neubert, R. A. Hluhan, J. M. Cordle, and K. Suprano. Their efforts were sincerely appreciated.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

1.0 PURPOSE

To determine the time history variation of the bulk room air temperature in the Palisades West Engineered Safeguards Room (WESGR) following a postulated design basis LOCA.

2.0 SUMMARY OF RESULTS

The calculated maximum bulk room air temperature within the Palisades WESGR that would occur during the ten days immediately following a design basis LOCA is approximately 135°F. The results indicate that this maximum would be reached within one hour, and would be maintained for approximately six hours.

3.0 GENERAL METHODOLOGY

This analysis was formulated with the intent to demonstrate that an acceptable room environment would be maintained during response to a design basis LOCA with the room cooling unit operating with one fan out-of-service and with 80°F service water temperature.

The mathematical heat transfer model of the WESGR was derived based on a conservative, simplified representation of the room and its contents. In general terms, the model was an extensive energy balance over the room volume. This time history balance includes heat sources, heat sinks, and an accumulation of energy by the air volume. The model accounts for a large number of pipe sections comprising the total piping that would be involved in a response to a LOCA. The time history performance of the cooling unit was accounted for using an appropriate performance factor designed to reflect service water conditions. Constant energy sources like the pump motors, cooling unit fan motor, and room lighting were also accounted for. The room boundaries were excluded from the heat transfer model for simplicity and because this exclusion was conservative.

The worst-case scenario of the WESGR response to a LOCA was specified by CPCO. Based on this specification, and pertinent design data for the room equipment and piping, Westinghouse was able to develop the necessary input data used in the analysis.

Conservatisms were introduced into the WESGR input data, model, and analysis to ensure that the calculated time history bulk room air temperature would represent an upper bound. However, in order to ensure that a realistic upper bound was obtained, arbitrary and unnecessary conservatisms were not included.

4.0 REFINEMENTS TO PREVIOUS ANALYSIS

An assessment was made to identify the differences between the previous Westinghouse and CPCO analyses, and will be presented later in this report. Based on the results of this assessment, it was decided to refine the previous Westinghouse analysis by improving the modeling of some effects and the accuracy of some input data. This refinement is summarized as follows:

(1) Improved description of the cooling unit performance curve of heat removal capability versus the difference between inlet air and service water stream temperatures. This data was used to derive an appropriate cooling unit performance factor.

(2) Improved modeling of the cooling unit overall heat transfer coefficient by reducing the margin between the calculated value and the value applied in the previous analysis, from 0.057 to 0.007. Rev. 1

(3) Eliminated consideration of heat emission into the room due to pump inefficiencies. It was determined that previous accounting of such heat emission was unnecessary since, realistically, the overwhelming majority of the pump heat goes to the fluid being pumped and the remainder goes to the bearings, which are cooled via service water.

(4) The calculations of heat emission from pump motors were adjusted to account for actual motor horsepower during pump operation.

(5) The total amount of piping involved during a WESGR response to a LOCA was increased to include a 3" x 16' section and a 8" x 24' section with surface areas of 15 ft^2 and 54 ft^2 respectively.

(6) The LPSI pump piping was removed along with the LPSI pump itself after the first 2000 seconds of the event.

(7) The two shutdown heat exchangers, previously modeled conservatively as constant heat sources, were modeled as large pipes.

5.0 MODELING ASSUMPTIONS AND CONSIDERATIONS

* The room air volume was assumed to mix perfectly and instantaneously.

* The containment sump water temperature, recycled as containment spray after 2000 seconds, varies according to the results of a 1986 Combustion Engineering analysis (Figure 3). Rev. 1

* The LPSI pump was assumed tripped at 2000 seconds and removed as a heat source. The associated piping, which would be full of water at only 88°F, was removed with the pump and motor at 2000 seconds. Since the air temperature at this time is higher than 88°F, these water filled pipes would actually respond as heat sinks.

* The auxiliary and room sump pumps will not operate during the event.

* One HPSI and two containment spray pumps will operate continuously during the event.

* Only one of two fans on the cooling unit will operate during the event.

* The balance of constant energy sources are the fan motors and the room lights. The heat input from all "constant" energy sources is assumed constant within any given time interval.

* Within a given time interval, the temperature variation of a fluid in any particular pipe section was approximated by a linear function of time.

* Credit was not taken for spatial variations of fluid temperature along the length any particular pipe section.

* The performance of the cooling unit throughout the event was modeled using a predefined performance factor reflecting service water conditions. The value of this performance factor was calculated using the relationship between the cooling unit's heat removal capability with one fan operating and the difference between inlet air and service water stream temperatures.

* There are certain sections of piping that are direct targets of the exit air stream from the cooling unit.

* The room boundaries and components within the room are potential heat sinks, and were not taken credit for in the analysis.

* The calculated heat transfer coefficients for the piping were based on actual velocities measurements made at the site.

* The coupling efficiency of all pump units was assumed to be 97%.

* The two shutdown heat exchangers were modeled as a large pipe with a variation in water temperature with time.

6.0 CONSERVATISMS

* Only one of two cooling unit fans will be operational throughout the event. If both fans were assumed to operate over a portion of the event, the resulting bulk air temperature time history could be significantly lowered.

* All radiant heat emitted within the room is assumed to be absorbed by the room air volume, though most of it would be absorbed by the room boundaries and equipment with a partial, radiative and convective, transmission to room air.

* No credit is taken for heat transfer resistance by pipe walls and the inside thermal boundary layer in the fluid streams. That is, it is assumed that the outside wall temperatures of all pipes are equal to the fluid temperatures within those respective pipes - instead of being some ΔT lower.

* The cooler unit directs air onto certain lengths of piping directly in the path of cooling unit exhaust.

* The fluid temperature in each pipe section was assumed uniform throughout the length of each section, though realistically, the energy lost to the room air would manifest itself in reduced fluid temperatures and, hence, lower overall heat flow into the air.

* The variation in water temperature in each pipe section was conservatively approximated by a bounding step function of temperature versus time.

* The initial bulk room air temperature was assumed to be $80^{\circ}F$, though it will somewhat lower at most times during the calendar year.

* The service water temperature at the inlet to the cooling unit was assumed to be $80^{\circ}F$, though it will also be somewhat lower at most times during the calendar year.

* The service water flow rate was assumed to be only 142 gpm throughout the event, though actual flow rates of 214 gpm would be achieved within approximately 20 minutes after diesel failure and will be maintained for the remainder of the event. Assuming the lower constant service water flow rate results in a lesser heat removal capability of the cooling unit throughout the event.

* Credit is not taken for subcooling the HPSI pump suction.

* The boundary walls, equipment, and piping masses were not accounted for as heat sinks. Calculations were made to confirm that including the heat capacity of these structures dramatically slows down the air temperature response.

* Convective Heat Transfer Coefficient calculation for all piping based on 140^{o}F air temperature. Rev. 1

7.0 THEORY AND CALCULATIONAL MODEL

The mathematical model for determining the transient response of the bulk room air temperature was established by equating the rate of change of the stored energy within the room to the sum of all the heat flows to or from the room air volume. The overall heat balance conservatively considered the heats sources and sinks associated with the room, as well as the mechanism by which the respective heat transfer occurs (i.e., conduction, convection, radiation).

$$dE/dt \,\Big|_{room} = \sum_{i=1}^{n} Q_i \qquad (1)$$

where: E = stored energy of the room volume

Qi = heat to/from the ith energy source/sink

n = number of sources and sinks

t = time

The mathematical representations for the heat sources and sinks, and the rate of change of the room volume stored energy are discussed next.

(A) Stored Energy

This term is equal to the sum of the energy stored by the room air, components within the room, and the room boundary walls. The individual contributions of these entities are each proportional to the product of their respective density, volume, specific heat, and temperature:

$$E = (\rho V C_p) \cdot T \qquad (2)$$

In order to maximize the rate of increase of the air temperature, and to ensure a conservative analysis, the respective heat exchange capacity of the boundary walls and components within the room were neglected. Consequently, the net heat transfer into the room is all used to increase the stored energy of the air volume:

$$dE/dt = (\rho V C_p)_{air} \cdot d\,T_b/dt \qquad (3)$$

(B) Motors and Room Lights

The energy emitted to the room air by these components is constant for any given time interval "k":

$$Q_m = Q(k)_{constant} \tag{4}$$

This was done to enable accounting for changes like removing the LPSI pump and motor after the first 2000 seconds of the event.

(C) Piping and Heat Exchangers

The energy emitted to the room air by these components is proportional to the difference between the fluid temperature of any given pipe and the room air temperature. This relationship is expressed as follows:

$$Q_p = U_o A \cdot (T_f - T_R) \tag{5}$$

where: Q_p = Rate of energy emission from a pipe or heat exchanger.

U_o = Overall heat transfer coefficient for the component. This coefficient was derived to account for convection, conduction, and radiation from pipes to the bulk room air.

A = Surface area of the component based on the same diameter as U_o.

T_f = Temperature of fluid within the component.

T_R = Temperature of the room air in the region of the component.

The temperature of the room air in the region of the particular pipe or heat exchanger would be either T_B (the bulk room air temperature at any time) or T_{ao} (the cooling unit exhaust air stream temperature at any time) depending upon where the particular component is located within the room.

Since the fluid temperature with any given pipe was to be approximated by linear functions of time:

$$T_f = A(k)\cdot t + B(k) \tag{6}$$

Therefore, for NP number of pipes that emit energy to air at T_B

$$Q_{P1} = \sum_{i=1}^{NP} (U_oA)_i \cdot [A(i,k)\cdot t + B(i,k) - T_b] \tag{7}$$

For NP2 number of pipes that emit energy to air at T_{ao} it was necessary to consider that T_{ao} would be a function of the performance of the cooling unit. This performance was modeled using the following performance factor:

$$EFF = \frac{T_b - T_{ao}}{T_b - T_{sw}} \tag{8}$$

where: EFF = cooling unit performance factor

T_{sw} = service water inlet temperature

The determination of the value used for EFF is discussed later in this report.

Solving equation (8) for T_{ao} and substituting along with equation (6) into equation (5) for NP2 number of pipes that emit energy to air at T_{ao} yields:

$$Q_{P2} = \sum_{\ell=1}^{NP2} (U_oA)_\ell\cdot[A(\ell,k)\cdot t + B(\ell,k) - (1\text{-}EFF)\cdot T_b - EFF\cdot T_{sw}] \tag{9}$$

(D) Cooling Unit

The energy removed from the bulk room air by the cooling unit is determined by:

$$Q_{fc} = (\rho \dot{V} C_p)_{air} \cdot (T_{ao} - T_b) \tag{10}$$

Combining equation (8) with equation (10) yields:

$$Q_{fc} = (\rho \dot{V} C_p)_{air} \cdot EFF \cdot (T_{sw} - T_b) \tag{11}$$

Solution

Combining equations (4), (7), (9), and (11) into equation (3), results in the following expression for the rate of change of bulk room air temperature with time:

$$dT_b/dt = -C1 \cdot T_b + C4 \cdot t + C5 \tag{12}$$

where:

$$C2 = 1.0/(\rho V\ C_p)\ \text{air}$$

$$C1 = C2 \cdot [\sum_{i=1}^{NP} (U_oA)_i + (1-EFF) \cdot \sum_{\ell=1}^{NP2} (U_oA)_\ell + (\rho\dot{V}C_p)_{air} \cdot EFF]$$

$$C4 = C2 \cdot [\sum_{i=1}^{NP} (U_oA)_i \cdot A(i,k) + \sum_{\ell=1}^{NP2} (U_oA)_\ell \cdot A(\ell,k)]$$

$$C3 = Q(k) - EFF \cdot \sum_{\ell=1}^{NP2} (U_oA)_\ell \cdot T_{sw} + EFF \cdot (\rho\dot{V}C_p) \cdot T_{sw}$$

$$C5 = C2 \cdot [\sum_{i=1}^{NP} (U_oA)_i \cdot B(i,k) + \sum_{\ell=1}^{NP2} (U_oA)_\ell \cdot B(\ell,k) + C3]$$

This differential equation governs the calculated time history response of the bulk room air temperature.

The solution for equation (12) is found using standard integration techniques for linear differential equations, that is, by finding a particular solution and the complementary function. The solution form is:

$$T_b = a1 + a2 \cdot t + a3 \cdot EXP[-C1(t-t_o)] \tag{13}$$

where the coefficients a_1, a_2, and a_3 can be expressed as follows:

$a_1 = C5/C1 - C4/(C1*C1)$

$a_2 = C4/C1$

$a_3 = T_{bo} + C4/(C1*C1) - C5/C1 - C4*\ t_o/C1$

where: T_{bo} = the bulk air temperature at the beginning of each of the time intervals used to describe the transient forcing functions.

t = real elapsed time from the initiation of the event.

t_o = the elapsed time from the beginning of the current time interval.

Since T_{bo} for each interval "n" is simply the bulk air temperature at the end of each interval "n-1", the overall time history solution of bulk room air temperature is an integrated sequence of solutions to equation (13) throughout the duration of the event. To facilitate the evaluation of various scenarios, and to facilitate verification of the calculational method, the analytical model was coded using FORTRAN so that the required calculations could be performed expeditiously.

7.1 Verification of a Calculational Model

In order to confirm that that the model performs as intended, a series of different sample cases was developed. The cases were designed such that exact temperature vs. time solutions could be independently developed for them by using hand calculations. The cases were also designed such that the various capabilities of the calculational procedure could be tested. The results of the hand calculations for these test cases were then compared with the results predicted by the coded model. For all of the cases, excellent agreement was obtained, thus verifying that the coded model performs as intended.

The values assumed for the various input parameters in each of the sample cases were selected to either produce particular bulk fluid temperature responses or to facilitate the hand calculations. Since the coded model has the capability to include various types of heat sources and heat sinks, one of the primary objectives of this study was to confirm that these features were functioning properly. Though some of the test cases do not necessarily reflect realistic situations, they were used to verify that terms in the mathematical model have been properly coded.

A summary of the various test cases follows:

CASE NO.	SIGNIFICANT FEATURES
1	Constant temp. pipe in a room that is initially colder than the pipe. No other heat sources or sinks.
2	Same as 1 except pipe is at initial room temperature plus a constant heat source in room.
3	Same as 1 except pipe temperature ramps up starting at initial room temperature.
4	Similar to 3 except 2 pipes in room, one of which has a linear increase in temperature with time and other having a linear decrease in temperature with time.
5	Similar to 4 except 2 pipes in room, one of which has a constant temperature with time and the other having a linearly increasing temperature with time.
6	Same as 1 except pipe temperature linearly ramps up from initial room temperature then stays constant (2 stage problem).
7	This case has no heat sources in the room. Only heat sink is conduction through walls with initial room air temperature larger than ground temperature.
8	No pipes in room. Only heat source is thru fans since initial room temperature is <u>less</u> than service water temperature to fan coolers. No heat sinks in room.

9 Same as 3 except a type 2 pipe is used (exposed to fan exhaust). Initial room temperature equals initial pipe temperature and are <u>less</u> than service water temperature to fan coolers.

10 Same as 6 except that after pipe stays at constant temperature for 1 hour, it linearly decreases to the starting temperature (3 stage problem).

11 This case considers two heated pipes in the room; one of which is cooled directly by the exhaust from the fan cooler and another which is cooled by the bulk air temperature in the room.

12 This case is similar to case 11 except that there are two pipes of each type in the room and one of each type has a constant pipe temperature whereas the other has a linear increase in temperature with time.

As noted previously, it was possible to perform hand calculations for all these problems and the results predicted by the coded model agreed with those calculations.

In addition to verifying that the coded model performs as intended, it was necessary to verify that the analytical model produces either realistic or conservative results. As noted in the previous section, the approach taken for this evaluation was to use a model which produces conservative results. This approach minimized the effort required to analyze the event and is consistent with good engineering practice. In addition, this approach reduced the uncertainty associated with the decision making process.

8.0 INPUT DATA

The input was derived based on CPCO's specification of the worst-case scenario of the WESGR response to a LOCA, and on pertinent design data for the room equipment and piping.

8.1 Pumps and Pump Motor

The heat emitted from the pump motors was based on actual performance flow rates, brake horsepower (BHP) and efficiences based on manufacturer's test performance curves for both the pumps and motors. The heat emitted from the motors is based on the inefficiency of the motor and coupling and the BHP of the respective units. The coupling efficiency for all pump motors was assumed to be 97%. The motor efficiencies used were:

P54B & P54C (Cont. Spray Pump Motors): 94.9%

P66B (HPSI Pump Motor): 94.3%

P67B (LPSI Pump Motor): 93.0%

The heat gain due to the inefficiency of the pumps was neglected since the majority of heat that is generated goes to the fluid being pumped, and since the balance of the heat is generated at the water cooled bearings and removed by the bearing cooling system. If 80% of pump inefficiency would be manifested as heat to the working fluid, it would result in raising the fluid temperature less than 1°F. The other 20% would be removed at the bearings via the bearing cooling system.

The equation used to determine the maximum heat emitted from the pump motors is as follows:

$$H.G. = 2544.5 \, \frac{BTUH}{HP} \left[\frac{(BHP)(1-\eta_m \cdot \eta_c)}{(\eta_m)\,(\eta_c)} \right] \qquad (14)$$

where: η_m = motor efficiency

η_c = coupling efficiency

The total pump motor heat emissions used as follows:

P54B (Containment Spray Pump)	50,524 BTUH
P54C (Containment Spray Pump)	50,524 BTUH
P66B (HP Safety Injection Pump, HPSI)	92,529 BTUH
P67B (LP Safety Injection Pump, LPSI)	109,075 BTUH
Total W/LPSI Pump =	302,652 BTUH
Total WO/LPSI Pump =	193,577 BTUH

The calculated heat emission from the room cooler fan motor into the room is based on the assumption that all heat generated goes into the room, therefore,

H.G. = (2544.5 BTUH/HP) (20 HP) = 50890 BTUH

The lighting load is 6800 BTUH, based on previous calculations for the design of the plant.

The total fixed heat gains are summarized as follows:

	WITH LPSI PUMP	WITHOUT LPSI PUMP
Pump Motors	302,652 BTUH	193,577 BTUH
Fan Motor	50,890 BTUH	50,890 BTUH
Lighting	6,800 BTUH	6,800 BTUH
TOTAL	360,342 BTUH	251,267 BTUH

8.2 Air Cooling Unit

The transient performance of the cooling unit was modeled based on equations (8) and (11) as identified in Section 7.0. Implementation of this model required calculating a value for the cooling unit performance factor EFF. The calculated EFF value was derived from the relationship between cooling unit heat removal capacity versus the difference between the inlet bulk air and service water streams' temperatures. This relationship, presented as Figure 1 (p.19), was developed by considering that the inlet air temperature is determined at each time throughout the event, assuming that the inlet Rev. 1

service water temperature is constant at 80°F, assuming that the heat removal capability of the cooling unit varies linearly with the ΔT between the inlet air and service water streams over the range of ΔT's pertinent to this analysis, and using the following heat transfer expressions:

$$Q_a = (\dot{M}Cp)_a \cdot (T_b - T_{ao}) \quad (15)$$

$$Q_{sw} = (\dot{M}Cp)_{sw} \cdot (T_{wo} - T_{w1}) \quad (16)$$

$$Q = U \cdot A \cdot LMTD \quad (17)$$

where:
- $\dot{M}_a$ = air mass flow rate (lbm/hr).
- $\dot{M}_{sw}$ = service water mass flow rate (lbm/hr).
- C_{pa} = specific heat of air (Btu/lbm °F).
- C_{psw} = specific heat of water (Btu/lbm °F).
- U = cooling unit overall heat transfer coefficient (Btu/ft^2 hr °F) (see below).
- LMTD = Log-Mean Temperature Difference (see below).
- T_b = air inlet temperature (°F).
- T_{ao} = air outlet temperature (°F).
- T_{w1} = service water inlet temperature (°F).
- T_{wo} = service water outlet temperature (°F).
- A = coil face area (ft^2).

Rev. 1

The overall heat transfer coefficient was calculated based on the cooling unit manufacturer's test data to account for a service water flow rate of 142 gpm instead of the test basis of 200 gpm. The applied equations are;

$$U_2 = [1/U_1 + A_s/A_T\ (1/h_2 - 1/h_1)]^{-1} \quad (18)$$

and;

$$h = (kw/d) \cdot [0.023(Re)^{0.8}\ (Pr)^{0.4}] \quad (19)$$

where: A_s = coil face area (ft^2).

A_T = coil total tube surface area (inside heat transfer tube area) (ft^2).

k_w = thermal conductivity of tube material (BtuH/ft ^oF).

h = heat transfer coefficient (BtuH/ft^2 ^oF).

Re = Reynold's Number.

Pr = Prandtl Number.

U_1 = Manufacturer's overall HTC (based on 200 gpm service water) (BtuH/ft^2 ^oF).

U_2 = Adjusted overall HTC (per equation (18)) (BtuH/ft^2 ^oF).

The LMTD is defined as:

$$LMTD = \frac{[(T_{ai} - T_{wo}) - (T_{ao} - T_{wi})]}{\ln \frac{(T_{ai} - T_{wo})}{(T_{ao} - T_{wi})}} \qquad (20)$$

The relationship between the cooling unit heat removal capacity and the ΔT between the inlet air and service water streams was developed by first redefining equation (11) as;

Rev. 1

$$Q = K\ (T_b - T_{wi}) \qquad (21)$$

then combining this equation with equation (15) to result in:

$$K\ (T_b - T_{wi}) = (\dot{M}Cp)_a\ (T_b - T_{ao}) \qquad (22)$$

Rearranging:

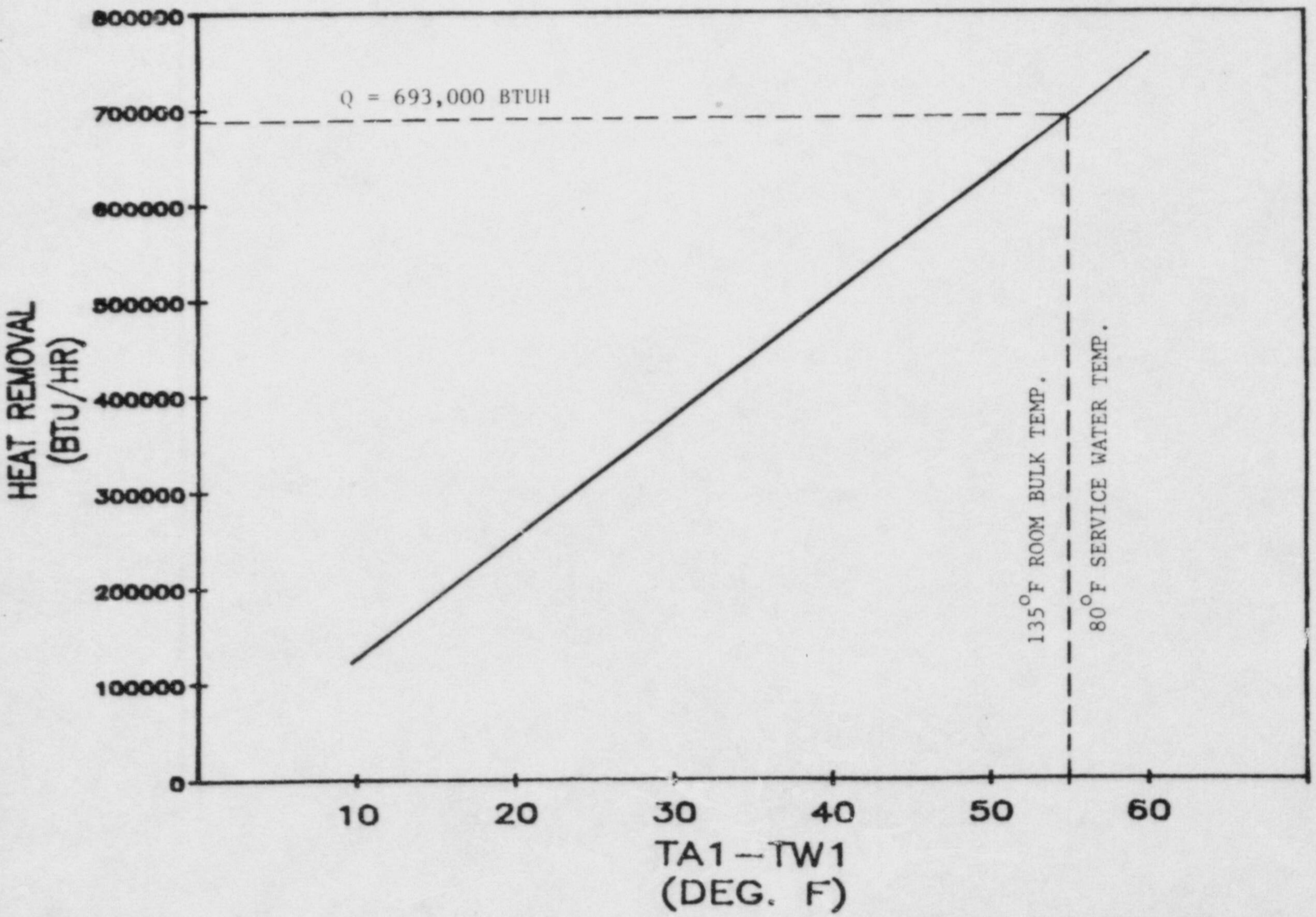
$$[K/(\dot{M}Cp)_a] = \frac{(T_b - T_{ao})}{(T_b - T_{wi})} = EFF \qquad (23)$$

Equation (23) is equivalent to equation (8) in section 7.0.

The second step was to rearrange equations (15) and (16) as follows;

$$T_{ao} = T_b - [Q/(\dot{M}Cp)_a] \qquad (24)$$

SAFEGUARDS ROOM AIR COOLER PERFORMANCE CURVE



LEGEND

TA1=Inlet air temp.

TW1=Inlet Water temp.

$Q=12600(TA1-TW1)$

$e=\frac{(TA1-TA2)}{(TA1-TW1)} = .85$

FIGURE 1

and:

$$T_{wo} = T_{wl} + [Q/(\dot{M}Cp)_{sw}] \tag{25}$$

For each ΔT ($T_b - T_{wl}$), equations (17), (24), and 25) were iterated upon unit equations (15), (16), and (17) were found to be equal. That is, for each T_b in ($T_b - T_{wl}$) a value for Q was assumed and used in equations (24) and (25) to solve for T_{ao} and T_{wo}. Next, these temperatures were used in equation (17) to solve for Q. If this Q was less than the Q value assumed in the previous iteration step, then the assumed Q was reduced and used in the next iteration step. This iterative process was carried out to the extent necessary to develop Figure 1 for the range of ΔT's pertinent to this analysis. The value of EFF was obtained by dividing the slope of the line in Figure 1 by the appropriate product (MC_p) for the air stream.

Based on cooling unit air flow of 13590 cfm (with only one fan operable), and service water at 142 gpm and 80^{o}F, EFF was calculated to be 0.857. This value was rounded down to 0.85 and used in the analysis.

8.3 Piping

The total heat transfer coefficient for pipes is based on an overall heat transfer coefficient defined as $h_t = h_o + h_r$

h_o = Heat transfer coefficient that accounts conduction and convection from the outer pipe surface (BTUH/ft^2 - ^{o}F).

h_r = Radiant heat transfer coefficient (BTUH/ft^2 - ^{o}F)

The film heat transfer coefficient is due to the air velocity from the room cooler across the pipes. The equation used was taken from Fan Engineering by Buffalo Forge Co. The equation is as follows:

$$h_o = 0.24\ (k_f/D_o)\cdot(\frac{D_oG}{\mu_f}) \tag{26}$$

h_o = heat transfer coefficient (BTUH/ft^2 - ^oF)

D_o = Pipe O.D. (ft)

K_f = Thermal Conductivity (BTUH-in./ft^2-^oF)

M_f = Absolute Viscosity of film (lbs. mass/ft. - sec.)

G = Mass Velocity of fluid G = .017V (lbs/ft^2 - sec.)

V = Velocity (ft/min)

The air velocities used for this analysis are based on actual velocity test data taken at Palisades. Figure (2) presents these velocities.

The radiation heat transfer coefficient is based on the Stefan-Boltzmann equation

$$h_r = 0.1713 \times 10^{-8} \cdot e \cdot \frac{\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right]}{T_1 - T_2} \quad (27)$$

Rev. 1

e = emissivity

T_1 = Temperature of the pipe surface, assumed to be the temperature of fluid (^oR)

T_2 = Assumed ambient temperature of 595^oF (135^oF). (To be conservative over the duration of the event).

h_r = heat transfer coefficient (BTUH/ft^2 - ^oF)

The fluid temperatures in the piping systems are based on the containment sump water temperature once recycle begins. The sump water temperatures were based on Combustion Engineering's analysis dated November 20, 1986. This data is presented in Figure (3). Prior to containment sump water recycle, it was assumed that most fluid temperatures were 88^oF, while two pipe sections contained 114^oF fluid.

WEST ENGINEERED SAFEGUARDS ROOM PIPING

(INVOLVED IN EVENT)

PIPE OD (IN.) NOMINAL	PIPE LENGTH (FT.)		PIPE AREA ($FT.^2$)		PIPE TEMP.	AIR TEMP.	AIR VELOCITY (FT./MIN.)	
	W/LPSI	W/O LPSI	W/LPSI	W/O LPSI				
3	17.67	17.67	16.20	16.20	TSO	TAI	150	
4	43.33	43.33	51.06	51.06	TSI	TAI	150	
6	15.83	15.83	27.45	27.45	TSI	TAI	150	
8	78.00	78.00	176.15	176.15	TSI	TAI	150	
8	4.00	4.00	9.03	9.03	TSI	TAI	200	
8	18.92	18.92	42.73	42.73	TSI	TAI	300	
8	7.25	7.25	16.37	16.37	TSI	TAI	450	
8	20.67	20.67	46.68	46.68	TSO	TAI	150	
8	6.58	6.58	14.86	14.86	TSI	TAO	1200	
10	99.92	76.00	281.19	213.88	TSI	TAI	150	
10	11.75	2.83	33.07	7.96	TSI	TAI	300	
10	51.50	51.50	144.93	144.93	TSO	TAI	150	
10	3.42	0	9.62	0	TSI	TAO	1200	
12	15.83	0	52.85	0	TSI	TAI	150	
12	7.58	0	25.30	0	TSI	TAI	250	
12	2.00	0	6.68	0	TSI	TAO	1200	
12	29.00	29.00	96.81	96.81	TCO	TAI	150	
14	156.83	82.67	575.00	302.99	TSI	TAI	150	
14	20.30	20.30	74.40	74.40	TSI	TAI	200	
14	15.00	15.00	54.98	54.98	TSI	TAI	500	
14	4.00	4.00	14.66	14.66	TSI	TAO	1000	
14	9.00	9.00	32.99	32.99	TSI	TAO	1200	
14	4.00	4.00	14.66	14.66	TSI	TAO	2000	
18	30.56	30.56	144.00	144.00	TCO	TAI	150	
24	44.42	44.42	279.11	279.11	TSI	TAI	150	
45	47.00	47.00	553.71	553.71	TCO	TAI	500	Rev. 1

TSI = TEMPERATURE OF SUMP WATER ENTERING HEAT EXCHANGER
TSO = TEMPERATURE OF SUMP WATER EXITING HEAT EXCHANGER
TCO = TEMPERATURE OF CCW EXITING HEAT EXCHANGER
TAI = TEMPERATURE OF AIR ENTERING AIR COOLER
TAO = TEMPERATURE OF AIR EXITING AIR COOLER

CONTAINMENT SUMP WATER TEMPERATURE PROFILE BY COMBUSTION ENGINEERING DATED NOVEMBER 20, 1986 *

TIME					
SEC.	MIN.	HRS.	DAYS	SUMP WATER TEMP. (Deg. F.)	
0	-	-	-	138	
1	-	-	-	169	
5.99	-	-	-	218	
33.4	-	-	-	220	
179.9	3	-	-	233	
255.9	4.27	-	-	240	
295	4.9	-	-	243	
353	5.9	-	-	246	
373	6.2	-	-	247	
392	6.5	-	-	247	
412	6.9	-	-	247	
431	7.2	-	-	247	
451	7.5	-	-	247	
490	8.2	-	-	247	
1,440	24	-	-	233	
1,540	25.7	-	-	231	
2,000	33.3	.56	-	223	Recirc. Begins
2,480	41.3	.69	-	224	
2,980	49.7	.83	-	226	
3,480	58	.97	-	225	
4,780	79.7	1.3	-	224	
5,680	94.7	1.6	-	224	
8,680	144.7	2.4	-	225	
15,400	256.7	4.3	-	226	
22,900	381.7	6.4	-	224	
42,400	706.7	11.8	-	213	
61,900	1031.7	17.2	-	202	
81,400	1356.7	22.6	-	195	
118,000	1966.7	32.8	-	185	
508,000	8466.7	141.1	5.9	153	
898,000	14966.7	249.4	10.4	144	

* CE REPORT, "FINAL REPORT, REV. 02 TO CPCO FOR PHASE 1 ANALYSIS TO DETERMINE THE PALISADES PLANT CONTAINMENT RESPONSE TO LOCAS AND SDC SYSTEM PERFORMANCE", TASK 601634, 11/26/86, JOB = AL2OVTZ, JSN = AVPC

Rev. 1

9.0 RESULTS

The estimated bulk room air temperature time history is presented in Figures (4) and (5).

The most significant change in the bulk air temperature would occur within the first hour immediately following response to a design basis LOCA. During this first hour, the heat transfer to the room air volume would reach a temporary steady state to be maintained for approximately six hours. At the steady condition where the bulk room air temperature reaches its maximum value of 135°F, the temperature of the cooling unit exhaust air is approximately 88°F. As anticipated, the profile of this time history indicates that the bulk room air temperature variation is being forced by the changes in the temperatures of the fluids flowing in the various pipe sections.

The corresponding heat flow from all piping involved in the transient at that steady state condition where T_{bulk} reached 135°F was estimated to be 399246 BtuH/hr. of this piping total, approximately 130000 BtuH is radiative. The corresponding total constant heat emitted into the air volume would be 251267 BtuH/hr, and the total heat flow into the air volume equaled that removed via the cooling unit (with only one fan operational) at 650513 BtuH/hr. Rev. 1

FIGURE 4

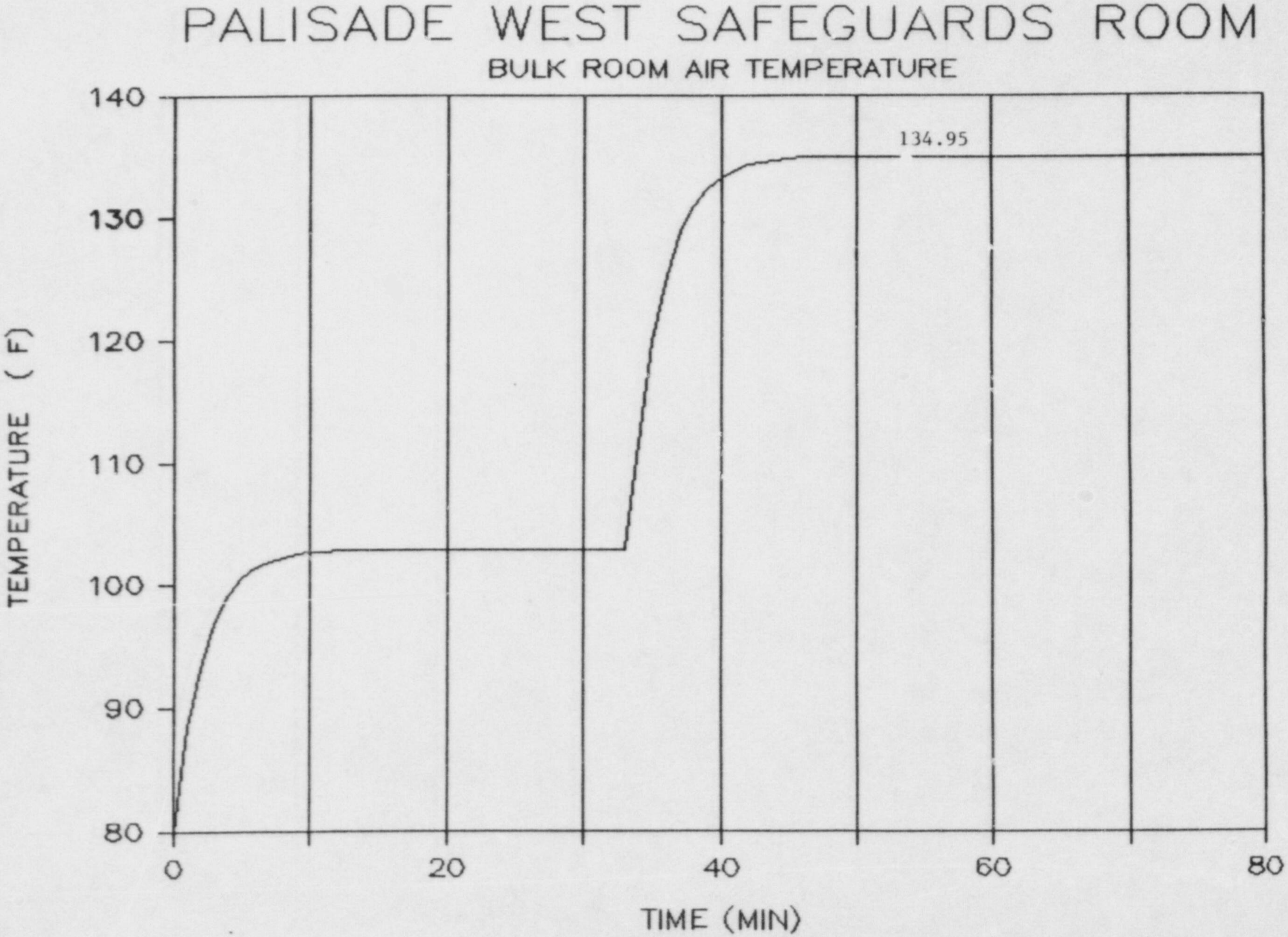
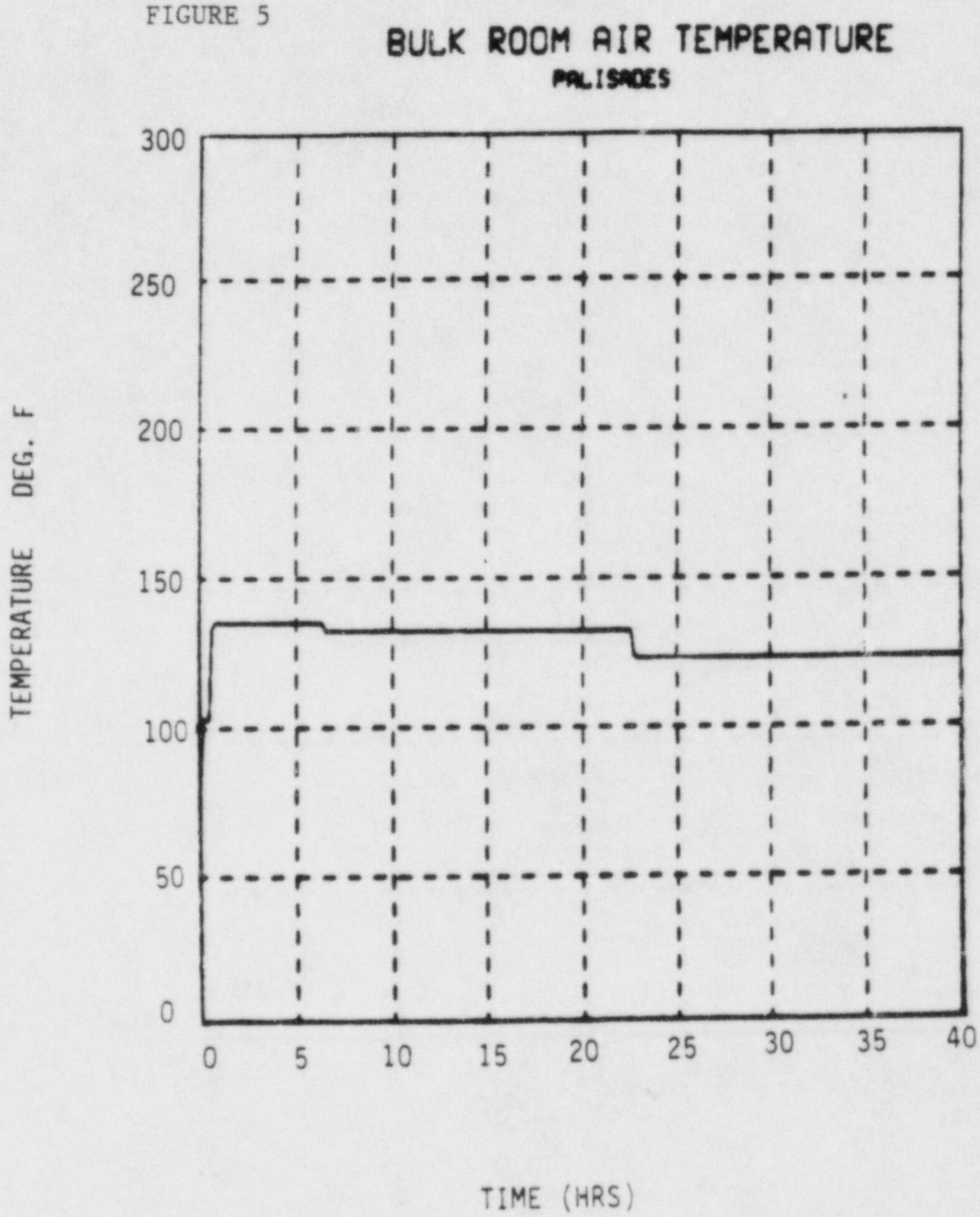


FIGURE 4

FIGURE 5


BULK ROOM AIR TEMPERATURE
PALISADES
TEMPERATURE DEG. F
300
250
200
150
100
50
0
0
5
10
15
20
25
30
35
40
TIME (HRS)

10.0 ASSESSMENT OF DIFFERENCES BETWEEN PREVIOUS WESTINGHOUSE AND CPCO ANALYSES

PURPOSE:

To delineate and reconcile the underlying differences between the previous W and CPCO heat transfer analyses of the WESGR. After reconciling the differences, the parameters used in the updated Westinghouse analysis were identified. These parameters are presented in Tables (1) through (4).

PREVIOUS W ANALYSIS CONCLUSION

With 80°F service water to the WESGR cooling unit, and only one fan operating, the bulk room air temperature of this room, during a design basis LOCA, will not exceed 139°F.

PREVIOUS CPCO ANALYSIS CONCLUSION

To prevent the bulk room air temperature of the WESGR from exceeding 135°F during a design basis LOCA, the service water to the room cooling unit, with only one fan operating, must be 59°F or less.

10.1 GENERAL DATA COMPARISON

The significant differences in the results of the two analyses are summarized below. Details regarding the reasons for these differences are contained in the attachments.

Parameter	Westinghouse Analysis	CPCO Analysis	Updated W Analysis
Pump Heat	110431*	0	0
Pump Motor Heat	154377	287600	193577
Piping Heat	313777	601544	399246
S.D. HX Heat	23798	(in piping)	(in piping)
Fan Motor Heat	50890	59810	50890
Room Light Heat	6800	6800	6800
Total Heat	660073	955754	650513

Temperatures used to Obtain Heat Loads

T_{Bulk}	139°F	100°F	135°F
T_{ao} (fan outlet air)	92°F	70°F	88°F
T_s (service water)	80°F	59°F	80°F

20% of pump inefficiency was assumed emitted into bulk room air.

NOTE: * All heat rate units are BTU/hr

10.2

TABLE 1

SUMMARY OF DIFFERENCES
WESTINGHOUSE AND CONSUMERS POWER COMPANY
THERMAL ANALYSES OF WEST ENGINEERED SAFEGUARDS ROOM

	PUMPS	PUMP MOTORS
Westinghouse	110431 Btu/hr	154377 Btu/hr
Q_T = 660073 T_B = 139.25 T_{ao} = 91.85 T_{sw} = 80.00 (Transient Analysis)	- Assumed 20% Pump Inefficiency Converted to Heat.	- Used design BHP. - Removed LPSI motor heat load after 2000 sec.
CPCO	0.0	287600 Btu/hr
Q_T = 955754 T_B = 100 T_{ao} = 70.0 T_{sw} = 59 (Steady State Analysis)	- Neglected heat due to pump inefficiency.	- Used name plate total HP. - Included auxiliary feedwater pump. - Excluded LPSI motor heat.
UPDATED WESTINGHOUSE	0.0	193577 Btu/hr
Q_T = 650513 T_B = 134.95 T_{ao} = 88.24 T_{sw} = 80 (Transient Analysis)	- Neglected heat due to pump inefficiency.	- Used actual motor HP applicable to pump operation.

TABLE 2

	FAN COOLER	FAN MOTOR
WESTINGHOUSE	660073 Btu/hr	50890 Btu/hr
	- Conservatively approximated performance in terms of HTC used and Q vs. ΔT.	- Used the name plate total HP.
CPCO	955754 (Btu/hr)	59810 (Btu/hr)
	- CPCO calculated design HTC and used Q vs. ΔT based on manufacturer profile test documentation.	- Used the calculated BHP divided by the motor efficiency.
UPDATED WESTINGHOUSE	650513 (Btu/hr)	50890 (Btu/hr)
	- Calculated an overall HTC based on manufacturer's test documentation - Refined Q vs. ΔT performance data to reflect anticipated performance.	- No change.

The nameplate motor horsepower is greater than the required horsepower for actual operating flow and temperature. Based on a performance report for the ESGR air coolers from Buffalo Forge Co. dated April 8, 1983 the BHP is 19.2 @ 14,400 CFM and 86°F. Since the operating conditions are 13,590 CFM and 88°F, fan inlet temperature, the BHP is corrected by using the Fan Law

$$HP_a = HP_b \times \left(\frac{CFM_a}{CFM_b}\right)^3 \times \frac{density_a}{density_b}$$

Subscript "a" represents the operating conditions and subscript "b" represents the design conditions. Rev. 1

$density_a$ @ 88°F = .07243 lbs/ft^3
$density_b$ @ 86°F = .07269 lbs/ft^3

Nameplate HP = 20 > Corrected HP of 16.08 ÷ Motor Efficiency °F .87

Therefore, if the BHP is corrected for actual temperature and flow rate, then using the nameplate motor horsepower is more conservative than BHP + motor efficiency. This conservatism was not removed since it is common HVAC practice to use nameplate HP to calculate heat load from a fan.

TABLE 3

	HEAT EXCHANGER	LIGHTING
WESTINGHOUSE	23798 Btu/hr	6800 Btu/hr
	- Conservatively modeled this as a constant heat source throughout event.	
CPCO	NA	6800 Btu/hr
	- Modeled as a large pipe.	
UPDATED WESTINGHOUSE	NA	6800 Btu/hr
	- Modeled as a large pipe.	- No change

TABLE 4

	PIPING
WESTINGHOUSE Q_T = 660073 T_B = 139.25 T_{AO} = 91.85 T_{SW} = 80.00	313777 Btu/hr - W did not consider 3" x 16' and 8" x 24' ESR sprayline piping. - W included heated LPSI pipe lines throughout the event. - Radiative HTC calculation based on 226°F pipe wall temperature and 135°F bulk air temperature (all radiant heat absorbed by air). - Convective HTC calculation for all piping based on 140°F air temperature. (Total heat flow from the piping is governed by the ΔT between the piping and air temperature at any time). Rev. 1
CPCO Q_T = 955754 T_B = 100°F T_{AO} = 70.0°F T_{SW} = 59°F	601544 Btu/hr - CPCO Modeled SD HXS as pipes. - CPCO included auxiliary feedwater pump discharge line. (L = 15.8' on 6.625") (L = 17.6' on 3.5") Rev. 1 - Convective HTC based on calcs. 100°F air temperature. (ΔT for heat flow was max. pipe temp. less 135°F air temp.). - Radiation from pipe handled as 80% to wall, then to air, and 20% directly to air. The ΔT for heat flow was max. pipe temp. less 100° air temp., and max. pipe temp. less 135°F boundary wall temp.
UPDATED WESTINGHOUSE Q_T = 650513 T_B = 134.95 T_{AO} = 88.24 T_{SW} = 80.00	399246 (Btu/hr) - Included 3" X 16' and 8" x 24' ESR sprayline piping (15 ft^2 and 54 ft^2 respectively). (Inadvertently omitted in previous W analysis. The 8" ESR spray line was added per CPCo request for additional conservatism. - Removed LPSI piping after 2000 sec. (Improved accuracy, and removed unneeded conservatism). - Modeled the two SD HX's as one large pipe. (Improved accuracy and removed unneeded conservatism).

10.3

TABLE 5

ASSESSMENT OF PUMP MOTOR DATA DIFFERENCES

Motor	Heat Load - (Btu/hr) Westinghouse	CPCO
LPSI (1)	0	0
HPSI (1)	69447	88570
Cont. Spray (2)	84930	110460
Aux. Feed (1)	0	88570
Sump Pump (1)	0	0
Total	154377	287600

	Westinghouse		CPCO	
Parameter	Cont. Spray	HPSI	Cont. Spray	HPSI
BHP	210	320	250	400
Basis of BHP	Design Point	Design Point	Name Plate	Name Plate
Motor Efficiency	0.949	0.943	0.92	0.92
Coupling Factor	0.97	0.97	1.0	1.0

Rev. 1

Note: o The updated Westinghouse analysis was based on pump motor data as described on page 16.

10.4

TABLE 6
ASSESSMENT OF PIPING DATA DIFFERENCES

Note that the updated Westinghouse analysis was based on all previous Westinghouse data plus the addition of the nominal 3" x 16' and 8" x 24' ESR pipe sections, and the model of the two S.D. HX's. Figure 2 on page 21 presents all the data for the pipes used in the updated W analysis. Also note that the CPCo analysis was based on estimating radiative heat flow to the room air directly from the piping and indirectly from the boundary walls by using conservative pipe, air, and wall temperatures. CPCo did not calculate and use effective radiative heat transfer coefficients. See the Table 4 section on piping differences for additional details. Rev. 1

(A) 24.0" Diameter

	Parameter	Westinghouse Analysis	CPCO Analysis
(1)	Length (ft)	44.4	44.5
(2)	Surface Area (ft^2)	279	280
(3)	Air Velocity (ft/min)	150.0	150.0
(4)	Air Temp. (^{o}F)	139	100.0
(5)	Pipe Temp. (^{o}F)	226	223
(6)	HTC Radiation (BtuH/ft ^{o}F)	1.43	-
(7)	HTC Convection (BtuH/ft^2 ^{o}F)	0.79	0.798
(8)	Total HTC (BtuH/ft^2 ^{o}F)	2.22	-

(B) 18.0" Diameter

	Parameter	Westinghouse Analysis	CPCO Analysis
(1)	Length (ft)	30.56	23.77*
(2)	Surface Area (ft^2)	144	112
(3)	Air Velocity (ft^2/min)	150	150
(4)	Air Temp. (^{o}F)	139	100
(5)	Pipe Temp (^{o}F)	144	147
(6)	HTC Radiation (BtuH/ft ^{o}F)	1.43	-
(7)	HTC Convection (BtuH/ft^2 ^{o}F)	0.88	0.89
(8)	Total HTC (BtuH/ft^2 ^{o}F)	2.31	-

*Piping isometric vaguely described junction of 12" and 18" diameter sections. Westinghouse conservatively used 18" for the entire pipe run. Rev. 1

TABLE 6 (CONTINUED)

(C) 14.0" Diameter

Parameter	Westinghouse Analysis	CPCO Analysis
(1) Length (ft)		
a	156.83	46.1*
b	9.0	88.7
c	4.0	-
d	15.0	-
e	20.3	-
f	4.0	-
(2) Surface Area (ft^2)		
a	575	169
b	33	325
c	15	-
d	55	-
e	74	-
f	15	-
(3) Air Velocity (ft/min)		
a	150	150
b	1200	1300
c	1000	-
d	500	-
e	200	-
f	2000	-
(4) Air Temp. (^oF)		
a	139	100
b	92	70
c	92	-
d	139	-
e	139	-
f	92	-
(5) Pipe Temp. (^oF)		
a	226	223
b	226	223
c	226	-
d	226	-
e	226	-
f	226	-
(6) HTC Radiation (a-f) (BtuH/ft ^oF	1.43	-

* Westinghouse included LPSI piping. The updated Westinghouse analysis did not include the LPSI piping.

Rev. 1

** a Through f reflects different pipe sections.

TABLE 6 (CONTINUED)

(C) 14.0" Diameter (Continued)

Parameter	Westinghouse Analysis	CPCO Analysis
(7) HTC Convection ($BtuH/ft^2 - {}^oF$)		
a	0.98	0.979
b	3.43	4.68
c	3.05	-
d	2.01	-
e	1.16	-
f	4.63	-
(8) Total HTC ($BtuH/ft^2 - {}^oF$)		
a	2.41	-
b	4.86	-
c	4.48	-
d	3.44	-
e	2.59	-
f	6.06	-

TABLE 6 (CONTINUED)

(D) 12.75" Diameter

Parameter	Westinghouse Analysis	CPCO Analysis
(1) Length (ft)		
a	29	39*
b	15.83	-
c	7.6	-
d	2.0	-
(2) Surface Area (ft^2)		
a	97	122.5
b	53	-
c	25	-
d	7.0	-
(3) Air Velocity (ft/min)		
a	150	150
b	150	-
c	250	-
d	2000	-
(4) Air Temp. (^oF)		
a	139	100
b	139	-
c	139	-
d	92	-
(5) Pipe Temp. (^oF)		
a	144	147
b	226	-
c	226	-
d	226	-
(6) HTC Radiation (BtuH/ft^2 - ^oF)		
a	1.43	-
b	1.43	-
c	1.43	-
d	1.43	-
(7) HTC Convection (BtuH/ft^2 - ^oF)		
a	1.05	1.04
b	1.05	-
c	0.851	-
d	3.53	-

*See note under the 18" diameter section of this table.

Rev. 1

TABLE 6 (CONTINUED)

(D) 12.75" Diameter (Continued)

Parameter	Westinghouse Analysis	CPCO Analysis
(8) Total HTC ($BtuH/ft^2 - {}^oF$)		
a	2.45	-
b	2.45	-
c	2.81	-
d	4.96	-

TABLE 6 (CONTINUED)

(E) 10.75" Diameter

Parameter	Westinghouse Analysis	CPCO Analysis
(1) Length (ft)		
a	3.4	11.4*
b	8.9	64.6
c	119.5	51.5
d	51.5	-
(2) Surface Area (ft^2)		
a	10	30
b	25	169
c	336	135
d	145	-
(3) Air Velocity (ft^2/min)		
a	1200	1300
b	300	150
c	150	150
d	150	-
(4) Air Temp. (^oF)		
a	92	70
b	139	100
c	139	100
d	139	-
(5) Pipe Temp. (^oF)		
a	226	223
b	226	223
c	226	144
d	147	-
(6) HTC Radiation (a-d) (BtuH/ft^2 - ^oF)	1.43	-
(7) HTC Convection (BtuH/ft^2 - ^oF)		
a	3.78	4.93
b	2.85	1.08
c	1.19	1.08
d	1.09	-
(8) Total HTC (BtuH/ft^2 - ^oF)		
a	5.21	-
b	4.28	-
c	2.62	-
d	2.52	-

*Westinghouse was unnecessarily conservative. This conservatism was reduced on the updated Westinghouse analysis. Rev. 1

TABLE 6 (CONTINUED)

(F) 8.625" Diameter

Parameter	Westinghouse Analysis	CPCO Analysis
(1) Length (ft)		
a	50.7	31.3*
b	4.0	80.5
c	19.0	19.1
d	7.3	-
e	6.6	-
f	19.0	-
(2) Surface Area (ft^2)		
a	115	71
b	9	182
c	43	43
d	16	-
e	2	-
f	43	-
(3) Air Velocity (ft/min)		
a	150	1300
b	200	150
c	300	150
d	450	-
e	1200	-
f	150	-
(4) Air Temp. (^oF)		
a	139.25	70
b	139.25	100
c	139.25	100
d	139.25	-
e	92	-
f	139.25	-
(5) Pipe Temp. (^oF)		
a	226	223
b	226	223
c	226	144
d	226	-
e	226	-
f	147	-
(6) HTC Radiation ($BtuH/ft^2$ - ^oF) (a-f)	1.43	-

*Westinghouse did not factor in containment spray discharge from the E... Per CPCo request, this was included on the updated Westinghouse analysis. | Rev. 1

(F) 8.625" Diameter (Continued)

Parameter	Westinghouse Analysis	CPCO Analysis
(7) HTC Convection (BtuH/ft^2 - ^{o}F)		
a	1.19	5.15
b	1.41	1.18
c	1.80	1.18
d	2.29	-
e	4.13	-
f	1.19	-
(8) Total HTC (BtuH/ft^2 - ^{o}F)		
a	2.62	-
b	2.84	-
c	3.23	-
d	3.72	-
e	5.56	-
f	2.62	-

TABLE 6 (CONTINUED)

(G) 6.625" Diameter

Parameter	Westinghouse Analysis	CPCO Analysis
(1) Length (ft)	15.8	31.7*
(2) Surface Area (ft^2)	28	55
(3) Air Velocity (ft/min)	150	150
(4) Air Temp. (^oF)	139	100
(5) Pipe Temp. (^oF)	226	223
(6) HTC Radiation (BtuH/ft^2 ^oF)	1.43	-
(7) HTC Convection (BtuH/ft^2 ^oF)	1.32	1.3
(8) Total HTC (BtuH/ft^2 ^oF)	2.75	-

*Westinghouse did not include the auxiliary feed pump suction line because the pump would not be operating. Rev. 1

(H) 4.5" Diameter

Parameter	Westinghouse Analysis	CPCO Analysis
(1) Length (ft)	43.3	42.3
(2) Surface Area (ft^2)	51	50
(3) Air Velocity (ft/min)	150	150
(4) Air Temp. (^oF)	139	100
(5) Pipe Temp. (^oF)	226	223
(6) HTC Radiation (BtuH/ft^2 ^oF)	1.43	-
(7) HTC Convection (BtuH/ft^2 ^oF)	1.54	1.51
(8) Total HTC (BtuH/ft^2 ^oF)	2.97	-

TABLE 6 (CONTINUED)

(I) 3.5" Diameter

Parameter	Westinghouse Analysis	CPCO Analysis
(1) Length (ft)	-	35.3
(2) Surface Area (ft^2)	-	32
(3) Air Velocity (ft/min)	-	150
(4) Air Temp. (^oF)	-	100
(5) Pipe Temp. (^oF)	-	223
(6) HTC Radiation (BtuH/ft^2 ^oF)	-	-
(7) HTC Convection (BtuH/ft^2 ^oF)	-	1.66
(8) Total HTC (BtuH/ft^2 ^oF)	-	-

(J) 45" Diameter (S.O. HX's)

Parameter	Westinghouse Analysis	CPCO Analysis
(1) Length (ft)	-	46.94
(2) Surface Area (ft^2)	-	553
(3) Air Velocity (ft/min)	-	150
(4) Air Temp (^oF)	-	100
(5) Pipe Temp. (^oF)	-	147
(6) HTC Radiation (BtuH/ft^2 ^oF)	-	-
(7) HTC Convection (BtuH/ft^2 ^oF)	-	0.627
(8) Total HTC (BtuH/ft^2 ^oF)	-	-

NOTE: The updated W analysis modeled the 3" nominal pipe and the two S.D. HX's as described in Figure 2 on page 21.

TABLE 7

IV. PIPING HEAT TRANSFER AREA COMPARISON

Pipe Diameter	Westinghouse	CPCO	Difference
(1) 24.0"			
- Exposed to T_B	279	280	-1
- Exposed to T_{AO}	-	-	-
(2) 18.0"			
- Exposed to T_B	144	112	32
- Exposed to T_{AO}	-	-	-
(3) 14.0"			
- Exposed to T_B	704	169	535
- Exposed to T_{AO}	62	325	-263
(4) 12.75"			
- Exposed to T_B	175	123	52
- Exposed to T_{AO}	7	0	7
(5) 10.75"			
- Exposed to T_B	506	304	202
- Exposed to T_{AO}	10	30	-20
(6) 8.625"			
- Exposed to T_B	226	225	1
- Exposed to T_{AO}	15	71	-56
(7) 6.625"			
- Exposed to T_B	28	55	-27
- Exposed to T_{AO}	-	-	-
(8) 4.5"			
- Exposed to T_B	51	50	1
- Exposed to T_{AO}	-	-	-
(9) 3.5"			
- Exposed to T_B	-	32	-32
- Exposed to T_{AO}	-	-	-
(10) 45" (HX's)			
- Exposed to T_B	-	553	-553

TABLE 7 (CONTINUED)

Pipe Diameter	Westinghouse	CPCO	Difference
TOTAL			
- Exposed to T_B	2138	1903	210
- Exposed to T_{AO}	96	426	-332
*Total	2234	2329	-122

NOTES: - CPCO had 332 ft^2 more surface exposed to T_{AO}.

- CPCO had 122 ft^2 more total surface area.

- CPCO Δ T's between pipe and air were 123°F and 151°F, while those used by W (at steady state of max. room bulk temp) were 86.75°F and 134°F respectively.

10.5 ASSESSMENT OF COOLING UNIT DIFFERENCES

The earlier section on analysis input data explained how the relationship between the heat transfer capability of the cooling unit and the unit's inlet air and service water streams temperature difference was used to calculate the value for EFF. In the previous Westinghouse analysis, the calculated value of 0.857 for EFF was conservatively rounded-down to 0.80. This value was based on air flow and service water stream conditions specific to the Westinghouse analysis. The previous CPCO analysis an EFF value was 0.83 based on air flow and service water conditions specific to their analysis. The updated Westinghouse analysis used an EFF value of 0.85.

Another difference regarding the cooling unit was in the respective overall heat transfer coefficients (HTC) used. Previously Westinghouse calculated an overall HTC of 1085 $BtuH/ft^2\ ^oF$ based on manufacturer's design information. This value was then rounded-down to 1000 $BtuH/ft^2\ ^oF$. Similarly, CPCO calculated, then used, 1050 $BtuH/ft^2\ ^oF$. The difference between the Westinghouse value of 1085 and the CPCO value of 1050 was due to the fact that Westinghouse used 142 gpm service water flow, while CPCo used 109 gpm.

The updated Westinghouse analysis used an overall HTC of 1085 $BtuH/ft^2\ ^oF$.

Rev. 1

Figure (6) illustrates the relationships between the heat removal capability of the cooling unit and the difference between the inlet air and service water stream's temperatures that were used in the previous Westinghouse and CPCO analyses. The slope of the curve used in the CPCo analysis is slightly steeper than that of the previous Westinghouse analysis in-part because the corresponding Westinghouse HTC of 1000 $BtuH/ft^2\ ^oF$ was less than the CPCo value of 1050 $BtuH/ft^2\ ^oF$. Clearly, the larger the HTC, the better the cooling unit's heat removal capability and hence, the steeper the relationship between Q and ΔT. The relationship used in the updated Westinghouse analysis appears on page 19.

SAFEGUARDS ROOM AIR COOLER PERFORMANCE CURVE

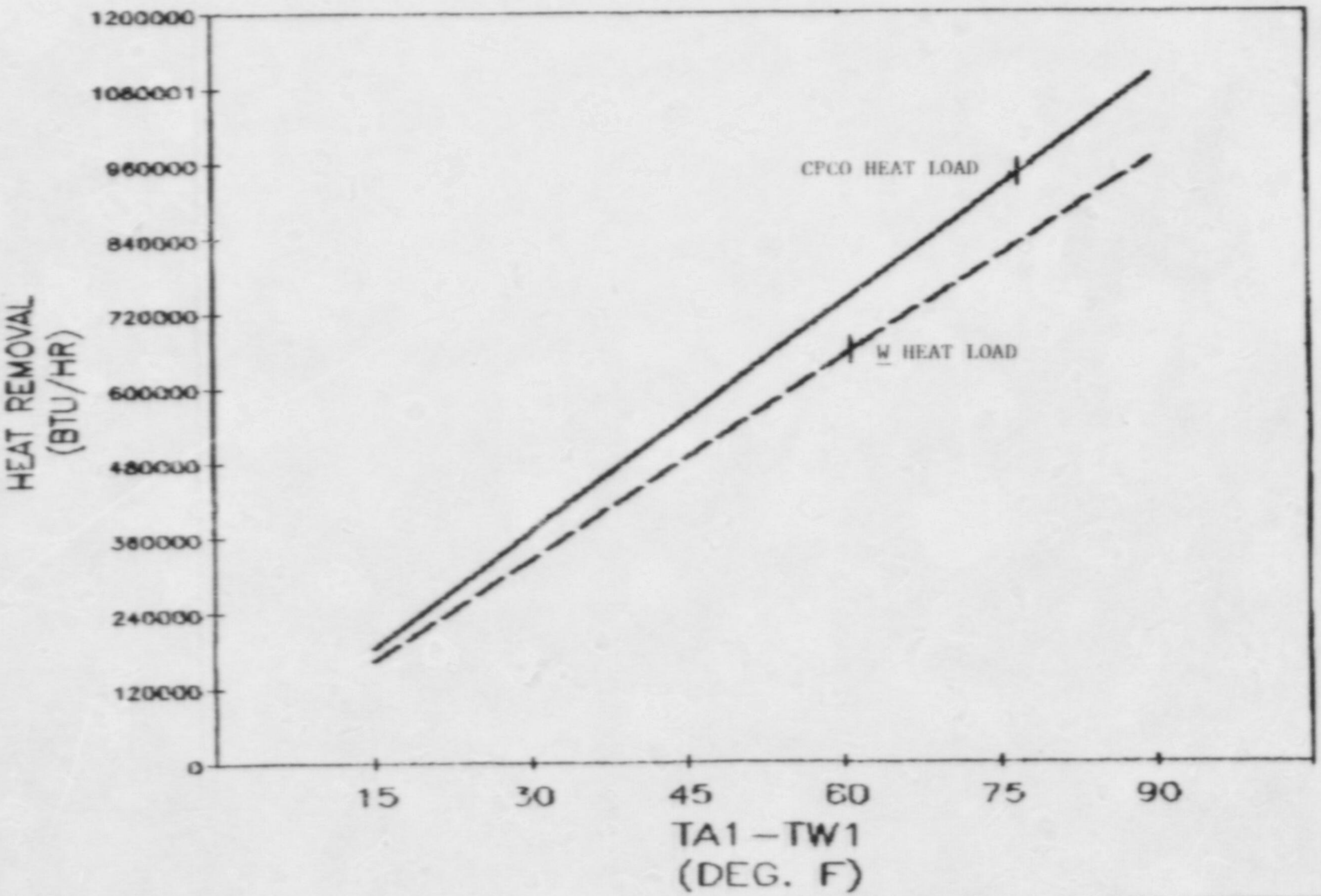


FIGURE 6

ATTACHMENT 3

Consumers Power Company
Palisades Plant
Docket 50-255

JUSTIFICATION FOR USING MIXED AIR
TEMPERATURE AS SINK TEMPERATURE

April 13, 1987

6 Pages

MI0387-0025AA-MM01

ATTACHMENT 3

The analysis by Westinghouse assumes perfect and instantaneous mixing of the air volume to simplify the west engineered safeguards room modeling. The assumption of perfect mixing is not realistic but is considered acceptable for the purpose of the analysis.

The analysis assumes perfect mixing to calculate the mixed air temperature. This temperature was then used to calculate the heat being transferred into the room from certain piping systems. The piping systems are divided into two categories: 1) Type 1 uses the mixed air temperature as the heat sink temperature for both convection and radiation; 2) Type 2 uses the air temperature from the air cooler discharge. Type 2 piping is all piping which has an actual air velocity at the pipe of 150 feet per minute or greater. Type 1 piping is everything else and is assumed to have an air velocity of 150 feet per minute. Type 2 piping uses the air cooler discharge temperature which would be the coolest temperature in the room and is conservative when used to calculate the heat rejected into the room. Use of the mixed air temperature is not as easily visualized. The following discussion applies to the Type 1 piping.

Radiative Heat Transfer

If the wall temperature is less than the mixed air temperature, there is no mechanism for the energy absorbed by the wall (walls, floor and ceiling) to get back into the room air because the quantity (Twall-Tmixed) is negative. Thus, both radiation and convection heat transfer would be from the room air to the walls. This would lead to a net loss of energy from the room air and would lower the room air temperatures. In considering radiation from the hot pipes to the walls, the lower the wall temperature, the larger the amount of heat transferred would be. While this amount of heat could then be conservatively assumed to be transferred to the air in the room rather than to the walls, it should be noted that the heat transfer for this situation requires that the direction of heat flow be out of the room; not vice versa. Conversely, if the wall temperature is larger than the mixed air temperature and the wall temperature is used as the sink temperature, the total radiated energy will be less than if the mixed air temperature is used as the sink temperature. Also, the amount returned to the room will be less than conducted through the wall or stored in the wall.

In summary, using the wall temperature as the heat sink temperature is only sensible if the wall temperature is larger than the mixed air temperature and causes heat to be transferred to the room. Even then, the radiant energy will be less than would obtained if the sink temperature is the mixed air temperature. Also, only a fraction of this is returned to the room since a portion of the heat is conducted through the walls. Consequently, using the mixed air temperature for the sink temperature is conservative for radiation.

Convective Heat Transfer

There were three areas which were reviewed to determinine if the mixed air temperature was an acceptable temperature to use for convective heat transfer at locations away from the cooler exhaust.

1) The air profile in the room

2) Location of the heat sources and heat sinks

3) The sensitivity of the mixed air assumption

Air Flow Profile

Figure 3 provides the room air velocity profile obtained in test T-160 "Safeguards Room Air Velocity Profile Test". The readings have an accuracy of ± 50 fpm. Therefore, velocities below 100 fpm were not recorded and 150 fpm was used in those areas for the purpose of convective heat transfer (conservative assumption). The measurements show the cooler discharge is directed down across the containment spray pump/motor assemblies into the north-west corner of the room. The majority of the air moves in clockwise from the lower Northwest corner (Elev. 570') flowing upward to VHX-27B inlet. As the sketch shows, the majority of the hot piping is in the center of the room. The high pressure safety injection and auxiliary feedwater pumps are located in the region where air velocities are below 100 fpm. The air flow as it moves by the shutdown heat exchanger toward the pumps will be increasing in temperature due to the heat loads from those components. The air will be drawn up across the piping above the pumps because of the fan operation. The distance the air travels before it reaches the majority of piping is greater than the distance after it leaves the piping and enters the cooler. The air also picks up the heat radiated to the walls as it flows along them.

Because of the direction in which the cooler exhaust is blowing, there is a possibility of a low velocity area in the piping off the north-east corner of the cooler. The temperature in this area may be above the mixed air temperature of the room. In general, the piping is located 8 to 12 feet above floor level. The temperature above this piping would be higher than the temperature at floor level. Also, the model for convective heat transfer uses a velocity of no less than 150 fpm. If the velocity were 100 fpm, the heat transfer would be approximately 25-30% less than the amount if 150 fpm were used. From Figure 3, it is seen that the majority of the piping will see 150 fpm or less.

Location of the Heat Sources and Sinks

The heat sources are the motors, piping and lights. The only sink assumed is the air cooler. The fan motor, lighting and piping are located above the pumps. The motors are located in the lower part of the room. One fan can process one room air volume every three minutes. There could be a temperature gradient from the floor to the ceiling, particularly in the lower velocity portions of the room. It would most likely occur in the center of the room where the air flow is low due to the direction of the fan discharge. The temperature in this area could be slightly higher than the mixed air temperature calculated. A temperature difference as high as 18°F (153°-135°) between this area and the surrounding areas cannot be reasonably assumed as the thermal currents would tend to reduce this difference. The air would rise, migrate, and displace cooler air in other locations, generating a natural circulation through the areas with low forced circulation.

Sensitivity of the Mixed Air Assumption

The above paragraphs have shown that the mixed air temperature is reasonable when the location of the heat sources and air flow profile of the room are considered. However, the mixed air temperature assumption also has the effect of applying (Tmixed-Tpipe) to most of the piping (The portion of the piping at cooler discharge sees a larger delta T). In reality, some of the piping will see an air temperature above Tmixed and some piping will see a temperature below Tmixed, as discussed above. To determine the impact of the air temperature applied to the pipes, a sensitivity study was done by Westinghouse. The results of this study are shown in Figures 1 and 2. These figures show that if a temperature 16° below that of Tmixed is used across the piping, the calculated Tmixed only rises to 140°F, still well below 153°F. This would indicate that the sensitivity of the calculated temperature to the mixed air assumption would not be excessive in relationship to the other factors (location of piping in relation to the equipment and conservatively high air flows across the piping).

Figure 1 shows how the mixed air temperature (bulk temperature) varies if temperatures other than the mixed air temperature are used for the heat sink temperature. The abscissa of the graph is the ratio of the air temperature assumed for the heat sink to the calculated mixed air temperature. Figure 2 depicts the change in mixed air temperature versus the difference between the assumed air temperature for the heat transfer and the mixed air temperature.

Some areas away from the cooler may have temperatures less than the assumed bulk air temperature of 135°. Although this causes a higher heat transfer from the pipes, the increased heat transfer from this effect is compensated by the reduced heat transfer that occurs because the actual air flow velocities by these pipes is less than that assumed in the analysis. It is also expected that some pipes would be seeing higher temperatures than the bulk air temperature. The heat transfer from these pipes would be less than calculated not only because the air flow is less than that assumed in the calculation, but because the delta T between the air and the pipe will be less than that assumed in the calculation.

Conclusion

The location of the hot piping, the direction of the air cooler exhaust, and the velocities of the air on Type 1 piping, all support the use of the mixed air temperature as the heat sink temperature. This assumption will result in a conservative temperature profile to evaluate the qualification of the equipment. The amount of heat transferred to the room is sensitive to the heat sink temperature; however, using a temperature 16°F less than the mixed air temperature for heat transfer from Type 1 piping only, results in a 5°F change in mixed air temperature. Therefore, it can be concluded that the heat rejected to the room is not very sensitive to changes in air temperature at the Type 1 piping.

FIGURE 1:

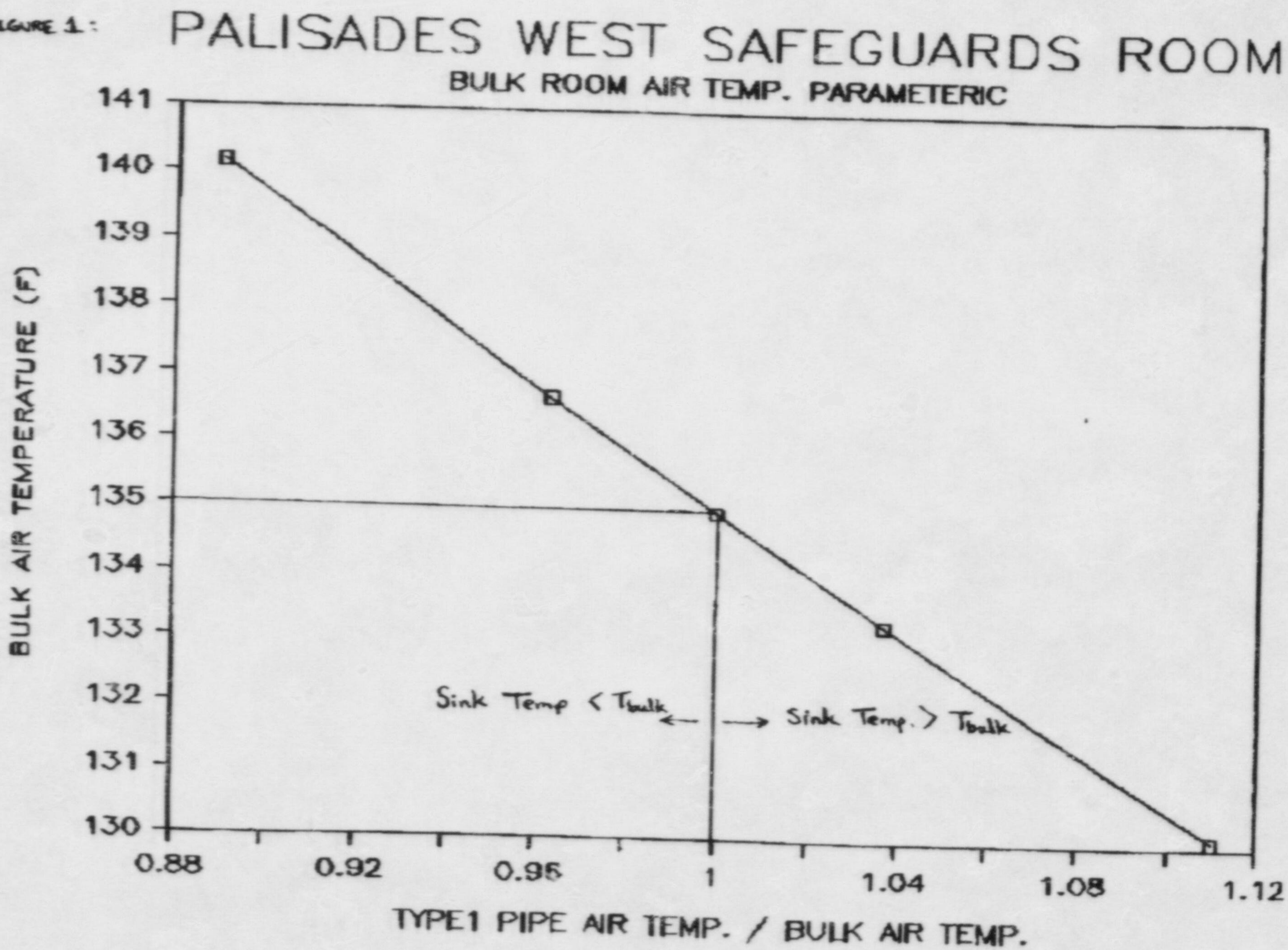

PALISADES WEST SAFEGUARDS ROOM
BULK ROOM AIR TEMP. PARAMETERIC
BULK AIR TEMPERATURE (F)
141
140
139
138
137
136
135
134
133
132
131
130
Sink Temp < Tbulk
Sink Temp. > Tbulk
0.88
0.92
0.96
1
1.04
1.08
1.12
TYPE1 PIPE AIR TEMP. / BULK AIR TEMP.

Figure 2:

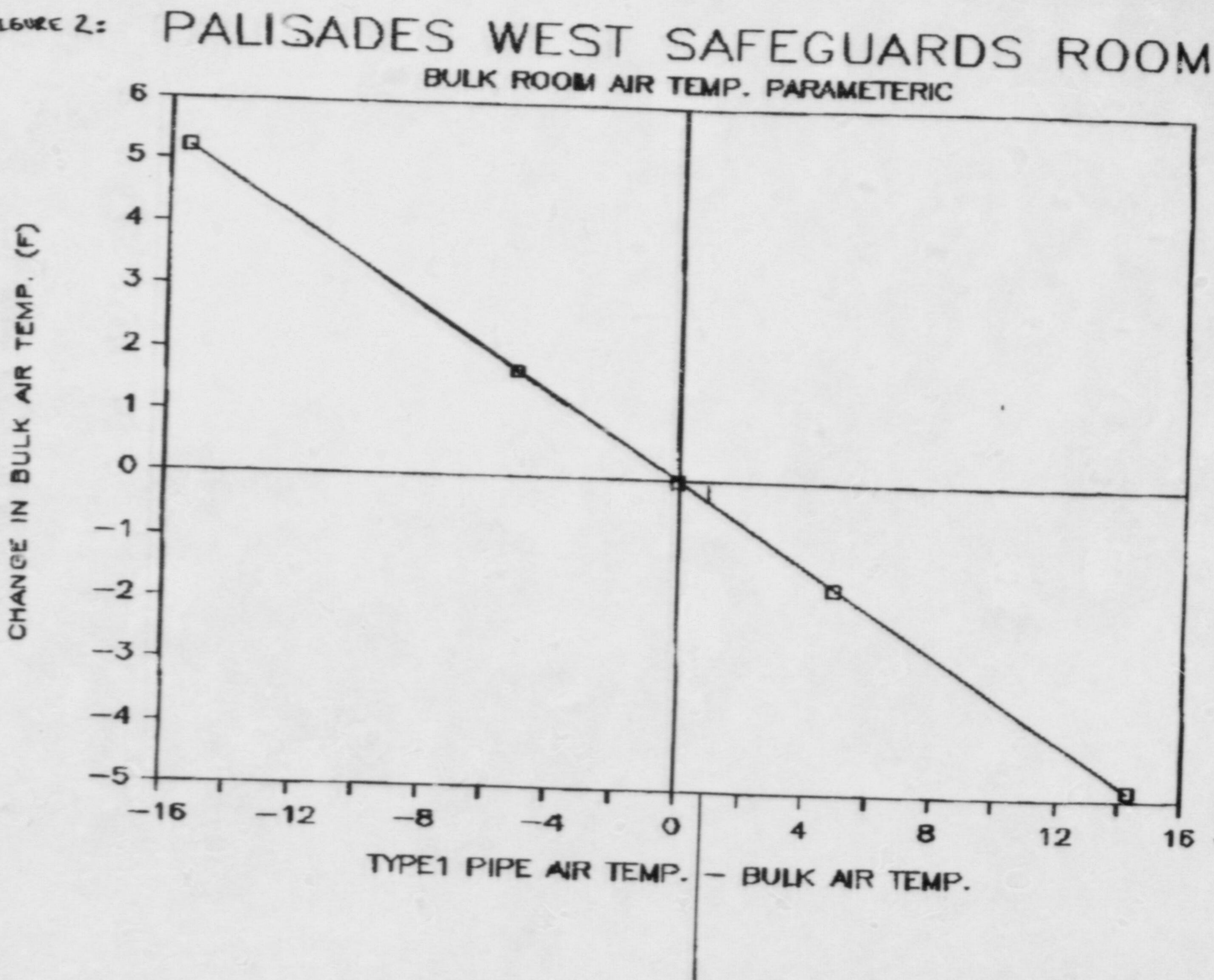
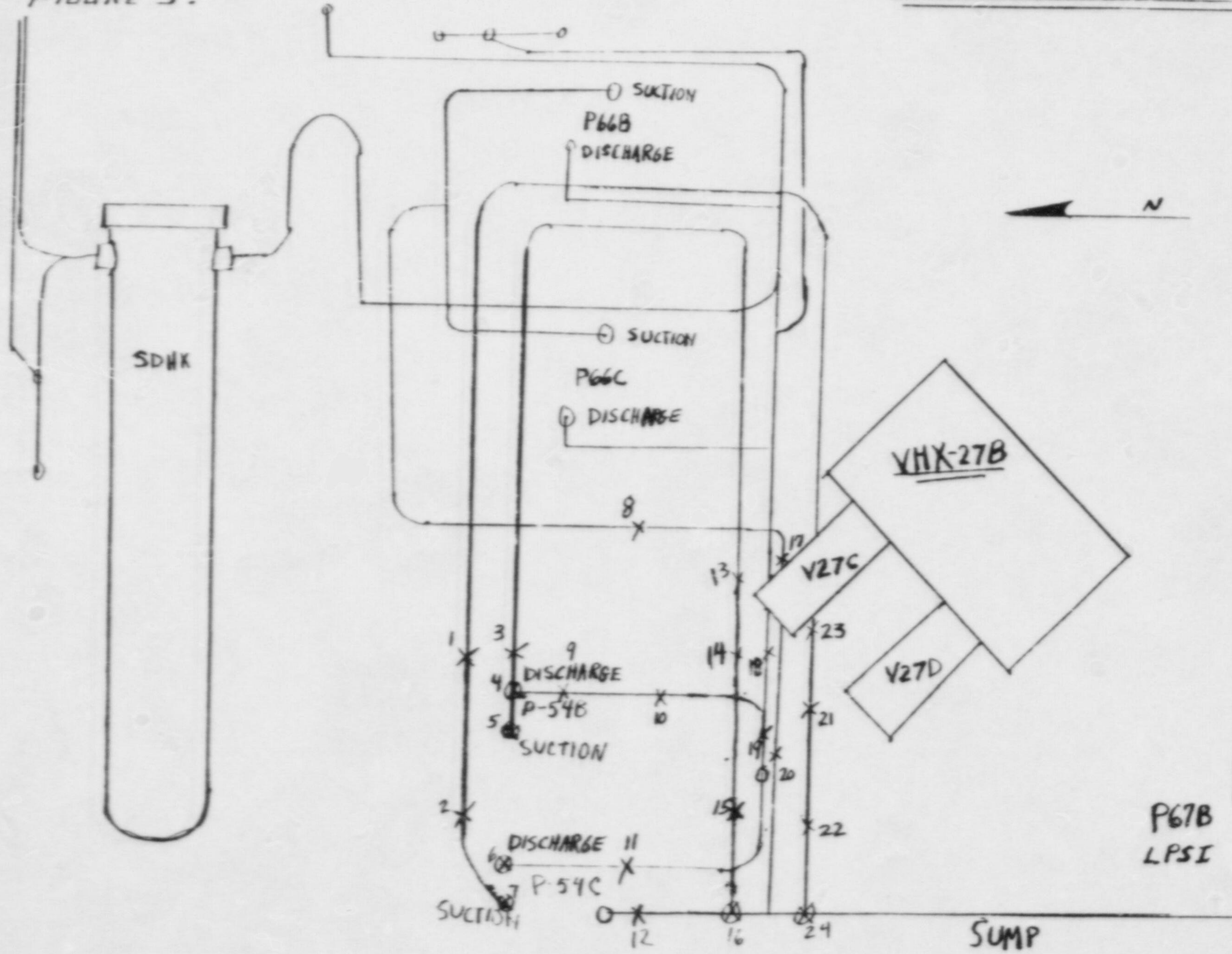

PALISADES WEST SAFEGUARDS ROOM
BULK ROOM AIR TEMP. PARAMETERIC
CHANGE IN BULK AIR TEMP. (F)
6
5
4
3
2
1
0
−1
−2
−3
−4
−5
−16
−12
−8
−4
0
4
8
12
16
TYPE1 PIPE AIR TEMP. − BULK AIR TEMP.

FIGURE 3:

WEST ENGINEERED SAFEGUARDS ROOM



POINT	VELOCITY
1	150
2	500
3	150
4	450
5	1200
6	275
7	175
8	100
9	250
10	1200
11	100
12	100
13	200
14	2000
15	500
16	100
17	100
18	100
19	250
20	175
21	1000
22	100
23	100
24	100

ATTACHMENT 4

Consumers Power Company
Palisades Plant
Docket 50-255

IMPACT ON EQUIPMENT QUALIFICATION

April 13, 1987

2 Pages

ATTACHMENT 4

Limiting Components

The equipment in the west engineered safeguards room was reviewed to determine whether it could be qualified to the following profile:

1) 24 hours at 153°F
2) then 24 hours at 140°F
3) then 28 days at 135°F

Based on this review, all components could be qualified with the above profile. The limiting components were the pump and fan motors and the converter for valve CV-3025. The room temperature profile is well below the above. The room mixed air reaches a peak temperature of 135°F and decreases as sump temperature decreases. A discussion of the sensitive components follow. The valve motors and solenoid valves were qualified for inside containment use, which is greater than 283°F. The other components were acceptable for temperatures much higher than 153°F.

Containment Spray Pump Motors

These motors are in the direct path of the air flow from the cooler. The flow exits the cooler at approximately 92°F and crosses two pipes which are at 226°F before hitting the pump motors. The velocity of the air across these pipes was measured at 2000 and 250 feet per minute. A heat transfer calculation was done to estimate the air temperature. Heat losses due to convection only were assumed. The air temperature hitting the motor was calculated to be 93°F. This is significantly less than the qualification temperature.

Room Cooler Fan Motor

This motor is in the air flow stream inside the duct. The motor will see the cooler discharge temperature which is about 90°F.

High Pressure Safety Injection Pump Motor

The ESR cooler fans direct flow downward across hot pipes and onto the containment spray pumps. The HPSI pump is located to the side of and slightly behind the cooler. Tests have shown that there is some air velocity near the motor, though it is small. This air movement will help carry any warm air created by the motor upward toward the coolers intake. The warm air will be replaced by the air from the cooler discharge which is at a lower temperature than the room bulk air temperature. Natural convection of any warm air created will also tend to create an upward air flow away from the motor.

High Pressure Safety Injection Pump Motor (cont.)

From an elevation standpoint, the hottest air will be above the pumps where most of the hot piping is located. The pumps, located on the cold floor, will see low air temperature since there is no heat source under the floor. Therefore, it can be concluded that the HPSI motor will see surrounding air temperatures that are no higher than, and most likely lower than, the room bulk air temperature.

Electro - Pneumatic Converter for CV-3025

This valve is normally closed and is opened only during normal shutdown cooling to direct reactor coolant from the shutdown coolers back to the reactor coolant system. It is therefore not opened during a LOCA. In the closed position, it directs flow from the shutdown coolers to the containment spray. In the accident scenario of concern, instrument air is lost causing the subject valve to fail (or remain) closed. Since this is its desired position for the accident, operation of its associated converter is immaterial.

Conclusion

The electrical equipment will be maintained below their qualification temperature throughout the event.

ATTACHMENT 5

Consumers Power Company
Palisades Plant
Docket 50-255

HISTORICAL DATA ON
SERVICE WATER TEMPERATURES

April 13, 1987

6 Pages

ATTACHMENT 5

Figure 1 is a cumulative time history of service water temperature readings since 1982 for the summer months when the lake temperature is highest. This chart uses the maximum temperature for a given day for an overall "maximum" profile.

Figure 2 presents more specific data for 1983 when temperatures exceeded 80°F twice. The data points are by shift (A, B, and C).

Table 1 contains selected temperature data by shift to illustrate the transient nature of the lake temperature. Specific days were chosen which had temperatures of 75°F or higher. The table provides the data for the shifts before the peak temperature was reached, showing the warming trend, and the data for shifts subsequent to the peak temperature being reached, showing the cooling trend.

As can be seen, the temperatures have only exceeded 80°F twice; once for one shift and a second time for two shifts. Data for five years prior to 1982 has also been reviewed and no temperatures exceeded 80°F in those years. Thus the enclosed figures and table represent those times 80°F has been exceeded for 10 years. The pre-1982 data is not consecutive due to data retrieval problems.

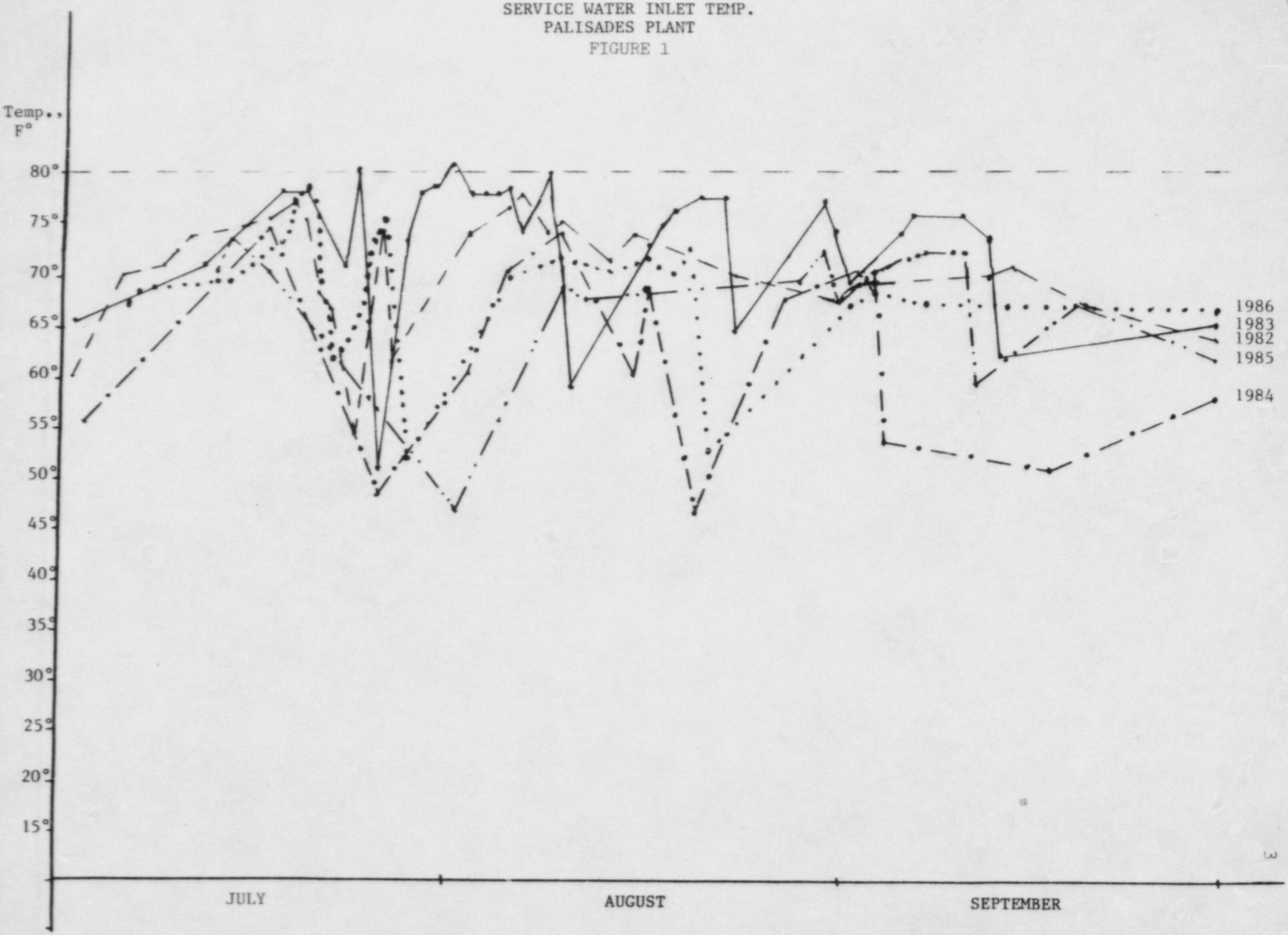
The occurrence of elevated lake temperatures begins in June when the surface waters become warm and a distinct upper layer is formed. Prior to this, the water is isothermal from top to bottom, with a temperature near 39.2°F (the temperature of maximum density). By about mid-July, an upper layer is established about 60 feet deep at about 68°F. A sharp thermocline separates the upper mass from the lower mass extending from a depth of about 75 feet to the lake bottom. The temperature of the lower layer is close to 39.2°F. This stratification exists until late September.

During the summer stratification, the warmer surface waters only rarely intermix with the cooler, deeper layer at the 40 foot depth contour of the Palisades intake. Stratification is, however, sometimes distorted by the effect of strong winds. During the summer, strong westerly winds from a westerly direction drift warmer upper water layers eastward toward the shore, thereby, increasing the depth of the warmer, upper layer near the shore, such as at Palisades. Conversely, winds from the opposite direction (north thru the southeast) tend to drift the warmer surface water away from the eastern shore, bringing the colder deep water to the surface near the eastern shore, so that significant surface temperature drops occur. The change in wind direction and reduced wind velocities tend to minimize the duration that warmer upper surface temperatures would be observed at the plant intake. The wide changes in temperatures shown in Figures 1 and 2 are the result of changes in these meteorological conditions.

The lake temperature, as determined by our review of the available data, rarely exceeds 80°F and only for very short durations. The longest time period recorded is conservatively assumed as 16 hours from C-shift on July 31, 1983 to B-shift on August 1, 1983. The temperature rose to 81°F during this period.

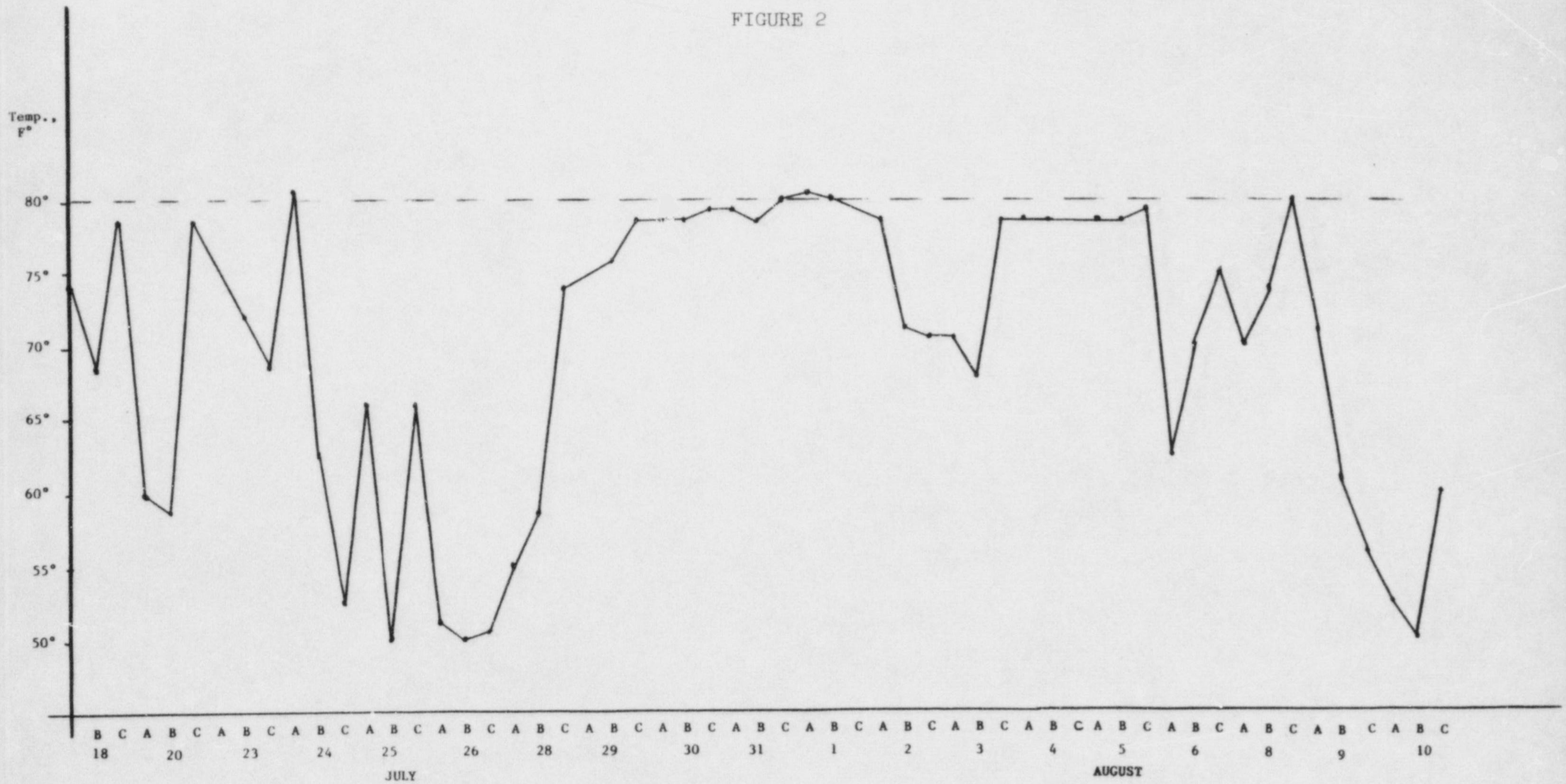
Because the length of time the lake temperature has exceeded 80°F is of a relatively short duration, a Technical Specification Action requirement would not have been completed before the temperature returned to the specified limit of 80°F. The Technical Specification 3.0.3 action statement requires that within one hour, action be initiated to go to hot standby which shall be completed in the next six hours. Hot shutdown is required in the following six hours, and cold shutdown in the subsequent 24 hours for a total of 37 hours to complete the action. This is more than twice the time of the longest assumed 16 hour period in 1983 when the lake temperature exceeded 80°F. Because temperatures exceeding 80°F are of a transient nature, and since the probability of exceeding 80°F is low: 5.5 E-04 events/year; Consumers Power Company concludes that a Technical Specification for the service water inlet temperature is unnecessary. The probability is based on assuming the temperatures exceeded 80°F for three 8-hour shifts during the 5-years in which data is given in Table 1. The probability would be halved if the five additional years of data were used.

SERVICE WATER INLET TEMP.
PALISADES PLANT
FIGURE 1



SERVICE WATER INLET TEMP.
PALISADES PLANT
1983

FIGURE 2



SERVICE WATER INLET TEMPERATURE – °F

PALISADES PLANT
TABLE 1

1982

A	B	C		
71	72	72	-	7/09
72	73	73	-	7/10
74	74	--	-	7/11
74	74	74	-	7/12
--	74	74	-	7/13
74	74	74	-	7/14
74	75	75	-	7/15
75	75	76	-	7/16
76	75	76	-	7/17
76	75	76	-	7/18
77	--	--	-	7/19
76	76	74	-	7/20
68	62	--	-	7/21
64	--	75	-	7/26
62	61	63	-	7/27
61	72	74	-	8/01
63	75	75	-	8/02
--	--	66	-	8/03
--	78	78	-	8/04
--	76	77	-	8/05
66	78	73	-	8/06
61	60	60	-	8/07
58	75	75	-	8/08
74	75	76	-	8/09
75	67	72	-	8/10
72	72	73	-	8/13
--	72	--	-	8/14
73	74	75	-	8/15
73	74	75	-	8/16
64	74	62	-	8/17
--	52	68	-	8/18

1983

A	B	C		
74	68	78	-	7/18
70	70	70	-	7/19
60	58	78	-	7/20
53	75	74	-	7/21
70	50	--	-	7/22
--	72	68	-	7/23
81	63	53	-	7/24
55	58	74	-	7/28
--	76	78	-	7/29
78	78	79	-	7/30
79	78	80	-	7/31
81	80	--	-	8/01
78	72	71	-	8/02
71	67	78	-	8/03
78	78	--	-	8/04
78	78	79	-	8/05
63	70	75	-	8/06
70	76	67	-	8/07
70	74	80	-	8/08
72	61	56	-	8/09
74	74	74	-	8/16
75	75	76	-	8/17
76	76	77	-	8/18
76	--	--	-	8/19
78	77	--	-	8/20
74	74	--	-	8/21
77	78	78	-	8/22
--	66	66	-	8/23
74	71	75	-	8/29
70	75	78	-	8/30
76	73	73	-	8/31
69	60	71	-	9/01
73	74	74	-	9/04

SERVICE WATER INLET TEMPERATURE - °F
PALISADES PLANT
TABLE 1 Cont'd

1983

A	B	C		
75	76	76	-	9/05
77	77	76	-	9/06
77	76	76	-	9/07
75	75	76	-	9/08
75	75	75	-	9/09
77	76	77	-	9/10
76	76	76	-	9/11
75	74	74	-	9/12
64	50	53	-	9/13

1984

A	B	C		
--	--	72	-	7/16
72	74	75	-	7/17
70	73	73	-	7/18
71	--	--	-	7/22
--	74	--	-	7/23
48	--	--	-	7/25
74	--	75	-	8/09
75	--	--	-	8/10
44	42	45	-	8/12

1985

No data greater than 75

1986

A	B	C		
73	74	--	-	7/18
76	76	77	-	7/19
78	77	74	-	7/20
70	54	55	-	7/21
70	73	73	-	7/25
76	61	70	-	7/26
65	52	53	-	7/27