C-2289

STP 3053 (10/91)
OEP-3.070

SOUTH TEXAS PROJECT ELECTRIC GENERATING STATION
HOUSTON LIGHTING & POWER COMPANY

CALCULATION COVER SHEET

CALC NO.	MC-6412
PRELIM.	
FINAL	✓
VOID	

BUILDING/AREA/SYSTEMS: CH UNIT: 1 & 2

SUBJECT: ESSENTIAL CHILLED WATER LOAD DISCIPLINE: MECH

QUALITY CLASS: Q

OBJECTIVE

SEE SHT. 7

SCOPE

SEE SHT. 7

SUMMARY OF RESULTS

SEE SHTS. 8 → 14

TOTAL NO. OF SHEETS

431 plus 4 diskettes

REV NO.	0						
PREPARER	[signature]						
REVIEWER	[signature]						
SE	[signature]						
DM	[signature]						
ISSUE DATE	10-14-93						

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EAB supply header flows & battery supply header flows are taken from Appendix H.

Train	Supply Header Flow, cfm	Battery Header Flow, cfm
A	34447	3425
B	26688	2046
C	37884	2443

Supply Fan Flow = (34447+26688+37884)/2 = 49510 cfm

Makeup flow = (3425+2046+2443)/2 = 3957 cfm

Return fan flow = 49510-3957 = 45553 cfm

Because train C supply fan flow is greater than C header supply flow, 49510-37884 = 11626 cfm of fan C discharge mixes with fan B discharge (supply header cross train flow).

The recirculated portion of train C header = 37884-2443 = 35441 cfm.

Because the return fan flow is higher than 35441 cfm, the flow from train B to train C return fan = 45553-35441 = 10112 cfm

Because all of the C header flow is from the C coil outlet, the C header temp. = 58.21 °F

The sum of all emergency (battery backed) lights in the train C area is 2592 watts x 3.413 watts/btu/hr ÷ 12000 btu/ton-hr = 0.737 tons (from Appendix I)

However, since non-1E power is not lost, the emergency lighting does not come on. Therefore the "use factor" = 0

Values for class 1E equipment, class 1E cables, normal lighting, non-1E equipment, and non-1E cables are likewise from Appendix I.

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The use factor for non-1E cables is less than 1, based on the re-evaluation of non-1E equipment in DCN EC-73 of Ref. 7. Before the DCN, the total non-1E equipment load was 94720 watts (from Appendix J). After the DCN, the total non-1E equipment load is 70964 watts (from Appendix I). The ratio is

70964/94720 = .749

Therefore, it is reasonable and conservative to apply a use factor of 0.8 to the train C non-1E cable load.

The total train C area heat load = 0.74*0 + 17.52*1 + 12.00*1 + 10.12*1 + 20.19*1 + 7.55*0.8 = 65.86 tons

Train C return header temp.

$$= 58.21 + \frac{65.86 \text{ tons} * 12000 \text{ btu/ton}\cdot\text{hr}}{1.09 * 35441} = 78.67 \text{ °F}$$

The train B return fan receives a mixture of trains A & B return air for an average temperature of 69.56 °F. The B return header and B return fan inlet ducts are very closely in-line, so the A & B flow would actually mix before the flows split to the B fan inlet & C fan inlet.

$$\text{C train return fan inlet} = \frac{78.67*35441 + 69.56*10112}{(35441 + 10112)} = 76.56 \text{ °F}$$

The heat added by the return fan = 15.77 tons (from App. F).

The heat added by the supply fan = 36.242 tons (" " ")

$$\text{Delta T across the return fan} = \frac{15.77 \text{ tons}*12000 \text{ btu/ton}\cdot\text{hr}}{1.09*45553}$$

$$= 3.81 \text{ °F}$$

$$\text{Temperature at supply fan inlet} = \frac{45441*(76.65+3.81) + 3957*95}{(45441+3957)}$$

$$= 81.62 \text{ °F}$$

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Delta T across supply fan = $\frac{36.242 \text{ tons} * 12000 \text{ btu/ton·hr}}{1.09*49510}$

= 8.06 °F

Cooling coil inlet temperature = 81.62 + 8.06 = 89.68 °F

The latent load = $\frac{4840 * 3957 * (139 - 66)}{7000 \text{ grains/lb} * 12000 \text{ btu/ton·hr}}$ = 16.64 tons

The coil "thermal effectiveness" is based on sensible heat transfer. The majority of condensation will occur on the coilder coils, which increase the water temperature seen by the rest of the coils. We assume the chilled water inlet temperature available for sensible heat transfer is actually:

$$52 + \frac{16.64 \text{ tons} * 12000 \text{ btu/ton·hr}}{500 \text{ lb/hr·gpm} * 600 \text{ gpm} * 1.0 \text{ btu/lb·°F}} = 52.67 \text{ °F}$$

The coil effectiveness, ϵ = 0.8503 (App. B)

Neglecting correction to scfm,

Cmin = Cair = C_p * lb/hr air

lb/hr air = 0.075 lb/ft^3 * 60 min/hr * cfm (ft^3/min)

C_p of air varys slightly with moisture content. Use C_p = 0.242 as representative of conditions in EAB and other conditioned spaces.

Cmin = 0.242*0.075*60*cfm = 1.09*cfm

Q = Cmin*ϵ*(Tair in - Tchwtr in) btu/hr

also Q = Cair * (Tair in - Tair out)

or Tair out = Tair in - Q/Cair

= 89.68 - .8503*(89.68 - 52.67) = 58.21 °F

If we had not made such a good guess to start with, the we would iterate the process with this value as the new trial coil outlet temperature.

The sensible cooling load

= 1.09*49510*(89.68-58.21)/12000 = 141.53 tons

Total coil load = sensible load+latent load

= 141.53 + 16.64 = 158.17 tons

[Note: the last digit can differ because the spreadsheet retains more significant digits than is printed or is used in this hand calculation.]

2. CONTROL ROOM ENVELOPE

Makeup flow = 1105 cfm from Appendix H.

Makeup fan heat into air = 0.8058 tons from Appendix F.

Heaters for makeup filter units = 4.5 KW from Ref. 5.

Air temperature at filter outlet

$$= 95 + \frac{0.8058*12000}{1.09*1105} + \frac{4.5\ KW * 3413\ btu/KW \cdot hr}{1.09*1105} = 115.8\ ^\circ F$$

This is not recorded on the spreadsheet because the air mixes before being introduced into the separate trains.

$$Avg.\ Temp. = 95 + \frac{(0.8058+0.7766)*12000}{1.09*(1105+1047)} + \frac{(4.5+4.5)*3413}{1.09*(11.05+1047)} = 116.19\ ^\circ F$$

$$Makeup\ to\ train = \frac{1047 + 1105}{2} = 1076\ cfm$$

Cleanup fan flow = 5912 cfm from Appendix H

Return fan flow = 16132 cfm from Appendix H

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Use a trial return air temperature of 65.38 °F

Return fan heat input to air stream = 6.5688 tons (App. F)

Return fan outlet temperature $= 65.38 + \dfrac{6.5688*12000}{1.09*16132} = 69.86$ °F

Cleanup fan inlet temperature

$$= \frac{69.86*(5912-1076) + 116.19*1076}{5912} = 78.29 \text{ °F}$$

Cleanup fan heat to air = 3.0141 tons   from App. F.

Cleanup fan outlet temp. $= 78.29 + \dfrac{3.0141*12000}{1.09*5912} = 83.90$ °F

Coil inlet flow = 16132 + 1076 = 17208 cfm

Coil inlet temp. $= \dfrac{5912*83.90 + (17208-5912)*69.86}{17208} = 74.68$ °F

Cair = 1.09*17208 = 18757 btu/hr•F

Coil effectiveness for sensible heat transfer = 0.9478 (App. B)

Latent load from personnel = 0.4 tons (Ref. 13)

Latent load $= \dfrac{4840*1076*(139-66)}{7000*12000} + 0.4 = 4.93$ tons

Note: The 66 grain humidity was based on expected EAB coil outlet conditions. The same value is used for other coils, including the control room coil, for convenience. At 54 °DB & 53 °WB, the outlet humidity is 58 grains. Thus the latent load is underestimated by about 0.5 tons.

The personnel latent load should be divided by the number of operating trains, or 2 in this case.

The combined error is 0.3 tons, which is not significant and which will be offset in other coils.

Effective chill water inlet temp. $= 52 + \frac{4.93*12000}{500*168} = 52.70\ ^\circ F$

Sensible load $= Cmin * \epsilon * (74.68-52.70)$

$= \frac{18757 * 0.9478 * (74.68-52.70)}{12000} = 32.56$ tons

Coil outlet temp. $= 74.68 - \frac{32.56*12000}{18757} = 53.85\ ^\circ F$

Supply fan heat to air = 7.257 tons (App. F)

Supply fan outlet temp. $= 53.85 + \frac{7.257*12000}{1.09*17208} = 58.49\ ^\circ F$

The flow from A, B, & C trains are mixed for the supply header temperature.

$= \frac{0.0*0.0 + 18486*58.64 + 17208*58.49}{(0 + 18486 + 17208)} = 58.57\ ^\circ F$

Electrical loads are from Appendix I, except for the Reheater load. This replaces the non-1E equipment load. The I & C equipment loads are not easily separated into class 1E and non-class 1E portions, so the entire load has been considered class 1E. The reheater load is included in its place. Three operable reheaters are installed in the control room supply headers, for a maximum load of:

$= \frac{(43 \text{ kw} + 60 \text{ kw} + 5 \text{ kw})*3413 \text{ btu/kw}\cdot\text{hr}}{12000} = 30.72$ tons

These are de-energized by an SI signal, so the use factor for steady state analysis is 0.

Personnel sensible load = 0.5 tons (per Ref. 13).

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The cleanup and supply fans are centrifugal, so motor heat is rejected into the equipment rooms. This load is lumped into the total load, which is easier than trying to separate these loads into the correct coil. The loads are similar so the error is small. It is conservative because train C is the limiting train and the actual train C fan motor heat loads are less than average.

The total header sensible load

$$= 19.33 + 0.5 + .818 + 0.771 + 0.338 + 0.320 = 22.08 \text{ tons}$$

Total header flow = 18486 + 17208 = 35694 cfm

$$\text{Return header temp.} = 58.57 + \frac{22.08*12000}{1.09*35694} = 65.38\ ^\circ F$$

This equals the trial temperature. Otherwise, the calculated temperature would be the new trial temperature, and so on.

Total coil load = 32.56 tons + 4.93 tons = 37.49 tons

3. CCW / ESSENTIAL CHILLER ROOM

Train C, MAB Room #067F

CCW flow after SI ≈ 13000 gpm (Ref. 35)

CCW pump bhp @ 13000 gpm = 780 hp (Ref. 33)

Motor efficiency = 0.943 (Ref. 34)

$$\text{Motor heat to room} = 780 \text{ hp} * \frac{(1-0.943)}{0.943} * 746 \text{ watts/hp}$$

$$= 35200 \text{ watts}$$

Essential Chilled Water Pump flow = 954 gpm (Ref. 38)

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Essential Chilled Water Pump bhp @ 954 gpm = 50 hp (Ref. 36)

Essential Chilled Water Pump motor efficiency = 0.937 (Ref. 37)

Motor heat to room $= 50 \text{ hp} * \frac{(1-0.937)}{(0.937)} * 746 \text{ watts/hp}$

$= 2508$ watts

Note: Higher values were used for both CCW pump motor and Chilled water pump motor based on preliminary estimates (39305 watts & 4610 watts respectively). The higher values have been retained because they are conservative and have little affect on the results.

The CCW Supplemental Cooler contains 3 fans, each with 1.5 bhp.

Motor efficiency = 0.80 (Ref. 32F)

Heat to room $= 3 * \frac{1.5 \text{ hp}}{0.80} * 746 \text{ watts/hp} = 4196$ watts

Chiller Area Supplemental Cooler

1 fan @ 3.7 bhp, motor eff. = 0.829 (Ref. 32H)

Heat to room $= \frac{3.7 \text{ hp} * 746 \text{ watts/hp}}{0.829} = 3330$

300 ton chiller compressor maximum input = 354 KW (Ref. 57)

Motor efficiency = 0.946 (Ref. 37)

Heat to room = 354 KW * (1-0.946) * 1000 watts/KW = 19116 watts

The chiller control panel would also release heat, say 1000 watts

Total = 20116 ≈ 20156 watts used from preliminary input

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The 150 ton essential chiller is hermetically sealed, so heat losses from the motor are removed by refrigerate to the condenser. The chiller does have a control panel and lube oil heater (when idle), so a small heat load is present.

Preliminary input used 1500 watts with a "use factor" of 0.8, which looks reasonable for the control panel and oil heater.

Lighting, electrical equipment, & cable loads are from Ref. 11.

Conduction load is from Ref. 15. (0 in this case)

The use factor for conduction is taken as 0 because normal MAB cooling is not lost in this scenario, so adjacent rooms will not heat up.

Piping heat load is from Ref. 15.

The total heat load in the room

= 180*0 +0*1 +1503*1 +2116*1 +0*1 +2790*1 +0*0 +0*0 +39305*1 +4610*1 +20156*1 +1500*0.8 +4196*1 +3330*1 = 79306 watts

Room vent flow = 1300 cfm   This is design flow from Ref. 14. Comparable Unit 2 spreadsheets used MAB final air balance data, which was close to the design value.

Vent temp. = 65 °F (Ref. 22)

Vent constant = 1.09*1300 cfm = 1417

Air flow to CCW supplemental cooler = 25748 cfm (Ref. 30)

Design ECW flow = 36 gpm (Ref. 32F)

ECW temperature of 100 °F is 1 degree higher than Tech Spec limit for ECP temperature. This scenario is within the first 30 minutes following a LOCA and ECW discharged to the ECP takes much longer to circulate back to the intake.

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The thermal effectiveness of the CCW supplemental coil is ϵ=0.8237 from Appendix B.

The AHU constant is the lower of 500*gpm*ϵ or 1.09*cfm*ϵ

500*36 = 18000 < 1.09*25748 = 28065

therefore AHU constant = 500*36*0.8237 = 14827

The chilled water AHU air flow = 3491 cfm (Ref. 30)

Chilled water design flow = 40 gpm (Ref. 32H)

Chilled water inlet temperature = 52 °F from case input

Thermal effectiveness of coil = 0.7778 from Appendix B

AHU constant = C_2 = 1.09 * 3491 * 0.7778 = 2960

Note: Only in the ESF Pump Room and CCW Supplemental Coolers does the cooling water have a chance to set Cmin. Most other AHU in the spreadsheet do have the decision process to select Cmin.

Latent load ≈ 0 because conditioned air is supplied from the MAB system.

$$T_{room} = \frac{79206 \text{ watts}*3.413 \text{ btu/watt.hr} + 14827*100 + 2960*52 + 1417*65}{(14827 + 2960 + 1417)}$$

T_{room} = 104.1 °F

$$\text{Chilled water sensible load} = \frac{2960*(104.1 - 52)}{12000 \text{ btu/ton.hr}} = 12.85 \text{ tons}$$

Total chilled water coil load = 12.85 + 0 = 12.85 tons

Note: The two coolers in this room act together. The CCW supplemental cooler basically keeps room temperature slightly higher than ECW inlet temperature, while the load on the chilled water coil is determined by the room temperature. The actual room heat loads have very little affect on the load to the chilled water train.

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4. ELECTRICAL PENETRATION ROOM, Train C, EAB 301

Electrical heat loads are from Ref. 7.

The normal AHU is running per the scenario.

6.3 hp * 746 watts/hp * 1/0.672 = 6994 watts (Ref. 26)

Essential AHU load to air

8.2 hp * 746 watts/hp * 1/0.838 = 7300 watts (Ref. 3)

Conduction load = 0 per assumptions 6 & 7.

Piping load = 0 per Ref. 12.

Total room load = 50676 watts

Vent flow = 830 cfm per design

Vent temp. = 65 °F (Ref. 22)

Room vent constant = 1.09 * 830 = 904.7

Normal AHU fan flow is assumed to 0 because the normal and emergency AHU fans share common ductwork and the emergency fan has considerably higher static pressure. Therefore the normal fan may be "shut-off" and be unable to move air.

Emergency AHU flow = 7241 cfm (Ref. 30)

Design chill water flow to coil = 78 gpm (Ref. 32F)

Coil effectiveness = 0.8006 per Appendix B

Coil const. = 1.09 *07451 * 0.8006 = 6319

Latent load = 0

$$T_{room} = \frac{50767*30413 + 6319*52 + 904.7*65}{(6319 + 904.7)} = 77.6\ ^\circ F$$

load to chilled water = 6319 * (77.6 - 52)/12000 = 13.47 tons

5. ESF PUMP ROOM, TRAIN C, FHB 004

Electrical loads are taken from Ref. 10

Conduction load = 0 (Ref. 18)

Note: Trains A & B have a steady state conduction load for the spent fuel pit, which is above the Train A room and partially above the Train B room.

Conduction = 2376.2 b/hr with SFP @ 125 °F and room 104 (Train A), per Ref. 18

Consideration of initial transient loads indicate a reduced initial room temperature is necessary if only the 300 ton chiller is available. It is conservative to use this 75 °F room temperature for conduction load in all cases for this room.

Room FHB 006, Train A ESF Room Conduction

$$= 2376.2 \text{ b/h} * \frac{(125 - 75)}{(125-104)} * \frac{1}{3.413} \frac{\text{watts}}{\text{b/h}} = 1658 \text{ watts}$$

Similarly, for FHB 005, conduction

$$= 1139.2 * \frac{(125 - 75)}{(125 - 104)} * \frac{1}{3.413} = 795 \text{ watts}$$

Piping heat load = 136012 b/h (Ref. 18)

= 136012 * 1/3.413 = 39851 watts

Note: The piping heat load was calculated assuming a room temperature of 120 °F, whereas the steady state room temperature is much less than 120 °F. Thus this portion of the load is underestimated by 3 to 4 tons. Overall the transient load is limiting by more than this margin, so this calc. will not be iterated to reflect actual room temperature.

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HHSI pump, Train C

BHP near runout = 880 hp (Ref. 8)

Motor efficiency = 0.949 (Ref.)

heat load to room = 880 hp * 746 watts/hp * $\frac{(1-0.949)}{0.949}$

= 35280 watts

LHSI pump

BHP near runout = 405 hp (Ref. 8)

Motor eff. = 0.931 (Ref. 8)

heat load to room = 405 * 746 * $\frac{(1-0.931)}{0.931}$ = 22392 watts

The Containment Spray pump and motor are identical to the LHSI pump.

Air Handling Unit has 2 fans @ 3.5 bhp each

Motor eff. = 0.82 (Ref. 32F)

load to room = 2 * 3.5 hp * 746 watts/hp * 1 / 0.82

= 6358 watts

Net room load = 133187 watts

Room vent flow = 1208 cfm (Ref. 29)

Inlet air temperature = 95 °F from outside

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Note: While non-1E power is still available, the SI signal opens the FHB inlet bypass dampers, which closes the normal inlet dampers and trips the supply fans.

Room vent const. = 1.09 * 1208 = 1317

AHU air flow = 34050 cfm (Ref. 29)

Chill water flow (design) = 62.5 gpm (Ref. 32F)

Chill water temperature = 52 °F per initial input

Note: Chill water flow to this AHU is changed in a subsequent section. A lower flow is desirable to minimize the transient load immediately after starting the AHU. The higher flow is conservative for steady state purposes.

Coil effectiveness = 0.8258 from Appendix B

C_1 = Cmin * ϵ, Cmin = lesser of Cair or Cwater

Cair = 1.09*34050 = 37115
Cwater = 500*62.5 = 31250
Use Cwater

C_1 = 31250 * 0.8258 = 25806

Latent heat load = $\frac{4840 * 1208 \text{ cfm} * (139-66)}{7000 \text{ grains/lb}}$ = 60970 b/h

Note: This assumes the coil outlet humidity obtained from 58 °F Dry Bulb & 57 °F Wet Bulb for the EAB coils. The outlet temperature for these coils will be higher than 58 °F Dry Bulb. This also assumes the room humidity equals the coil outlet humidity, whereas the room humidity is higher than coil outlet humidity. Both factor increase the calculated latent load relative to actual conditions and are therefore conservative.

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Effective Chilled Water Temperature = $52 + \frac{60970}{500*62.5} = 53.95$ °F

$T_{room} = \frac{133187*3.413 + 1317*95 + 25806*53.95}{(1317 + 25806)} = 72.70$ °F

Coil sensible load = $\frac{25806 * (72.70 - 53.95)}{12000} = 40.32$ tons

Total load = $40.32 + \frac{60970}{12000} = 45.40$ tons

Note: Calc. No. MC-5275 and Ref. 23 both indicate a very small available margin in the AHU, whereas this calculation shows a huge margin. It is obvious that vendor data from Ref. 32F was misinterpreted. The heat transfer data in Ref. 32F is for only one of the two coils installed in the air handling unit. This can be readily seen by calculating heat transfer from air flow.

Note: The actual ventilation flows to the 3 ESF pump rooms per Refs. 29 (Unit 1) and 45 (Unit 2) are very different from the design ventilation flows, and very different from each other. The result is a big difference in latent load between different rooms. It is noted that MC-5275 did not consider latent loads in the ESF pump rooms.

6. ESF SUMP VALVE ROOM, TRAIN C, FHB 009

The method is the same as the previous room, with the exception of the vent temperature. In the emergency mode, air is exhausted from this room via the exhaust ducting, but the supply fans are tripped & supply fan inlet dampers are closed. The only source of air is transfer air from the ESF pump room. Thus the inlet air temperature is the same as the ESF pump room temperature calculated in the previous example. The latent load is taken as 0 because moisture in the air has been removed in the ESF pump rooms.

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7. SPENT FUEL PIT PUMP ROOM, TRAIN C, FHB 107

Electrical load are from Ref. 10.

Piping heat load is from Ref. 18, converted to watts.

SFP Pump bhp = 172 (Ref. 41)

motor eff. = 0.930 (Ref. 41)

heat to room = 172 hp * 746 watts/hp * $\frac{(1-0.93)}{0.93}$ = 9658 watts

AHU fan bhp = 0.54 (Ref. 32I)

motor eff. = 0.689 (Ref. 3)

heat to room = 0.54 * 746 * 1 / .0689 = 585 watts

Note: This AHU is actually outside the room and the fan motor may be outside the conditioned space. This would decrease the heat load slightly, but the value used is conservative.

Total room load = 14909 watts

Room ventilation in emergency = 680 cfm (Ref. 29)

Room ventilation temperature is 95 °F.

Room ventilation constant = 1.09 * 680 = 741.2

Note: This is the only case, in either unit, in which the air handling unit air flow is significantly less than the design air flow.

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Chill water flow = 12 gpm (Ref. 32I)

Coil effectiveness = 0.7559 from Appendix B

coil constant = 1.09 * 680 * 0.7559 = 560

Latent load = $\frac{4840 * 680 * (139 - 66)}{7000}$ = 34320 b/h

Note: This is more than twice the actual latent load because it ignores the humidity removed by exhaust flow. Because the ventilation flow is the same as the AHU flow, the humidity removed by the exhaust will exceed the humidity removed by the AHU. This overestimates latent load on the coil and is conservative.

Effective chilled water temperature = 52 + $\frac{34320}{500*12}$ = 57.72 °F

$T_{room} = \frac{14909*3.413 + 741.2*95 + 560*57.72}{(741.2 + 560)}$ = 118.06 °F

Sensible load = $\frac{560 * (118.06 - 57.72)}{12000}$ = 2.82 tons

Total load = 2.82 + 34320/12000 = 5.68 tons

This is the only room in either unit where the steady state temperature approaches the design temperature.

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8. RADWASTE CONTROL ROOM, TRAIN C, MAB 217

Electrical loads are from Ref. 11.

Conduction load and piping load are from Ref. 15, converted to watts. However, the "use factor" for conduction is taken as zero because MAB HVAC is available.

Personnel load is from Ref. 15.

AHU bhp = 9.2 hp (Ref. 32.J)

motor eff. = 0.845 (Ref. 3)

heat to room = 9.2 * 746 * 1/0.845 = 8122 watts

Total room load = 24375 watts

room ventilation flow = 2000 cfm (Ref. 14)

room ventilation temperature = 65 °F (Ref. 22)

room ventilation constant = 1.09 * 2000 = 2180

AHU cfm = 16519 (Ref. 30)

Chill water flow to coil = 49 gpm (Ref. 32J)

coil effectiveness = 0.4739 per Appendix B

coil constant = 1.09 * 16519 * 0.4739 = 8533

latent load = 0

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$$T_{room} = \frac{24375*3.413 + 2180*65 + 8533*52}{(2180 + 8533)} = 62.40$$

$$\text{coil sensible \& total load} = \frac{8533 * (62.4 - 52)}{12000} = 7.40 \text{ tons}$$

This AHU is considerably oversized. Also, the room does not contain safety related equipment (Ref. 22). If chiller capacity is limited, this AHU could be electrically disabled to reduce both steady state and transient load.

9. OTHER ROOMS

The remaining rooms use the same methods and the same or comparable references. The only room with a different twist (for this scenario) is MAB 226. This room contains redundant B & C AHU's, which are both operable in the case being analyzed (loss of A train Safety Injection). This is the same down to calculation of room temperature. If both AHU's are operable per the particular scenario, the total room load is adjusted by adding the AHU fan load from train B and including the train B AHU constant in the room temperature equation.

$$T_{room} = \frac{(2064+216*1.0)*3.413 +239*52 +687.7*65 +239*52}{(239 + 686.7 + 239)} = 66.35 \text{ °F}$$

$$\text{coil sensible load} = \frac{239 * (66.35 - 52)}{12000} = 0.29 \text{ tons}$$

B. FAILURE OF ONE CHILLED WATER TRAIN

Failure of a chilled water pump or a single chiller leaves one EAB fan train and one Control Room fan train running without cooling chilled water flow until operator action is taken to restore the normal lineup. The control room fan adds heat to the operating coils. In the EAB, the air in the affected train will mostly be recirculated without cooling, because the air exchange between air distribution trains is limited. In this the heat absorbed by EAB room walls will limit the heatup until operator action is taken.

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The spreadsheet in Appendix G tabulates floor areas and volumes for each room in the EAB, and totals the areas served by each train. The length and width of each room is not tabulated. However, the minimum perimeter can be found assuming each room is square.

$$W = \text{width} = \text{length} = (\text{area})^{1/2}$$

Thus a low estimate of the surface area of each room (walls, ceiling and floor) is:

$$H = \text{height (ft.)} = \frac{\text{Vol.}}{\text{area}}$$

$$\text{Surface Area} = \text{area} * 2 + 4 * (\text{area})^{1/2} * H$$

The surface areas for rooms in the EAB served by each train header are totaled in Appendix G.

Assume walls are 1 ft. thick (most walls are at least 1 ft. thick).

From Ref. 31, the heat transfer properties of concrete are:

$k = 0.54$ btu/hr.ft.°F

$Cp = 0.20$ btu/lbm.°F

density = 144 lbm/ft^3

$a = 0.019$ ft^2/hr

Surface conductance for concrete ≈ 1.5 btu/ft^2.hr.°F

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1. Loss of "Train A" Chilled Water System

(See Appendix M4 for summary, File M4 for complete results)

This scenario results in all 3 trains of EAB & CR envelope fans running, but no cooling from the A train coils.

In the EAB, most of the heat load in A train areas will be absorbed in the walls. The small amount of air exchanged between fan trains limits the affects on the other EAB trains.

In the control room envelope, the extra fan train sharply increases the load on the two active coils.

Rooms that contain A & B or A & C air handling units see an increased load on the other coil.

This particular scenario has an interesting twist for the EAB HVAC system. The four trains of QDPS are located in rooms 015, 015B, 015C, & 015D. These rooms are on the 10 ft. elevation and receive air from only the "A" distribution header. If the "A" fan is running but "A" chilled water is not, these rooms will receive only warmer air.

The A train return header temperature = 63.71 °F in dual train normal operation (from a following section).

Case M4

header load	= 61.76 tons
return fan	= 15.60 tons
supply fan	= 36.40 tons
total	= 113.76 tons

delta temp. across fan train = 4.52 + 14.23 = 14.23 °F

The A train area has at least 125000 ft^2 of wall, floor, & ceiling at an initial temperature of 63.7 °F.

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Using charts from section 4.5 of Ref. 31:

$$\frac{k}{h*L} = \frac{0.54}{1.5*0.5} = 0.72$$

The ratio of surface temperature (wall temp.) to centerline temperature after 30 minutes (0.5 hrs) is taken from figure 4-9.

$$\frac{T(x,t) - T_\infty}{T(0,t) - T_\infty} = 0.56 \text{ for } x/L = 1.0 \text{ (x=0 is centerline)}$$

at t= 0.5 hrs. $at/L^2 = 0.19*0.5/(0.5)^2 = 0.038$

from fig. 4-8, temperature at centerline has not changed.

$T(0,t)-T_\infty = 63.7 - 77.9 = -14.2$ °F

$T(0.5,t)-T_\infty = 0.56*(-14.2) = -7.95$ °F

$T(0,t)-T_\infty = T_{cl} - T_{room}$

$T(0.5,t)-T_\infty = T_{wall} - T_{room}$

$$\frac{T_{wall} - T_{room}}{T_{cl} - T_{room}} = \frac{T_{room} - T_{wall}}{(T_{room} - T_{cl})} = 0.56$$

Conduction from room to wall = $h * A * (T_{room} - T_{wall})$

Ventilation load = $1.09 * cfm * (Tair.in - T_{room})$

For Quasi steady state

Qventilation + Qelectric = Qconduction

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$$Q\ conduction = \frac{1.09*38778*14.2}{12000} + 61.76 = 112\ tons$$

$Qconduction = h * A * (T_{room} - T_{wall}) = 112$ tons * 12000 b/hr.ton

$h * A = 1.5 * 125000$

$$(T_{room} - T_{wall}) = \frac{112*12000}{1.5*125000} = 7.15\ °F$$

$0.56*(T_{room} - T_{cl}) = 7.15$

$T_{cl} = 63.7$ °F

$T_{room} = 63.7 + 7.15/.56 = 76.5$ °F

Tair.in = 76.5 + 14.2 = 90.7 °F Use 91 °F for "trial coil outlet" in Appendix M4 spreadsheet.

Look at rooms 015, 015B, 015C, & 015D

Total elect. load = 16500 watts * 3.413 b/hr.watt = 56300 b/h

Total wall area (minimum) = 2191 +1060 +865 +1060 = 5176 ft^2

air flow = 3100 +1170 +1170 +1170 = 6610 cfm

delta temp. at design air flow = 56300/(1.09*6610) = 7.8 °F

Initial wall temperature ≈ 58 + 8 = 66 °F

Try room temperature = 88 °F

$T_{\infty} = 88$

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$T(0,t)-T_\infty = T_{cl} - T_{room} = 66 - 88 = -22$

$T(0.5,t)-T_\infty = T_{wall} - T_{room} = -22*0.56 = -12.3$

Wall temperature = 88-12.3 = 75.7

Qconduction = 1.5*5176*(88-75.7) = 95500 b/h

Qvent. = 1.09*6610*(92-88) = 28800 b/h

Qelect. = 56300 b/h

95500 > 28800 + 56300, therefore with air supply temperature of 92 °F, the room temperature would be below 88 °F.

Per discussion with Ron Falstreau (I&C Cognizate Engineer):

- QDPS is qualified up to 135 °F
- The temperature inside the cabinets runs 13 to 15 °F higher than room temperature.

The Limiting Condition for Operation (LCO) for the QPDS rooms requires an engineering evaluation within 24 hours if the temperature inside the QDPS cabinets exceeds 110 °F for more than 8 hours. In this case the temperature inside the cabinets could exceed 94 °F for a brief period (roughly 10 to 20 minutes) but the temperature would not exceed 110 °F and would not approach the maximum qualified temperature of 135 °F.

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2. Loss of B Train Chilled Water

The rooms served by the B train EAB ventilation header serve only B train equipment. Thus we are only concerned with the possible affect of higher B train temperatures on Trains A & C chiller loads.

Return header temperature = 59.38 °F initially (Section VII.E.1)

From M5 spreadsheet:

B header load	= 37.63 tons
return fan	= 16.93 tons
supply fan	= 36.47 tons
Total	= 91.03 tons

delta temp. across fan train =4.91 + 9.73 = 14.6 °F

Area > 32400 ft²

B return header flow = 30803 cfm

ventilation load = 1.09 * 30803 * 14.6 / 12000 = 40.85 tons

40.85 + 37.63 = 78.5 tons

$$T_{room}-T_{wall} = \frac{78.5 * 12000}{1.5 * 32400} = 19.4\ °F$$

$T_{room}-T_{wall} = 0.56*(T_{room}-T_{cl}) = 19.4$

$T_{room} = 19.4/0.56 + 59.4 = 94.0$

$T_{supply} = 94.0 + 14.6 = 108.6$

Use 110 °F as "trial coil outlet" temp. for B train EAB coil in case M5.

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3. Loss of C Train Chilled Water

C return header temperature = 65.86 °F

header load = 65.86 tons

delta temp. across fans = 4.57 + 9.67 = 14.24 °F

area > 74512 ft²

C header flow = 44301 cfm

$$\text{Ventilation load to area} = \frac{1.09*44301*14.2}{12000} = 57.1 \text{ tons}$$

ventilation load + elect. load = 57.1 + 65.9 = 123 tons

$$T_{room}-T_{wall} = \frac{123 * 12000}{1.5 * 74212} = 13.3 \text{ °F}$$

$$T_{room}-T_{wall} = 0.56*(T_{room}-T_{cl}) = 13.3$$

$$T_{room} = 13.3/0.56 + 68.6 = 92.4$$

$$T_{supply} = 92.4 + 13.3 = 106.6$$

Use 107 °F as "trial coil outlet" temp. for B train EAB coil in case M5.

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LOSS OF "TRAIN A" EAB FAN TRAIN (M7)

This senario results in increased heat loads in the EAB, relative to loss of Safety Injection, because Train A related heat loads are not lost. These heat loads are seen by the train B coil. The Train C coil is increased to a lesser degree by cross flow from the B return fan inlet.

The overall effect is reduced by reduction of the CR load (per train) because the total load is shared between 3 trains. Sharing of load between redundant room coolers also reduces the per train load.

This is the limiting steady state case for Unit 1 (M7)

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This case (and other EAB fan failure cases) results in higher EAB return fan inlet temperature than loss of a safety injection train, and is therefore is not the limiting case for EAB transient. The same is true of loss of chilled water.

Therefore, loss of Train A Safety Injection was selected for transient analysis.

"Loss of EAB Fan" refers to loss of the supply fan. The supply fan has more than twice the static pressure of the return fan. It appears that loss of a return fan would reduce the fan train flow somewhat but probably not cause loss of the train. Loss of a supply fan would

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probably leave the return fan unable to overcome the static pressure of the 2 operating fan trains.

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INITIAL TRANSIENT – EAB

The initial load on the EAB coil can be larger or smaller than the steady state load. The EAB structure is massive, with a huge potential for either absorbing heat to moderate temperature increases or releasing heat to moderate temperature decreases. The heat added or absorbed by the structure will act to maintain a constant return header temperature for the period of interest in this calculation (about 30 minutes following a LOCA). If the coil outlet temperature and humidity decrease significantly as the result of control actions from the SI signal and [illegible] return header temperature

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are lower than initial temperatures, then the initial load will be much higher than the steady state load. On the other hand, if coil outlet temperature and humidity do not change, and equilibrium return temperatures are higher than initial temperatures, then initial loads will be less than calculated steady state loads.

The spreadsheet model CHMODEL can be used to predict the initial load, assuming constant EAB return header air temperatures, by adjusting trial coil outlet temperatures to obtain the initial header return temperatures instead of matching the coil outlet temperatures.

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JUSTIFICATION FOR ASSUMPTION OF CONSTANT RETURN AIR TEMP. FOR TRANSIENT ANALYIS

The minimum surface area of the rooms in the Train A distribution system is $\approx 125{,}000\ ft^2$ (App. G)

Say $h = 1.5\ BTU/ft^2\,hr\,°F$

For a 1°F step decrease in air temp.

$$\dot{Q} = 1.5\ \frac{BTU}{hr\,°F\,ft^2} \times 125000\ ft^2 \times 1°F = 1.88 \times 10^5\ B/hr$$

$$= 15.6\ \text{tons}$$

If the flow is 35420 cfm (sum of design flows from calcs), this amount of heat transfer would heat the air by $\Delta T = \dfrac{1.88 \times 10^5\ B/hr}{1.09 \times 35420} = 4.9\ °F$

Of course a 1°F temp decrease can not cause a 5°F increase in outlet temperature. Actually the first 1/5 of the rise would raise the air temperature back to the original temperature & thermal equilibrium between the remaining walls & air.

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Very few walls in the EAB are less than 1 foot thick. Assume 0.5 feet of concrete surrounds each room in Train A.

$$Vol = 0.5 \times 125000 = 62500 \ ft^3$$

$$Mass = 144 \ \frac{lb}{ft^3} \times 62500 \ ft^3 = 9 \times 10^6 \ lbm$$

Total potential heat transfer for the 1°F step change

$$Q = 0.20 \ \frac{B}{lbm \ ^\circ F} \times 9 \times 10^6 \ lbm \times 1^\circ F = 1.8 \times 10^6 \ BTU's$$

The heat transfer necessary to raise 35420 cfm by 1°F is

$$1.09 \times 35420 \times 1 = 3.86 \times 10^4 \ B/hr$$

$$\frac{1.8 \times 10^6 \ BTU}{3.86 \times 10^4 \ BTU/HR} = \underline{46.6 \ hrs}$$

Thus it will take something like 2 days for the wall to approach thermal equilibrium with the step 1°F change in air inlet temperature.

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Initial EAB return header temperatures were calculated using the basis CHMODEL spreadsheet for 2 train operation, assuming the same normal lighting load, non-1E equipment, and non-1E cable load as used in the steady state load analysis. Inlet air temperature and humidity correspond to expected daily average conditions instead of design criteria maximums. The resulting chiller load is comparable to observed chiller loads. The return header temperatures for the 3 normal fan lineups are as follows:

FAN TRAINS OPERATING	RETURN HEADER TEMP. TRAIN A	TRAIN B	TRAIN C	
A & B	63.84	59.45	68.45	(App. Q1)
B & C	63.64	59.22	68.78	(App. Q2)
A & C	62.65	59.47	68.69	(App. Q3)

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The EAB & Control Room HVAC systems have been operated in the past with a single fan train and chiller train in operation. The return air temperatures should be higher in the single train mode of operation. CHMODEL was modified to predict EAB and control room temperatures during single train operation. The resulting return header temperatures are:

FAN TRAIN OPERATING	RETURN HEADER TEMP. TRAIN A	TRAIN B	TRAIN C
A	73.29	65.25	81.62
B	73.22	65.18	81.55
C	73.26	65.22	81.59

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There is no significant variation between different fan lineups, so the average temperature from the 3 cases will be used.

DUAL TRAIN

$RT_A = 63.71$

$RT_B = 59.38$

$RT_C = 68.64$

SINGLE TRAIN

$RT_A = 73.26$

$RT_B = 65.22$

$RT_C = 81.59$

Return air temperatures for the 9 steady state cases for Unit 2 are shown in the following table. While cases N1, N7, & N8 have about the same steady state load, case N1 will have a higher transient load (because of the lower return air temperature).

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EAB RETURN HEADER TEMPERATURES

CASE	TRN A	TRN B	TRN C
N1	$\boxed{64.94}$	74.67	80.46
N2	77.87	61.04	80.14
N3	78.39	74.99	69.85
N4	89.45	71.94	76.07
N5	76.15	88.99	77.49
N6	74.15	71.86	91.65
N7	78.39	75.62	80.81
N8	78.42	75.77	80.75
N9	78.42	75.71	80.59

CIRCLED TEMPS ARE LOWER THAN NORMAL TEMPERATURES IN SINGLE TRAIN OPERATION.

BOXED TEMPS ARE LOWER THAN NORMAL TEMPERATURE IN DUAL TRAIN OPERATION.

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Recalling that EAB transient load is maximumized when steady state post accident return temperatures are <u>less than</u> temperatures before the accident, it is apparent that case N1 (Loss of Train A safety Injection) is the worst case for EAB transient loading.

The return fan inlet temperature in case N1, modified to assume the 3 header return temperatures are constant, can be determined for single train and dual train operation prior to the accident, as follows:

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	A	B	C
Return Hdr flow, cfm	33576	23690	32421

Train A fans are not running, so return hdrs A & B are mixed at the inlet to B return fans will the resulting temperature.

$$T_B = \frac{33576 \times RT_A + 23690 \times RT_B}{(33576 + 23690)}$$

Only 44845 cfm of the mixed flow goes to the B return fan, with the balance (33576 + 23690 - 44845) = 12421 cfm going to C train

$$T_C = \frac{32421 \times RT_C + 12421 \times T_B}{(32421 + 12421)}$$

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The results for single train & dual train operation:

	Return Fan Inlet Temp. B	Return Fan Inlet Temp. C
Dual Train	61.92	66.78
Single Train	69.93	78.36
STEADY STATE FOR N1	68.97	77.27

Thus the EAB structure will act to decrease the initial EAB coil load, relative to the steady state, after an accident starting from normal dual train operation. The EAB structure will slightly increase the load on the train C coil and slightly decrease the load on the Train B

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coil, relative to steady state loads after an accident starting from single train operation.

The results are quantified using CHMODEL to match the return fan inlet temperatures to the calculated values, instead of match trial coil outlet temperature to actual coil outlet temperature. The transient loads are predicted for single train and dual train operation, assuming either 42, 48, & 52 °F chilled water supply temperaturres. The single train results are in Appendices S1, S2, & S3 while the dual train results are in Appendices T1, T2, & T3.

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INITIAL TRANSIENT — CONTROL ROOM

The control room envelope supply ducts contain electric reheaters of the following capacities:

43 KW
60 KW
5 KW

total 108 KW $\times \frac{3413 \text{ BTU/KW-HR}}{12000 \text{ BTU/TON-HR}}$ = 30.72 tons

These reheaters are de-energized by an SI signal. However, the heat from these coils will largely be replaced by heat transferred from the walls during the period of interest following a safety injection signal.

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The control room could credibly be operating with a room temperature (and return air temp) of 78°F. (Per Tech Spec)

There is sufficient building structure to hold the return air approximately constant at 78°F during the initial transient.

Thus, even with no change in cont. room coil water flow rate or air outlet conditions, the control room transient will place a larger initial load than the steady state load.

The transient load can be obtained from the spreadsheet by using a trial control room temperature of 78°F without iterating to match the calculated result.

assuming no changes in chill water flow, and dual train initial conditions, the transient can also be approximated by assuming all control room reheaters are fully energized before and after the accident. The heaters are de-energized by the SI signal, but the transient input from walls, etc will approximately replace the heat input.

The steady state Temperature run from dual train operation indicates even with all reheaters energized the control room Temperature is less than 76°F. However, the control room envelope transient analysis in Appendices S1, S2, S3, T1, T2, & T3 assume control room Temp = 78°F, which is conservative.

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Initial transient - other rooms with AHU's

Few of the individual room coolers served by the essential chilled water system are actually needed in normal operation. In most of these rooms the expected post accident temperature is considerably less than the design temperature. If the room is at design temperature when the AHU is started, the rate of heat removal is considerably higher than the steady state loads included in CHMODEL.

The heat transfer from an AHU

$$\dot{Q}\,(BTU/HR) = \underbrace{C_{min}\,\mathcal{E}}_{\text{constant } C_1 \text{ or } C_2}\,(T_{room} - T_{CH\,WTR})$$

$$\text{Load, tons} = \frac{\dot{Q}}{12000} = \frac{C_1\,(T_{room} - T_{CH\,WTR})}{12000}$$

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APPENDICES K1 & L show the maximum transient room AHU loads for Unit 1 & Unit 2. AHU constants are taken from appendix M1 & N1. The total of all coolers in each train is calculated assuming all rooms start at design temperature, for chill water temperatures of 42, 48, & 52 °F.

The 4th run in Appendix L assumes restrictions on initial temperature in selected rooms, with 52 °F chill water. The restrictions assumed are shown on the next page.

The total load was slightly greater for Unit 1, so Unit 1 was selected for additional case studies. Appendix K2 gives initial transient loads with Unit 1 AHU constants and with selected rooms at reduced temperature. Appendix K3 gives initial transient loads for Unit 1 AHU's with selected rooms at reduced temperature and reduced AHU constants for the ESF Pump room AHU's. The revised constants are from Appendix W.

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Assume the following:

Radwaste Control Room - AHU's locked out

Electrical Penetration Rooms	≤ 75 °F	
ESF Pump Rooms	≤ 75 °F	
CCW/Chiller Rooms	≤ 104 °F	
Rad Monitoring Room	≤ 104	* design temp
RWST Room	≤ 104	* design temp
Other Rooms	≤ 104 °F	* design temp

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INITIAL TRANSIENT FROM SINGLE TRAIN
(C TRAIN)

	42°F *	48°F *	52°F *	← chilled water supply header temp.
EAB	202.7	178.2	161.5	
CR	69.5	60.1	53.8	
OTHER (App. K1)	241.3	217.5	201.7	
TOTAL	514	456	417	TONS

* all rooms initially at design temperatures

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TRANSIENT FROM SINGLE TRAIN
WITH REDUCED ROOM TEMPERATURES

	42°F	48°F	52°F	
EAB	202.7	178.2	161.5	
CR	69.5	60.1	53.8	
App. KZ	135.4	115.9	102.9	
	408	354	318	tons

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INITIAL TRANSIENT FROM DUAL TRAIN
(C TRAIN)

	42°F (1)	48°F (1)	52°F (1)
EAB	162.4	137.9	121.2
CR	69.5	60.1	53.8
KI	241.3	217.5	201.7
TOTAL, TONS	473	416	377

(1) All rooms initially at design temperature.

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TRANSIENT, FROM DUAL TRAIN, RESTRICTED ROOM TEMPS.[2]

	42	48	52	← CHILL WTR TO COILS
EAB	162.4	137.9	121.2	
CR	69.5	60.1	53.8	
K2	135.4	115.9	102.9	
TOTAL	367	314	278	tons

(2) Radwaste control room AHU locked out plus reduced room temp in Electrical Penetration Rooms, ESF pump rooms, & CCW/chiller rooms.

TRANSIENT, FROM DUAL TRAIN, RESTRICTED ROOM TEMP. & RESTRICTED FLOW TO ESF ROOM COOLERS[3]

EAB	162.4	137.9	121.2	
CR	69.5	60.1	53.8	
K3	121.5	105.5	93.2	
TOTAL	353	304	268	tons

(3) Restrictions in Note (2) plus ESF pump room cooler chill water flow adjusted to 40 ± 3 gpm.

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The transient load is comfortably less than 300 tons only if single train operation is prohibited (or restricted) and certain room temperatures are restricted below design values. "Normal" room temperature were estimated in appendices Q1, Q2, Q3, R1, R2, & R3 for dual train & single train EAB/control room lineups. The biggest problem with the restricted room temperature appears to be the ESF Pump room for Train A. This room has no air supply, exhaust, or transfer, in the normal mode. If Train A chill water is operating the AHU can be operated to maintain the room at a very cool temperature. If not, it's not clear how the room temperature is maintained.

With current operating procedures & design, the chilled water temperature would rise during the initial transient until the transient load equaled the combined chiller capacity of 450 tons. This occurs at about 48°F chill water. With an eye toward cold weather restrictions, the cold weather operation will need to consider chiller outlet temperatures up to 48°F regardless of the actual controller setpoint.

One 300 ton chiller can handle the steady state heat load in any train, with chilled water supply temperatures @ 52°F. With administrative limits on room temperatures in the ESF Pump Rooms, CCW/Chiller rooms, and Electrical Penetration rooms; the peak transient

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load is less than 300 tons.

There are minor differences in chilled water flow rates shown in different documents.

	150 ton	300 ton
York data	360	637
Train A flow diagram	303	607
Train B " "	293	587
Train C " "	318	636

For York data

$$\Delta T_{300} = \frac{300 \text{ tons} \times 12000 \text{ B/Hr-ton}}{500 \times GPM_{300}} = 11.30$$

$$\Delta T_{TOTAL} = \frac{300 \text{ tons} \times 12000}{500 \times (GPM_{300} + GPM_{150})} = 7.22$$

Say 300 ton outlet = 48°F

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York Data

Return Wtr Temp = $48 + \Delta T_{300}$ = 59.30 °F

Supply header temp = 59.30 − 7.22 = 52.08 °F

Using Flow diagram data

	TRAIN A	TRAIN B	TRAIN C
ΔT_{300}	11.86	12.27	11.32
Ret Wtr T.	59.86	60.27	59.32
ΔT_{total}	7.91	8.18	7.55
Supply Hdr T	51.95	52.08	51.78

If the 300 ton chiller is operating alone with a setpoint of 48 °F, the supply header temperature will be very close to 52 °F.

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From Appendix K, the peak transient load varys with chilled water temp.

@ 50° 110.41

@ 52° 103.95

6.46 tons

or 3.23 tons per degree for Room AHU's

EAB & Control Room loads from

	48°	52 °F
EAB	137.9	121.2
CR	60.1	53.8
	198.0	175

ΔLOAD = 23 tons

or 5.75 tons per degree for CR & EAB

COMBINED TOTAL = 8.98 tons/degree say 9.0

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Applying $-2°F$ to the $48°F$ setpoint

& 279 ton transient load

$279 + 2 \times 9 = 297$ tons ≤ 300 tons

The actual capacity of the 300 ton chiller would be greater than 300 tons during the time frame of interest because the ECP outlet will not have seen the increase from the accident.

With $+2°F$ tolerance, chiller load would be less, but EAB coil outlet temperatures would increase.

Chiller load $= 279 - 18 = 261$ tons

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As before, using Train C flow diagram values.

Tout of 300 ton = 50 °F

$$\Delta T_{300} @ 261 \text{ tons} = \frac{261 \times 12000}{500 \times 636} = 9.85$$

Return Wtr Temp. = 50 + 9.85 = 59.85

$$\Delta T_{total} = \frac{261 \times 12000}{500 \times (636 + 318)} = 6.57$$

Supply hdr T. = 59.85 − 6.57 = 53.28

The +2°F tolerance only causes a 1.3 °F increase in supply hdr T. The 9 tons per degree is actually only header T., so this will be iterated

279 − 9 × 1.3 = 267.3

$$\Delta T_{300} = \frac{267.3 \times 12000}{500 \times 636} = 10.09$$

Return Wtr T. = 60.09

ΔT_{TOTAL} = 6.72

Supply hdr T. = 53.37 °F Close enough

say 1.4°F increase

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The limiting EAB transient is analyzed for 52°F in appendix T3.

The "computed EAB coil outlet air temperature" reflects the expected coil perfomance. For train C the coil outlet temp. = 56.69°F The affect of a 2°F increase in chilled water out of the 300 ton chiller, operating alone, gives a 1.4 increase in supply header temperature, which would raise the EAB coil outlet by 1.4°F.

Coil outlet = 56.7 + 1.4 = 58.1 °F

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The following rooms cooled by the EAB HVAC system have TECH SPEC temperature limits:

Switchgear Rooms ≤ 85 °F
(010, 212, & 318)

QDPS Rooms ≤ 94 °F (Measured inside QDPS cabinets)
(15, 15B, 15C, & 15D)

Per Ref. 12

	010	212	318	015	015B	015C	015D
DESIGN AIR FLOW	17800	18800	19230	3100	1170	1170	1170
" SUPPLY AIR TEMP	55	55	55	55	55	55	55
CALC LEAVING AIR "	79.8	80.3	80.1	67.7	72.3	68.8	75.1
CALC HEAT LOAD	4.8×10^5	5.1×10^5	5.2×10^5	4.3×10^4	2.2×10^4	1.7×10^4	2.5×10^4
MARGIN TO TECH SPEC	5.2°	4.7°	4.9°	26.3° −15 = 11.3	21.7° −15 = 6.7	25.2° −15 = 10.2	18.9° −15 = 3.9
Limiting Case		59.7°					

Considering the arbitrary margins added to heat loads in Ref 12 the real margins are considerably greater.

Per EIC responsible engineer Ron Falstream, the QDPS internal cabinet temperatures run 13 to 15 °F above room temperature

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58.1 < 59.7°F so the limiting transient will not risk exceeding the Tech Spec in the CAB switchgear rooms. Considering how much extraneous margin is included in the Reference 12 heat loads and considering the large affect the room walls would have on moderating temperature increases, the actual margin is much larger than the above figures indicate.

F.
COIL HEAT TRANSFER MODEL, Description & data sources
(appendix B & C, D, & E)

Flow, SCFM — Actual flow from air balance is generally used. EAB & Control room flows are approximate.

Test conditions are generally close to standard conditions so cfm's are used for scfm.

Total load, Sensible load, Dry Bulb in, Wet bulb in, Dry Bulb out -- These are not used in calculating thermal effectiveness (E). These values may be design values or some other case of interest.

Water Inlet & outlet temperatures -- These values are used for the water side film coefficient. This has a relatively small effect on E and is not changed for each case.

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FACE AREA in ft^2 = area of one coil

No. Coils = No. of coils per AHU

No. rows = No. of tube rows in a coil

Tubes/row = No of tubes in each row

Tube OD = Usually this is the nominal tube dia. before expansion. Not used

Water velocity - vel in tube per data sheet

Fins/inch from data sheet

Length = length of each tube, inches

Tube thickness = nominal tube thickness before expansion, from data sheet

Tube & fin material from data sheet

F.F. = Fouling Factor, from data sheet

P.O. # & dwg reference show the data sheet log #.

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Items from DWG. REF. to AIR_IN are explained in App. A.

Air_In & Air_Out are calculated from the sensible load, # of coils per AHU, and air flow. This is for info only

TIN. EFF. is effective chill water temp calculated from latent load & gpm. This is used in calculating Air_IN & AIR-OUT and is for info only.

THEFF is the thermal effectiveness used in other spreadsheets for the applicable unit.

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Ø	[illegible] 10-13-93	Cad 10/8/93

APPENDIX D

This is the same spreadsheet as B & C, except all AHU data is per the design.

AIR_IN & AIR_OUT can be compared to DB IN & DB OUT for an indication of how closely the heat transfer model reflects the design performance data.

also EFFSPEC is thermal effectiveness calculated from design performance data.

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Appendix E Description & data source

This appendix uses AFF catalog data to find the air side heat transfer coefficient as a function of flow, for 2, 4, 6, & 8 rows.

$h \propto G^n$ Ref 19

$G = \dot{m}/A_{min}$

n = constant for the configuration, usually between .55 & 0.8

Preparer VH 10-5-93 Rev Ø
Reviewer CAD 10/5/93

		1
FACE VEL	600	2
CAT B/H/SQFT	10300	3
DELTAT	10	4
		5
TAIR_IN	88	6
TAIR_OUT	72.3	7
VTUBE	2	8
WTR OUT	55	9
TW_IN	45	10
TFACTOR	45.03126	11
TWTR_EFF	45.03126	12
CR	0.634951	13
EFFECTIVENESS	0.365382	14
XFACTOR	1.10008	15
EFFPASS	0.215167	16
		17
LMTD	30.04331	18
SENSIBLE Q	68162.31	19
U (uncorrected)	7.853983	20
CORR. FACTOR	0.993179	21
GAMMA	0.153839	22
NTU/PASS	0.263084	23
U	7.907921	24
ho*Efin	11.69355	25
ho	12.51192	26
GPM	13.67521	27
TUBE OD	0.645	28
TOTAL,B/H	68376.07	29
SENSI,B/H	68162.31	30
LENGTH,IN	79.66144	31
OD	0.645	32
ID	0.591	33
NA	8	34
NR	2	35
NF	8	36
FT	0.01	37
TL	79.66144	38
SCFM	3983.072	39
HREF	12.6527	40
GREF	5148.741	41
N	0.576505	42
KTUBE	228	43

 Preparer: VJL 2-5-93 Rev Ø

Reviewer: CAW 10/8/93

KFIN	228
NW	8
TFILM	50
Ri	0.0003
HH	12
TT	0.027
WW	79.66144
DD	3
AFIN	272.372
AOTUBE	16.50073
ATOTAL	288.8727
AITUBE	16.43399
ARATIO	17.57776
MINFA	3.481205
FACEAREA	6.638453
MASSAIR	17923.82
FACEVEL	600
G	5148.741
HAIR	12.6527
M	11.54064
FINL	0.041283
ML	0.476426
FINEFF	0.930631
ETAo	0.934593
AWALL	17.18478
Rw	0.000166
Ri	0.0003
HWTR	495.161
UCOIL	7.967876
Cair	4341.548
Cwtr	6837.607
Cmin	4341.548
Cmax	6837.607
CRATIO	0.634951
NTU	0.530157
NTU/NR	0.265078
GAMMA	0.15491
EPASS	0.21649
	1.211905
EFFECTIVENESS	0.365382
THEFF	0.367283

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Rows 2 ->10 are data from AAF catalog data

$$TFACTOR = TW\_IN + \frac{(TOTAL, B/A - SENSI, B/A)}{(500 \times GPM)}$$

TWTR_EFF = Max of TW_IN & TFACTOR

(Before all bugs were worked out TFACTOR was sometimes less than TW_IN. This step is left over from debugging)

$$CR = \frac{C_{min}}{C_{max}} = \frac{(WTR\ OUT - TWTR\_EFF)}{(TAIR.IN - TAIR\ OUT)}$$

$$EFFECTIVENESS = \frac{TAIR.IN - TAIR.OUT}{TAIR.IN - TWTR\_EFF}$$

Note: The air side is C_{min} in all cases of this appendix.

$$XFACTOR = \left[\frac{1 - EFFECTIVENESS \times CR}{1 - EFFECTIVENESS}\right]^{1/NR}$$

Rows

Note: This is from $\varepsilon = \dfrac{\left(\dfrac{1 - \varepsilon_p C_{min}/C_{max}}{1 - \varepsilon_p}\right)^n - 1}{\left(\dfrac{1 - \varepsilon_p C_{min}/C_{max}}{1 - \varepsilon_p}\right)^n - \dfrac{C_{min}}{C_{max}}}$ (Ref 21)

$$XFACTOR = \frac{1 - \varepsilon_p C_{min}/C_{max}}{1 - \varepsilon_p} \qquad CR = \frac{C_{min}}{C_{max}}$$

solved for XFACTOR

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$$EFFPASS = \varepsilon_p = \frac{(1 - XFACTOR)}{(CR - XFACTOR)}$$

from definition of XFACTOR solved for ε_p

$$LMTD = \text{Log Mean Temp Diff.} = \frac{(TAIR\_IN - WTR\,OUT) - (TAIR\_OUT - TWTR\,EFF)}{LN \dfrac{(TAIR\_IN - WTR\,OUT)}{(TAIR\_OUT - TWTREFF)}}$$

$$SENSIBLE\ Q = 500 \times GPM \times (WTR\,OUT - TWTR\_EFF)$$

$$U(\text{uncorrected}) = \frac{Q}{LMTD \times AREA}$$

$$CORR.\ FACTOR\ (\text{for LMTD}) = \frac{U(\text{uncorrected})}{U\ \text{actual from later calc.}}$$

This is a sanity check. Results should be very close to, but not exceed or equal 1.0.

$$GAMMA = -CR \times LN\,(1 - EFFPASS) \qquad \text{(Ref 21)}$$

This is from $\varepsilon = 1 - e^{-\Gamma \frac{C_{max}}{C_{min}}}$ solved for Γ

ε is ε_p for one cross flow pass.

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D	JJf 10-5-93	CAD 10/8/93

$$NTU/PASS = -(1/CR)\,LN(1-GAMMA)$$

from $\Gamma = 1 - e^{-NTU(C_{min}/C_{max})}$ (Ref 21)

solved for NTU, which is NTU for one pass.

$$U = \frac{(NTU/PASS * C_{air})}{(ATOTAL/NR)}$$

from definition of $NTU = \frac{UA}{C_{min}}$ except NTU is per pass & area is per pass

Thus U can be found from the performance data without knowing the LMTD correction factor.

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$$h_o \times E_{fin} = \left[\frac{1}{U} - R_W - \frac{ARATIO}{HWTR} - ARATIO \times R_i\right]^{-1}$$

$$h_o = \frac{h_o \times E_{fin}}{ETA_o} \qquad E_{fin} \equiv ETA_o$$

A coil must be constructed to fit the data

$$GPM = \frac{VTUBE}{1.17} \times NW$$

from Velocity = 1.17 × GPM per tube for AFF standard coil construction

NW = No. of water circuits = 8 in all cases

The imaginary coil is 12" high ⇒ 8 tubes

TUBE OD = 0.645" (after hyd. expansion)

TOTAL LOAD, BTU/HR = 500 × GPM × (WTROUT − TW.IN)

SENSIBLE LOAD, BTU/HR = Cair (TAIR.IN − TAIR.OUT)

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$$\text{LENGTH, IN (of tubes)} = \underbrace{\left(\frac{\text{TOTAL LOAD}}{\text{CATELOG BTU/H/FT}^2}\right)}_{\text{req'd face area, ft}^2} \times 12\ \frac{\text{in}}{\text{ft}}$$

$$\text{Length, ft} \times \underset{\text{L, ft}}{\text{Ht}} = \text{Face Area}$$

The remaining steps are the same heat transfer model used in B, C, & D, with standard coil construction (tube thickness = 0.028")

For this example, @ 600 scfm/ft², the air side heat transfer coeff. is 12.51192.

The average of 4 different data sets with 2 rows @ 600 scfm/ft² is 12.64536

Simarly, the average of 4 corresponding data sets for 2 rows @ 400 scfm/ft² is 10.01746

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$$G = \frac{\dot{m}}{A} \propto \frac{SCFM}{FACE\ AREA}$$

$$h \propto G^n$$

$\propto SCFM/ft^2$

$$\frac{h_1}{h_2} = \left(\frac{G'_1}{G'_2}\right)^n$$

use $G' = SCFM/ft^2$

$$\ln\left(\frac{h_1}{h_2}\right) = n \ln\left(\frac{G'_1}{G'_2}\right)$$

$$\ln h_1 - \ln h_2 = n\left[\ln G'_1 - \ln G'_2\right]$$

$$n = \frac{\ln h_1 - \ln h_2}{\ln G'_1 - \ln G'_2}$$

$\ln h_1 = \ln(12.64536) = 2.53729$

$\ln h_2 = \ln(10.01746) = 2.304329$

$\ln G_1 = \ln(600) = 6.39693$

$\ln G_2 = \ln(400) = 5.991465$

$$n = \frac{2.53729 - 2.304329}{6.39693 - 5.991465} = 0.574553$$

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G_{REF} is arbitrary set at 5148.741 (600 scfm/ft^2)

Thus h_{ref} = 12.64536

Because h_o is used to calculate fin efficiency the result depends to a small degree on the HREF inserted in row 40. Thus HREF used in the spreadsheet may differ in the 4th digit from the calculated value. This is a trivial difference.

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6. Sensitivity Analysis of Chilled Water Flow to Cooling Coils

The chilled water system has flow elements only on the flow through the chiller. The original flow balance, performed during preoperational testing, adjusted flow based on ΔP across the coils. This is satisfactory for most of the coils because water flow has little affect on performance. However, chilled water flow has a significant effect on the ESF PUMP ROOM COOLERS. These coolers in turn have a controlling effect on the transient load. It is important that chilled water flow to the ESF PUMP ROOM COOLERS NOT EXCEED THE 62.5 gpm DESIGN FLOW.

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Appendix U contains the results of the sensitivity analysis of coil performance vs. chill water flow. The entire spreadsheet is in file A:/U

Appendix C (unit 2 coils) was duplicated for the starting point. The spreadsheet was duplicated twice. In one case the GPM was changed to 0.75 × Design Flow and in the 2nd case GPM is 1.25 × Design Flow

C @ 100% = Cmin × E at design flow

C @ 75% = Cmin × E at 75% of design flow

C @ 125% = Cmin × E at 125% of design flow

$$\text{Delta \%} = \frac{(C@125\% - C@75\%)}{C@100\%} \times 100$$

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The C values for the EAB coolers and ESF Pump Room coolers are for ONE of two installed coils per AHU.

The limiting case for minimum chilled water flow to the ESF Pump Room Coolers is Unit 1 Train B. This is because this room has a much higher flow and therefore much higher latent load than any other ESF pump room in either unit.

Appendix V contains the ESF Pump Room model for Unit 1. A flow of 35 gpm to the B Train AHU, with the corresponding thermal effectiveness from Appendix W, gives a room temperature of 90 °F. This is significantly

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below the 104°F design temperature. This margin is appropriate because of the large latent load and the non-rigid method used in this calculation for latent loads.

The initial transient load on the ESF Pump Room AHU's, which makes up a large fraction of the total transient load, can be reduced by reducing the maximum chilled water flow. By limiting the flow to no more than 45 gpm (40 gpm ± 5 gpm), the transient load is reduced by about 10 tons.

Appendix W gives the thermal effectiveness for a min flow of 35 gpm in the limiting case for room temperature (Unit 1, Train ?)

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and the coil constant (÷2 for AHU constant) for all 3 rooms in both Units.

The flow to the ESF Pump Room AHU's should be administratively controlled between 37 & 43 gpm (leaving ± 2 gpm or 5% for instrument error) using a portable ultrasonic flow meter, if the chilled water train is operable with only a 300 ton chiller operable.

FIGURE 1

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