MSRE Cooling Tower

Value bought (8450)

TCU 858-1

Thermal Ball

DAA B 40509 Rev C

Control Reg's - Gus Smith: Temp + Charcoal Bed

Natural Draft Tower = 2 Fans

Cooling Tower Supply - P. Harley R. Guyman

Rating 550 gpm 95°F → 85°F @ Tw = 79°F

Marley Double Flow Acqua Tower No. 8320

Charcoal Bed temp reg's - Gus Smith

Want steady 85°F (controlled by steam heat)

Space Cooler Reg's:

Cooling tower water → 85°F →

Space cooler demineralized water → 95°F

Per Harley & Guyman want pretty good control on Cooling Water temp outlet so that I cell air will stay fairly constant. (They assume heat load will be constant)

70 HP

70 HP × 1.2

Space Cooler 225 kW
Treated-Water Heat Exchanger

Characteristics - per Tom Pickel

Ex. Area = 1885 ft²
4 tube pass, 1 shell pass
360 tubes @ 20" O.D. x 18 ga.
Shell = 32" O.D.

Design

- Tube Vel. = 3.1 ft/sec
- Shell Vel. = 2.2 ft/sec (1200 gpm)
- T = 54.9 °F

Tube 900 gpm for 3.1 ft/sec:

- Tube ID. = 1.0" - 2 (0.049") = .902"
  & 18 ga. thickness

- Flow area = \( \frac{\pi (0.902)^2}{4} \times \frac{360 \text{ tubes}}{144 \text{ in}^2/\text{ft}^2} = 1.6 \text{ ft}^2 \)

- 3.1 ft/sec \( \rightarrow \) 3.1 \( \times \) 1.6 = 4.96 ft³/sec

- 4.96 ft³/sec = 4.96 \( \frac{ft^3}{sec} \times \frac{7.481 gal}{ft^3} \times \frac{60 \text{ min}}{min} = 2,230 \text{ gpm} \)

- Per Design Data CF 62-9-55, p. 28

Cooling tower water flow rate = 540 gpm total; 260 gpm to treated water

Treated water flow rate = 310 gpm (100 gpm to thermal shield)

Per Pickel, treated water in tubes.
MSRE Cooling Tower
Treated Water He

Assume treated water in tubes

Tube Velocity = \frac{310 \text{ gpm}}{2230 \text{ gpm}} \times 3.1 \text{ ft/sec} = 0.43 \text{ ft/sec}

Velocity in shell = \frac{260 \text{ gpm}}{1200 \text{ gpm}} \times 2.2 \text{ ft/sec} = 0.476 \text{ ft/sec}

Reynolds no. - tubes

\text{Re} = \frac{50.6 \cdot \frac{Q}{\mu}}{d \cdot \nu} \\
Q = \text{gpm} \\
\nu = \frac{1}{4} \cdot 62.4 \\
d = \text{pipe ID (in)} = 0.902

\mu = \text{viscosity, centipoise} = 0.75 @ 70 \degree F

\text{Tube Re} = \frac{(50.6)(310 \text{ gpm total})(62.4)}{(360 \text{ tubes})(0.902)(0.75)} = 4020

\approx \text{in transition zone; probably turbulent}

Since shell velocity > tube velocity, assume turbulent also

Find \( h \) tubes for design case (2230 gpm)

\text{Empirical Phenom p 399} \rightarrow \frac{hD}{K_b} = 0.026(\text{Re})^{0.8} (Pr)^{0.53}

Pr = 4.0

\text{Re} = 4020 \times \frac{2230}{310} = 23,900

D = 0.902'' = 0.0752 \text{ ft}

K = 1.38 \text{ BTU/HR-F}
tube side: \[ h_t = \frac{(38)(0.026)(28,905)^{0.33}}{(0.0752)} = 770 \text{ BTU/hr ft}^2 \text{°F} \]

Assume tube side resistance zero, then shell side \( h_s \) is

\[ \frac{1}{U} = \frac{1}{h_s} + \frac{1}{h_t}; \frac{1}{54.9} = \frac{1}{h_s} + \frac{1}{770}; h_s = \frac{59.1}{\text{BTU/hr ft}^2 \text{°F}} \]

Assume \( h_t \) varies as \( W^{0.8} \)

For MSRE conditions, \( h_t = \left(\frac{312}{2,000}\right)^{0.8}(770) = 158.5 \text{ BTU/hr ft}^2 \text{°F} \)

Assume \( h_s \) varies as \( W^{0.6} \)

For MSRE conditions, \( h_s = \left(\frac{260}{12,000}\right)^{0.6}(59.1) = 23.6 \text{ BTU/hr ft}^2 \text{°F} \)

So overall \( U \):

\[ \frac{1}{U} = \frac{1}{158.5} + \frac{1}{23.6}; U = 20.5 \text{ BTU/hr ft}^2 \text{°F} \]

So \( UA = (20.5)(1885) = 38,700 \text{ BTU/hr °F} \)

\[ \text{Heat Exchanger Efficiency } \varepsilon = \frac{\Delta T_{\text{net}}}{T_{\text{H in}} - T_{\text{cold in}}} \]

Assume counterflow:

\[ N_1 = \frac{\Delta T_{\text{e}}}{\Delta T_{\text{h}}} = \frac{\dot{m}_h c_p}{\dot{m}_c c_p} = \frac{310}{260} = 1.19 \]

\[ N_2 = NTU = \frac{UA}{\dot{m}_h c_p} = \frac{38,700 \text{ BTU/hr °F}}{(310 \text{ lb/min}) (800 \text{ Btu/lb °F})(1 \text{ BTU/hr °F})(60 \text{ min/hr})} = 0.26 \]

\[ N_2 = 0.26 \]
Cooling efficiency \( \varepsilon = \frac{1 - \exp \left[ \frac{(1 - |N_1|)}{N_2} \right]}{1 - |N_1| \exp \left[ \frac{1 - |N_1|}{N_2} \right]} \)

\[ \varepsilon = \frac{1 - \varepsilon_{.0995}}{1 - 1.19 \varepsilon_{.559}} = \frac{1 - 1.0508}{1 - 1.251} = \frac{-0.502}{-1.251} = 0.4 \]

(For low \( \varepsilon \) and geometric assumption ok)

So a change in the hot side \( DT \) of 12°F

(for the case of no full load on Thermal Shield), if a fixed cold stream inlet temp., the hot inlet temp would go up \( \frac{12°F}{0.4} = 30°F \).

(Eqado)

Recalculate Cooling efficiency:

\( \varepsilon = \frac{\Delta T_{\text{Hot}}}{T_H(\text{in}) - T_C(\text{in})} \)

New Thermal Shield heat load = 100 kW (HC Carbone)

\( N_1 \) = same as before = 1.19

\[ N_2 = \frac{N_{tu}}{N_H C_H} = \frac{(232)(1885)}{(310 \times \text{min})(8.44 \times \text{gal})} \times \text{BTU/hr} \times 60 \text{ min/hr} \]

\( N_2 = 2.94 \)

\[ \varepsilon = \frac{1 - \exp \left[ \frac{(1 - 1.19)(2.94)}{2.94} \right]}{1 - 1.19 \varepsilon_{.559}} = \frac{1 - 1.75}{1 - 2.082} = \frac{0.75}{1.082} = 0.7 \]

For \( \Delta T_{\text{Hot}} \) of 2°F (0-100% Shield load), Hot inlet = \( \frac{2°F}{0.7} = 2.86°F \)

So \( \Delta \text{Hot out} = 1°F \)
Design Conditions

Tower Basin Water = 85°F, 550 gpm (260 gpm to Treated Water H.E.)

Design Tower inlet Temp. = 95°F (at Ttwxball = 79°F)

MARLEY DOUBLE FLOW AQUA TOWER No. 8320.

Estimated treated water design load = (100 kW + 225 kW + 70 HP) x 1.2
(by R. Gunson & P. Harley) = 450 kW (max.)

For 310 gpm treated water flow, \( \Delta T = \frac{Q}{\dot{W} C_p} = \)

\[
\frac{450 \text{ kW} \times 0.947 \text{ BTU/SEC} \times 60 \text{ SEC/MIN}}{310 \text{ GPM} \times 8.3 \text{ LB/GAL} \times 1.0 \text{ BTU/LB/OF}} = 10^\circ \text{F}
\]

For treated water cooling efficiency \( \varepsilon = 0.693 \):

\[
0.693 = \frac{10^\circ \text{F}}{T_{\text{hot(out)}} - 85^\circ \text{F}} \Rightarrow T_{\text{hot(out)}} = 90^\circ \text{F}
\]

\( \Delta T_{\text{cold}} = 10^\circ \text{F} \times 310 \text{ gpm} = 12^\circ \text{F} \); \( T_{\text{cold(out)}} = 97^\circ \text{F} \)

Temperature of Tower return water NOT going to H.E.: 97°C x 260 gpm + T x 290 gpm = 95°F x 550 gpm

\( T = 91^\circ \text{F} \)

Total Tower Load (Rated) = \( \frac{AT}{(550 \text{ gpm} \times 8.3 \text{ LB/GAL} \times 60 \text{ SEC/MIN}) \times \frac{1}{947.519 \text{ SEC}}} \)

= 800 kW
ΔP across Tower by-pass valve when full open:

TCV 858-1: (MSRE-48) \( P_{\text{upstream}} = 5 \text{ psi} \)

Max flow = 565 gpm
6" valve, equal % trim
CV = 360
Fail open (v.s. fail closed on flow out)

\[
\Delta P = \frac{P}{62.4} \left( \frac{Q}{CV} \right)^2 = \frac{62.1}{62.4} \left( \frac{550}{360} \right)^2 = 2.34 \text{ psi}
\]

Head of water = \( \frac{2.34 \text{ lb/ft}^2 \times 144 \text{ in}^2}{62.1 \text{ lb/ft}^3} = 5.4 \text{ ft} \)

So, with valve wide open, no flow will go thru tower. (6% of 565 gpm)

To get flow distribution vs. valve CV need \( \frac{Q}{\Delta P} \) tower.
This is probably just equal to head needed to get to top (6%)

Re: Tower Characteristics - to get description of control system behavior, we'd need to know cooling tower characteristics as functions of \( T_{WB}, \) Flow, # Fans on, & Load.

However, if it can be shown that by-pass temp. control is good for various changes in tower capacity (eg. pump flow, \( \Delta T_{WB}, \) AF flow, etc.) then maybe the only problem would be to find stability for tower \( \frac{Q}{\Delta P} \) characteristics.

Possibility of using cascade system with treated water source temp. controlled at say 90° should be investigated.

Call Marley: Find Capacity vs \( T_{WB}, \) Fans, & Flow
Pipe from Cooling Tower water pump to Jet 6" pipe

To Jet = 10'

From Jet to Diesel House Wall = 136'

From Diesel House Wall to H. Ex. Conn = 52'

Total length @ 540 gpm = 10'

Total length @ 260 gpm = 188'

Per Curve, p B-14

Vel @ 540 gpm = 6 ft/sec  \( \tau = \frac{10}{6} = 1.7 \text{ sec} \)

Vel @ 260 gpm = 2.9 ft/sec  \( \tau = \frac{1.08}{2.9} = 64.8 \text{ sec} \)

Total transport lag = 66 sec

Shell Volume: Say = 30" ID x 20' long

\[ V = \pi \left( \frac{1}{4} \right)^2 20 = 98 \text{ ft}^3 \]

360 tubes x 20' x 1" OD take up

\[ 360 \times 20 \times \pi \left( \frac{1}{4} \right)^2 = 39.2 \text{ ft}^3 \]

So Shell holdup volume = 59 ft\(^3\) = (59)(7.48 gal/ft\(^3\)) = 442 gallons.

Shell Holdup Time = \( \frac{442 \text{ gal}}{260 \text{ gpm}} = 1.7 \text{ min} = 102 \text{ sec} \)

Tube holdup time = \( \frac{(1.6)(20) \text{ ft}^3 \times 7.48 \text{ gal/ft}^3 \times 60 \text{ sec/min}}{310 \text{ gpm}} = 46.3 \text{ sec} \)
More Cooling Tower - 8 - SBall

4/26/63

- Tower Bypass Valve Characteristics

If drop across tower is fixed at head of water (6.5'), or 2.8 psi, then (except when valve is wide open), the
ΔP value will be 2.8 psi. Thus

\[ Q = \frac{C_v \sqrt{ΔP (62.4)}}{62.1} = 1.68 \, C_v \, \text{gpm} \]

\[ Q_{\text{TOWER}} = 540 - Q_{\text{VALVE}} \]

In order to have a constant gain control, the trim on the valve should be linear, not e.g. 20.

- Piping between Mixing pt # JCT E5D & E53 downstream of
  pumps = 12' of 6" pipe

@ 540 gpm = 26 ft/sec = 2 sec transport lag

Regardless of Tower Characteristics, control of temp.

would involve simple mixing of Hot by-pass water
with cold basin water, & the only lags in the
system would be:

~ 2 sec transport lag
~ 3 sec T.C. for thermal well response
~ 5 sec T.C. for Valve operator
~ 2 sec T.C. for Controller

The gain of the controller would depend on

the valve position (higher gain @ more by-pass flow
with equal percentage trim), & on difference between
Tower return water temp. and basin temp.

eg: if Tower Return water temp = 95°F & Basin temp = 75°F,
a 100% ΔCv = 720°F ΔT; while if \( T_{\text{TRW}} = 90°F, T_b = 80°F, \)
a 100% ΔCv = 7 only 10°F ΔT,
Say control loop to have gain = 1.0 when phase lag = 120°.

From Bode Diagram, \( w_{120°} = 0.2 \text{ rad/sec} \), \( MR = 0.55 \)

So Max. System gain w/ controller = \( \frac{1}{0.55} = 1.8 \)

- Max. gain of equal-percentage valve (Proc. Inst. & Cont. HB, p.10-2)
  \[ \frac{\% \Delta \text{Flow}}{\% \text{Stem travel}} \approx 3.0 \]

- Say max. \( \Delta T \) return water – basin \( \approx 95 - 45 = 50^\circ F \)

- Say temp. controller span = 50°F

Then \( PB = 50\% \) would be good starting pt

React times of 1.0 min or greater wouldn't affect stability very much

To control fans:

1. If manual control of fans is preferred, an alarm could be given if:
   a) Control valve were fully closed or
   b) Controlled temp > 85°F,

indicating that a fan should be turned on.

AND

Another alarm could be given if the control valve were by-passing more than say 50% flow, indicating that a fan could be turned off.

2. Or: All this could be done automatically
FREQUENCY RESPONSE DIAGRAM

MSRE COOLING TOWER WATER TEMP. CONTROL

ASSUMES:
1. 2 sec. TRANS. LAG
2. 5 sec. TC - VALVE OPERATOR
3. 5 sec. TC - CONTROLLER
4. 5 sec. TC - WELL (TROUGH)

FROM FREQUENCY RESPONSE DATA

TRANSAIRE GAS SYSTEM

1. 5 sec. TC, 50° F SPAN, 0-150° F LIMIT
2. 1/8" ID BULB IN WELL
   WITH THERMOSPEED SLEEVE
3. WATER VEL = 1.15 FPS

(COMPARE D.B. FIG. 32)
Per Rm. it would be nice to have a simple-type all-outdoor detector & controller, & even nicer to have a temperature bulb in a well so it could be removed "on stream" for maintenance.

Per Taylor Eng Response: N.B.: Fig 32-5, a Gas-filled bulb in a "Thermospeed" well & sleeve has a primary TC of ~ 13 sec. Adding the characteristics Fig 32-5 to the previous composite (the 3 sec TC for orig. well is thrown in for good measure), the red curve on p. 10 is obtained.

\[ W_{120^\circ} = .083 \text{ rad/sec}; \quad MR = .60 \]

So controller gain would be about the same as before, but response time would be slower.

(Osc. cycle time ~ 40 sec/cycle)

Reset times of 2.0 min. or greater could probably be used without affecting stability.

(& with Speed Act: (Set for 15 sec.).

Faster response could be obtained, so I say)

Controller: Taylor Fulacope indicating Controller
Type 164 R. (PB, Reset, Derived)

(An equivalent.)
From Bode Diagram of System w/reset contr. & Block
Nichols chart, Controller gain shown should be reduced
by x0.7 to give critically damped response.
Time constant of response = 1/0.1 = 10 min.
Modified controller gain = 1.1, i.e. if Temp. span of
Primary & secondary controllers are equal, P.B. = 90%
FREQUENCY RESPONSE DIAGRAM

MSRE COOLING TOWER - CASCADE CONTROL FROM H.E.X. TREATED WATER OUTLET TEMP.

WITH RESET CONTROLLER
RESET TIME = 10 MIN.

COMPOSITE W/O CONTROLLER

S. J. Ball
4/20/63
-.Cascade Control
  1. Taylor Pneumatic Set Indicating Controller
     with PB, Reset, Deriv; & gas bulb in well
     $660.58
  2. Taylor Fulscope Indicating Contr. Type 164R
     with PB, Reset, Deriv; & gas bulb in well
     $550.58
     $12.00
     $1500

- With Control Valve in CTW line to H. Ex.:
  2 ea of Item 2, above
  $11.00
  $13.00
  $45.00
  $17.50
  (+ valve installation)

- No Control of Treated Water Temp
  1 ea of Item 2, above
  $550.58
  $600

Per Ralph Graymon (4/30/63), "Let's take the $600 model now, test the system to see if the (~3°F) change in treated water temperature screws up the heat rate calculation, and if it does, add the system with the control valve -- & look at using a ~2'' by-pass line w/ value, rather than throttling in the 6'' line."
FREQUENCY RESPONSE DIAGRAM

MSRE Cooling Tower Water Temp Control
T/TC = B58

TRANSPORT LAG = 2.0 SEC.
VALVE OPERATOR = 5 SEC. T.C.
CONTROLLER = 2 SEC. T.C.

GAIN = 1
(GAIN CAN BE 1.4)

THERMAL BULB IN WELL: PER FIG. 3.25 IN TAYLOR FREQUENCY RESPONSE N.I.B.

CLOSED LOOP FOR CONTR GAIN = 1.0
GAIN = 1.4

2.85 SEC. TO TAYLOR BULB IN WELL

COMPOSITE OPEN LOOP
APPROXIMATE STEP RESPONSE OF TAYLOR THERMAL BULB IN WELL

(3/8" WELL IN THERMOSTEP SLEEVE, WATER VELOCITY ~1 FPM)

CHARACTERISTICS ASSUMED FOR

MScE COOLING TOWER WATER CONTROL STUDY

% RESPONSE

T, SECONDS

S.J. Ball
7-21-64