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REPORT 974-007

CALCULATING THE EFFICIENCY OF A TANK WITH AN INCORPORATED THERMAL CONTROL

STATINTL

February 1965

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ABSTRACT

Among the major objectives of this Processor Development Program are those to reduce equipment size, to reduce power consumption, and to investigate the modular concept of processor design. In the HTA-5 Processor, the concept of separate service units in which all support equipment was mounted led to increased installation space and decreased efficiency. This assignment, in conjunction with assignment 974-008, investigates the possibility of eliminating the service units and separate temperature-control equipment as a step towards meeting these objectives.

INTRODUCTION

1. CONVENTIONAL TEMPERATURE CONTROL

One conventional method of regulating solution temperature in a processing tank,is to provide a control system in which solution is drawn from the tank,and pumped through a coil-type heat exchanger and through a small tank containing heating units. The solution is then returned to the processing tank through a specific recirculation system designed to eliminate striation (Figure 1-1).

Chilled water at a temperature of about 45°F is circulated through the jacket of the heat exchanger to extract heat from the solution as it passes through the coils. Should the solution already be below control temperature, its temperature is automatically raised by electrical resistance heaters as it passes through the small tank.

The temperature of the solution in the processing tank is continually monitored by a temperature probe mounted in the tank. The probe compares this temperature against that set in a Wheatstone bridge or other control circuit. An increase in the temperature of the solution in the tank causes a circuit to open a valve in the chilled water line to the heat exchanger. Conversely, a drop in the temperature of the solution will cause a circuit to close a solenoid switch and operate the heaters in the small tank. This equipment is both bulky and inefficient.

TECHNICAL DISCUSSION

2. INTEGRAL THERMAL CONTROL

To provide a built-in thermal control while permitting liquid bearings to be installed at first seemed to present a technically difficult problem. However, the concept of using the bearing pump for recirculation offers one approach to the problem (Figure 2-1). If the processing tank were divided into two sections by a heat-exchanging wall and the bearings were mounted in this wall, a new mode of operation could be attained. Then,solution from the bearing side of the tank could be pumped into the pressure side of the heat exchanger wall and thence returned through the bearings to the front section of the tank. Recirculation from one side of the tank to the other would provide a flow of liquid around the heat exchanger wall, with the added advantage that little energy would be lost to the ambient environment. To further improve thermal efficiency, the module (of which the tank would be an integral part) would be insulated to reduce heat loss to the atmosphere.

To maintain an 88°F solution temperature with a 65°F room ambient, a heating load of approximately 3530 BTU/hr is required. This does not consider the heat from the bearing pump which, for the purpose of this study is 3 horsepower, which is rated at 7635 BTU/hr. On the other hand, to maintain a solution temperature of 68°F in a 75°F room ambient with a 75°F solution replenishment temperature, 8720 BTU/hr would be required as a cooling load.

It is safe to assume that chilled water at 45°F and hot water at 160°F are available in most modern processing laboratories at flows of 1-3/4 and 3/4 gpm respectively and at 10°FTD. The tank size selected (Figure 2-1) provides a surface area of 4.95 square feet for heat exchange. This area can handle a 9000 BTU/hr load, which is very close to the required 8720 BTU/hr cooling load.

If the heat exchanger is used as the back wall of the tank, then the pulldown rate is approximately 1.4°F per hour. If the bottom surface of the tank is used as heat exchanger, the U factors could be a little higher. In this case, however, an area of only 3.1 square feet and 8350 BTU/hr would be available. The 3-horsepower pump could be used to obtain steady state cooling, but the pulldown rate would be only 0.5°F per hour.

The following calculations assume the use of a plate-coil wall comprised of 3/8 inch single-embossed small tubes. The pressure drop is high, but such devices could be employed in parallel circuits to reduce the losses. Several types of circuitry can be used for temperature control. On the assumption that both chilled and hot water are available in the quantity required, the system illustrated in Figure 2-2 offers a clean compact unit with a minimum of outside controls.

Parameters:

(1) Tank Capacity: 75 gallons.

(2) Length of Film in Tank: 44-1/2 feet.

(3) Approximate Film Transport Speed: 20 fpm.

(4) Development Time Required: 2 minutes.

(5) Bearing Diameter: 2 inches.

(6) Operating Temperature: 60° to 88°F.

(7) Bearing Flow (Assumed): 12 gpm.

(8) 13 Bearings per Tank: 156 gpm total.

(9) Replenishment Rate (Assumed): 5 to 10 gpm.

(10) Tank Insulation: 1 inch of Styrofoam.

(11) Ambient Temperature Range: 65° to 75°F at 50 percent RH.

Total Volume 38" x 32" x 14"
= 17024 Inches.^{3}
= 9.85 Feet^3 (use 10)

Gallons 9.85 x 7.48
= 73.6 gallons (use 75)

Weight 73.6 x 8.345
= 615 lbs. (use 625)

The following mathematical analysis assumes the use of stationary sleeve-type liquid bearings. However, assignment 974-013, which considers a type of bearing incorporating a built-in pump, would influence the parameters given to the extent that no bearing pump would be required. Therefore, the 3-horsepower heat input load from the pump would not exist. Furthermore, the idea of a rear pressure compartment would not be required. A further analysis, using the bottom and sides of the tank, will be made when the processor module now under design study (assignment 974-008) has advanced sufficiently.

To find heat losses during high temperature processing (88°F) assume all sides and bottom are insulated.

$$A_{si} = 2 \times \frac{38 \times 32}{144} + 2 \times \frac{18 \times 38}{144} + \frac{18 \times 32}{144}$$

$$= 16.9 + 9.5 + 4.0 = 30.4 \text{ ft}^2 \text{ (inside diameter)}$$

$$A_{so} = 2 \times \frac{39 \times 33}{144} + 2 \times \frac{19 \times 39}{144} + \frac{19 \times 33}{144}$$

$$= 17.85 + 10.3 + 4.35 = 32.5 \text{ ft}^2 \text{ (outside diameter)}$$

$$A_{s_{\text{top}}} = \frac{4 \times 32}{144} = 0.89 \text{ ft}^2 \text{ (could be insulated too.)}$$

$$A_{s_{\text{open surface}}} = \frac{14 \times 32}{144} = 3.1 \text{ ft}^2$$

Let $A_s = 33 \text{ ft}^2$

$A_{s_{\text{open}}} = 3.1 \text{ ft}^2$

$U = 0.35$ BTU/hr - ft^2 - °F (1" Styrofoam)

$$Q_{\text{sides}} = UA_s\Delta t = (0.35)(33)(\underset{\text{High Soln}}{88} \overset{23}{-} \underset{\text{low room}}{65}) = 266 \text{ BTU/hr}$$

$U^1_{\text{top sens}} = 300$ BTU/hr - ft^2 @ 23°TD Still air cond. (Unvented)

$U^1_{\text{top sens}} = 300 \times 1.8 = 540$ @ 23°TD and 300 fpm (vented)

Assume normal B & W operation with no venting

$Q_{\text{top sens}} = 300 \times 3.1 = 930$ BTU/hr

$Q_{Latent} = KU'' A_s$

$U'' = 95$ BTU/hr-ft^2 @ 88°F surf and 75° Room

f @ 65°F room = 1.4

$KU'' = 1.4 \times 95 = 133$

$Q_{Latent} = 133 \times 3.1 = 412$ BTU/hr

$Q_{Replen} = wc_p \Delta t$

Let W = 10 gph = 83.4 #/hr

$C_p = 1.0$ BTU/#-$^{\circ}$F

$\Delta t = 88 - 65 = 23^{\circ}$F (assume replen supply @ room temp.)

$Q_{Replen} = 83.4 \times 1.0 \times 23 = 1920$ BTU/hr

$Q_{\substack{Total\\Heating}} = Q_{\substack{wall\\loss}} + Q_{\substack{top\\sens}} + Q_{\substack{top\\lat}} + Q_{replen} - Q_{pump}$ Neglect = O (Act = 7632 B/H)

$= 266 + 930 + 412 + 1920 = 3528$ BTU/hr

NOTE: If walls of tank were <u>not</u> insulated

$Q_{wall} = UA\Delta t$ where $U \cong 2.0$ or $6 \times U_{insul}$

$Q_{wall} = 2 \times 33 \times 23 = 1515$ BTU/hr

$Q_{tot} = 4777$ BTU/hr or $\frac{1249}{3528} =$ 35% incr w/o insul.

USE INSULATION

To find max heat gain or cooling loads for low temp processing (68°)

Let surf areas & U-factors = approx those stated previously

$Q_{\substack{wall\\gain}} = UA_s \Delta T = (0.35)(33)(75^7-68) = 81$ Btu/hr

$Q_{\substack{top\\sens}} = U^1 A_s \Delta T = 15 \times 3.1 \times 7^{\circ} = 326$ Btu/hr

$Q_{\substack{top\\lat}} = U^{11}A_s = 30 \times 3.1 = 93$ Btu/hr
Actually 0

$Q_{replen} = Wc_p\Delta T = 83.4 \times 1 \times 7^o = 585$ Btu/hr

$Q_{pump} = 2544 \times hp = 2544 \times 3 = \underline{7632}$ Btu hr

$Q_{\substack{Total\\cooling}} = 8717$ Btu/hr

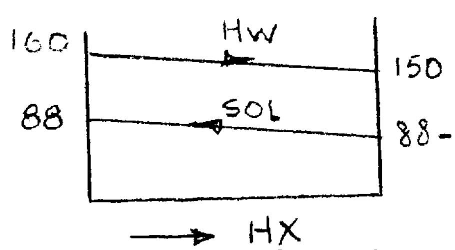
Assume hot & cold water can be supplied to HX at 160^oF & 45^oF respectively.

W_{HW} Reqd $= \dfrac{Q}{c_p\Delta T} = \dfrac{3600}{1.0 \times 10^o \times 500} = 0.72$ GPM

W_{CHW} Reqd $= \dfrac{8720}{10^o \times 500} = 1.75$ GPM

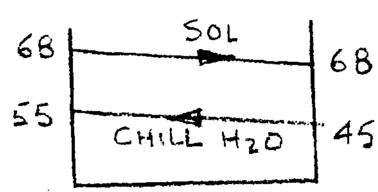
HX LMTD's where $\Delta T_{SOL} = \dfrac{Q}{W_{SOL}c_p} = \dfrac{8720}{156 \times 500 \times 1} \approx .11^oF$

HEATING



LMTD Heat = 67^o

COOLING



LMTD Cool = 18^o

NOTE: To determine surf area reqd, factors are LOAD (Q), U Factor, & LMTD

$Q = UA\ LMTD$

For given U

As reqd for heating $= f\left(\dfrac{Q}{LMTD}\right)$

$= \dfrac{3600}{67} = \dfrac{53.6}{U}$

& A_s reqd for cooling $= \dfrac{8720}{18} = \dfrac{484}{U}$

Therefore cooling requirement is max for A_s determination.

Surf area available in present tank design

$$A_s \text{ avail} \approx \frac{38 \times 32}{144} = 8.45 \text{ ft}^2$$

if U = 100

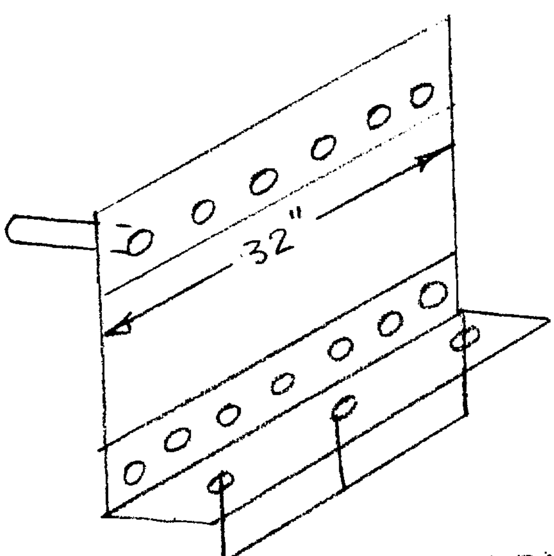
then Q = UA LMTD = (100) (8.45) (18) = 15,200 Btu/hr Cooling

& Q = (100) (8.45) (67) = 56,500 B/hr heating

Apparently plenty of surface is avail in present design for heating; cooling is ZX

Now check U-Factor assumed

$$U = \frac{1}{\frac{1}{h_{sol}} + \frac{t}{K_{ss}} + \frac{1}{h_{wat}}} \quad \text{where } \frac{t}{k} = \frac{.06}{115} = .00052 \quad \left(\frac{k}{t} = 1920 \right)$$



At 12 GPM/bearing the bottom bearings will have a tendency to be short circuited from largest HX surface

A_s avail to bottom bearing flow $= \frac{3'' \times 32}{144} = 0.665 \text{ ft}^2$

$$V_{sol} = fps = \frac{cfs}{ft^2} = \frac{\overset{\text{GAM}}{156 \times 2.29 \times 10^{-3}}}{4 \times 32/100} = \frac{156 \times 2.29 \times 144}{128 \times 1000}$$

$V_{sol} = 0.402$ fps ; $\underline{h_{sol} = 400}$ & $h_{sol} = 200$ upper portion

Assume use of single embossed, small pass platecoil

$\approx$ 3/8" OD pipe; where $a_{cs} = 0.1 \text{ in}^2 = .000695 \text{ ft}^2$

$$V_{cw} = fps = \frac{cfs}{ft^2} = \frac{.25 \times 2.29 \times 10^{-3}}{.000695} = \frac{.25 \times 2.29}{.695} =$$

$V_{cs} = 0.82$ fps @ 0.25 GPM ; $\underline{h_{cw} = 250}$

$V_{hw} = 2.46$ fps @ 0.75 GPM ; $\underline{h_{hw} = 500}$

$$U_{cooling} = \frac{1}{\frac{1}{400} + .00052 + \frac{1}{250}} = \frac{1}{.0025 + .00052 + .004}$$

$$= \frac{1}{.00702} = \underline{142}$$

$$\& \; U_{heat} = \frac{1}{.0025 + .00052 + .002} = \frac{1}{.00502} = \underline{199}$$

Below lower bearings only

To find U-factor on wall between bearings

$$U_{cool} = \frac{1}{\frac{1}{200} + .00052 + \frac{1}{250}} = \frac{1}{.005 + .00052 + .004} = \frac{1}{.00952}$$

$$= \underline{105}$$

$$U_{heat} = \frac{1}{.005 + .00052 + .002} = \frac{1}{.00752} = \underline{133}$$

$$Q_{cool} = Q_{low\ as} + Q_{Hi\ as}$$

$$= \left[(142)(.665) + (105)\left(\frac{32+28}{144}\right) \right] 18^{\circ} \text{ LMTD}$$

Getting close to reqmnt

$$= \left[94.5 + 405 \right] 18 = 9{,}000 \text{ Btu/hr cool}$$

$$\& \ Q_{heat} = \left[(199)(.665) + (133)(4.29)\right] 67'' \text{ LMTD}$$

$$= \left[132 + 570\right] 67 = \underline{47,000 \text{ Btu/hr Heat}}$$

To get above cooling or heating, find flow rates!

$$W_{chw} = \frac{9000}{10^{o} \times 500} = 1.8 \text{ GPM}$$

NOTE: Then $V_{chw} = \frac{1.8}{.25} \times .82 = \underline{5.9 \text{ fps}}$ (Good)

$$W_{hw} = \frac{47,000}{10'' \times 500} = 9.4 \text{ GPM}$$

NOTE: $V_{hw} = \frac{9.4}{1.8} \times 5.9 = \underline{30.8 \text{ fps}}$ (Too Hi)

$\Delta p \geq$ ——— Assuming 26 passes X2-1/2' long

$\Delta p = 7X.42$ psi x (26 x 2.5 + 50 x 3)

$= 7 \times .42 \times 212 = 625$ psi

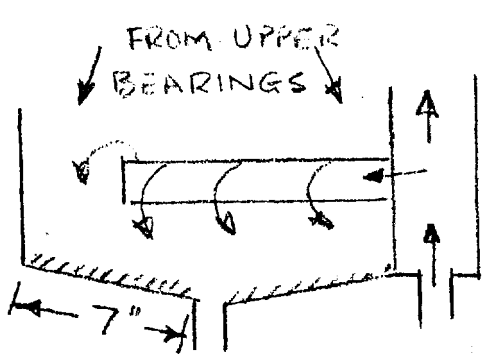
@ 1.8 GPM for cooling

$\Delta p = 7 \times .025$ psi/ft $= 1.75$ psi/ft x 212 = $\underline{370 \text{ psi}}$

Way too high

This HX will need parallel flow paths and increased flow rates to maintain steady-state velocity & h factors.

Try HX at bottom of tank (cleaning tank may be a prob.)



A_s avail $= \frac{14 \times 32}{144} = \underline{3.1 \text{ ft}^2}$

$V_{sol} = \frac{156 \times 2.29 \times 144}{4 \times 14 \times 1000} = 0.92$ fps

$h \approx 450$

U_C = 150 for cooling

U_H = 200 for heating

8720 Reqd: not enough surface!!

Q_{cool} = (150)(3.1)(18) = 8,350 Btu/hr

Q_{heat} = (200)(3.1)(67) = 41,500 Btu/hr

Q avail by A_s

Add pump HP to cooling loads

W_{sol} = 156 GPM

P = 20 psi (?)

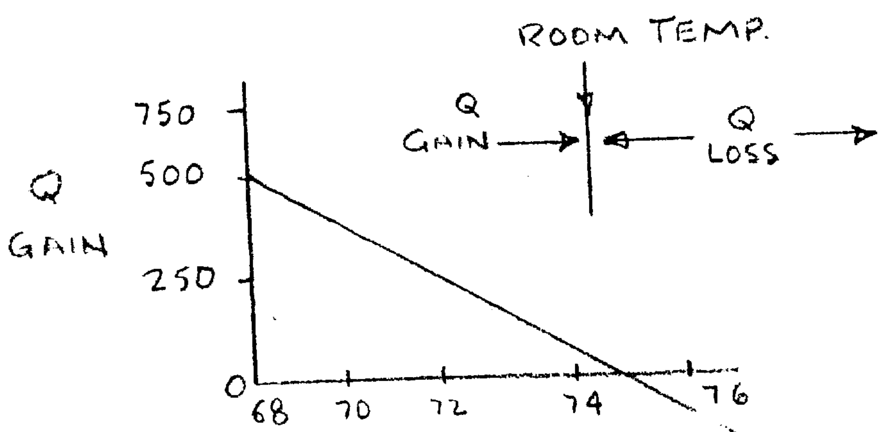
$$\text{bhp} = \frac{\text{GPM} \times P}{1714 \times N_{eff}} = \frac{156 \times 20}{1714 \times .6} = 3.02 \text{ hp}$$ (1st try a 3HP pump is antic.)

$\therefore Q_{pump}$ = 3x2544 = 7632 Btu/hr

What is pulldown rate w/ 9000 B/hr HX?

$Q_{wall \,\&\, top\ gains}$ = 0 @ 75° solution W/ 75° room

= 500 B/hr @ 68° sol. W/ 75° room



Assume $Q_{replen} = 0$ since system not in oper.

let $Q_{pump} = 7630$ B/HR

$\therefore$ $Q_{total\ gain} = 8130$ B/HR @ 75°F start

$Q_{avail} = 8130 + mcp\ \Delta T/hr = 9000$

$\Delta T/hr = \frac{870}{625 \times 1} = 1.4^{\circ}F/hr$

Pulldown from $75 \rightarrow 68^{\circ} = \frac{7}{1.4} = \underline{5 \text{ hrs}}$

Using 8350 Btu/hr bottom type HX

What is pulldown time?

$Q_{start} = Q_{pump} = 7630$

$Q_{end} = Q_{pump} + Q_{gain} = 8130$

$\Delta T/hr = 8350 = 7630 = \frac{720}{625} = 1.15^{\circ}F/hr$ @ start

$\frac{7}{1.15} = 6+$ hrs

$\Delta T/hr = 8350 - 8130 = \frac{220}{625} = 0.35^{\circ}/hr$ @ end

$\frac{7}{.35} = 20$ hrs

$\overline{\Delta T}/hr$ actual $=$ Avg $= \frac{6+20}{2} = \underline{13 \text{ hrs}}$

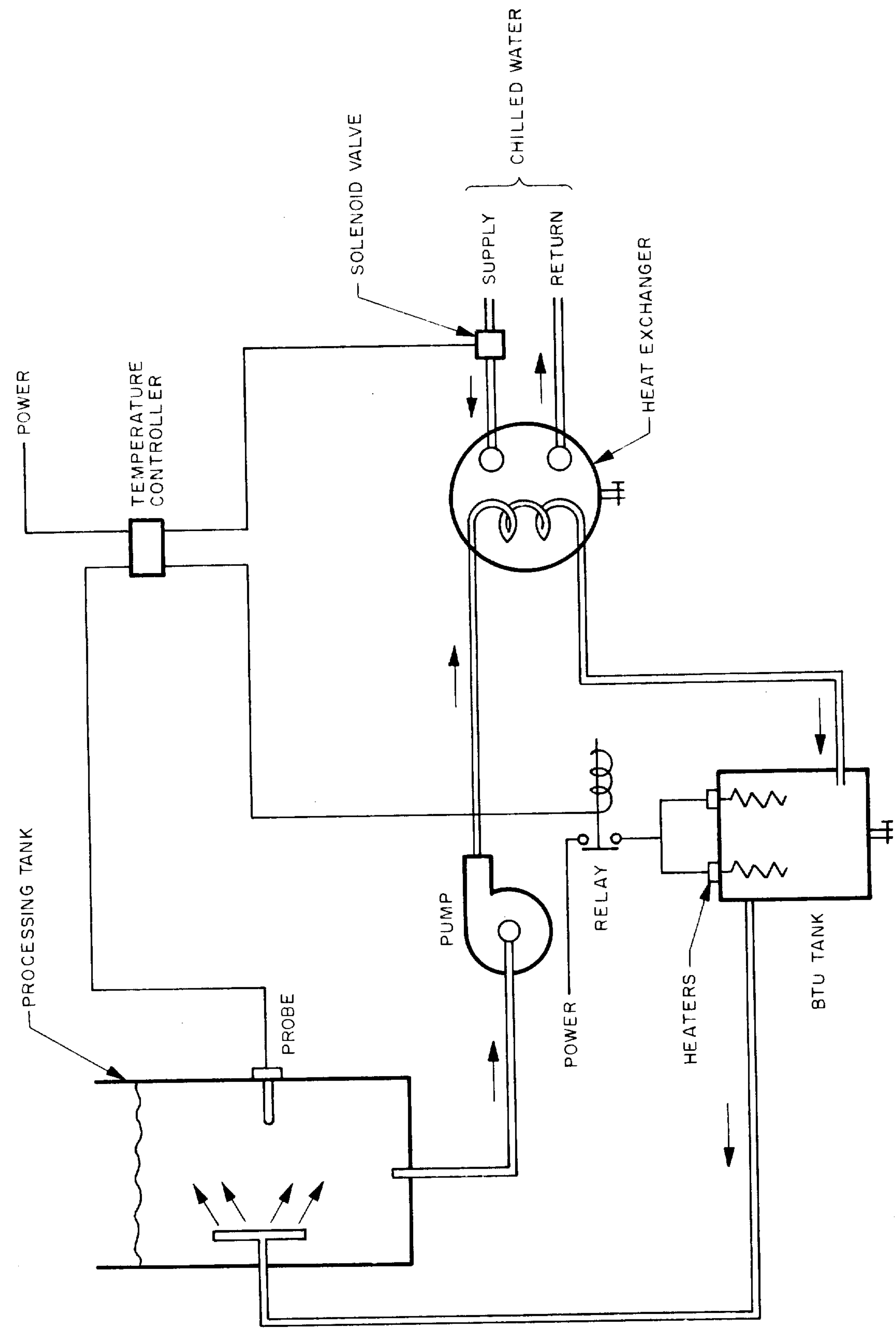

CHILLED WATER
SOLENOID VALVE
SUPPLY
RETURN
HEAT EXCHANGER
POWER
TEMPERATURE CONTROLLER
PROCESSING TANK
PROBE
PUMP
RELAY
POWER
HEATERS
BTU TANK

FIG. 1-1

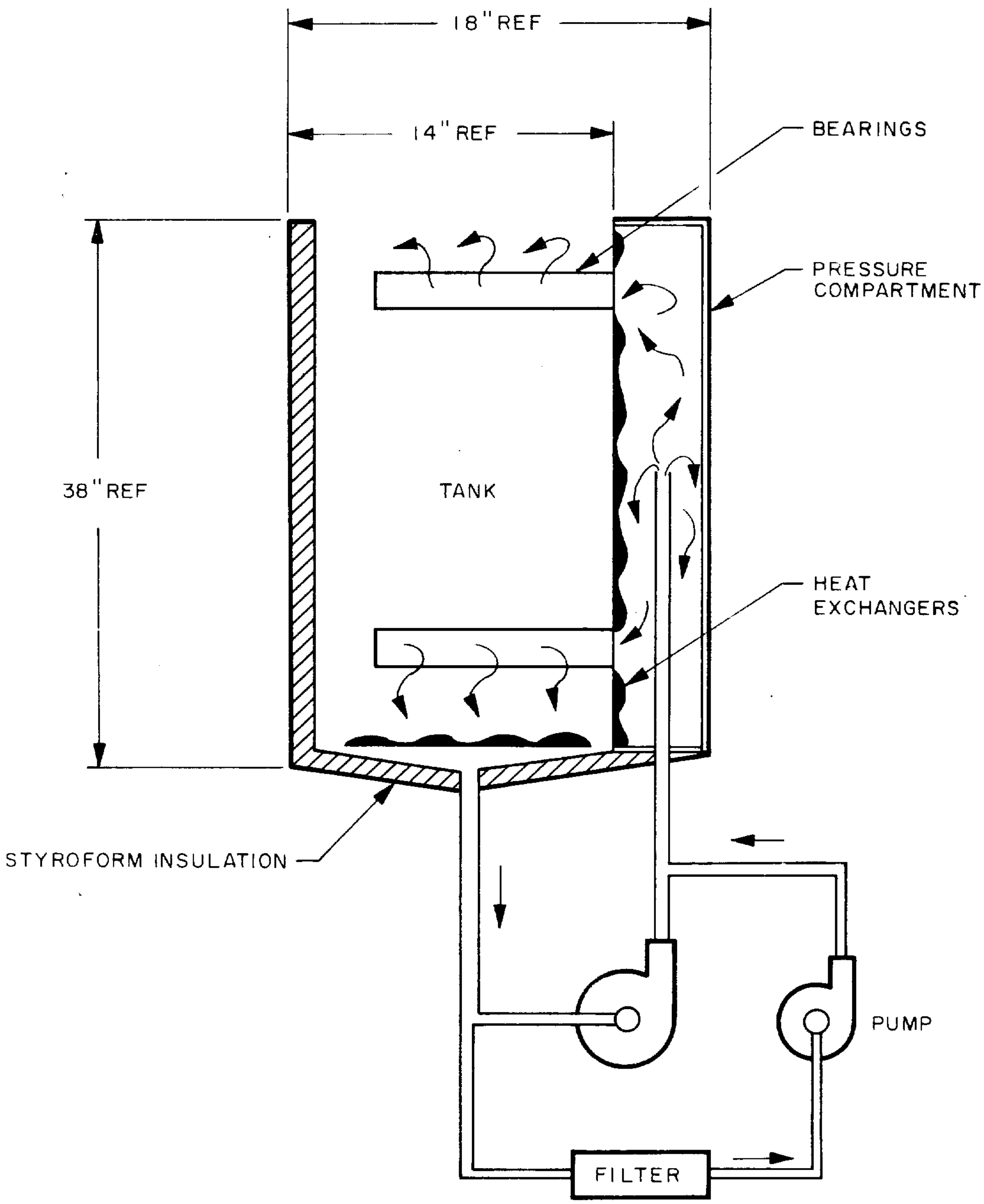

18" REF
14" REF
BEARINGS
PRESSURE
COMPARTMENT
38" REF
TANK
HEAT
EXCHANGERS
STYROFORM INSULATION
PUMP
FILTER

FIG. 2-1

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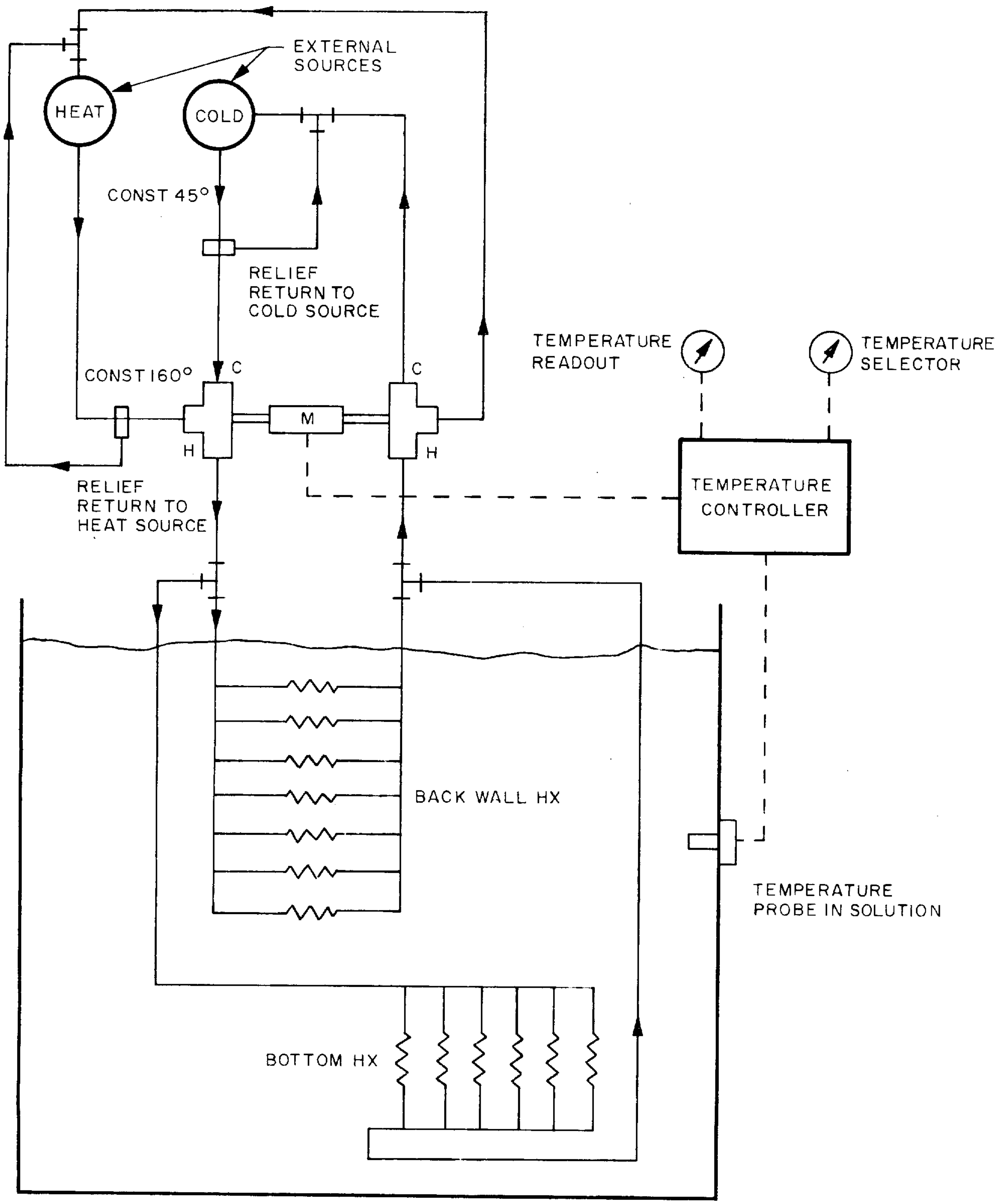

FIG. 2-2

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