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METALLURGICAL PROJECT

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TECHNOLOGICAL RESEARCH P-6 STUDIES SECTION

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THERMAL SYPHON IN HOMOGENEOUS PILE

by

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December 2, 1943

* * *

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I - SUMMARY

Problem Under Investigation

The circulation of the P-9 oxide slurry from the homogeneous pile through the heat exchangers by means of pumps introduces mechanical problems which may be eliminated by the use of a thermal syphon. In this type of a system, density differentials, due to temperature changes, provide the motivating force. It was the purpose of this investigation to determine the practicability of such a scheme from the point of view of utilization of P-9. The system considered herein is that of a horizontal heat exchanger placed directly above the pile, fed from the top of and delivering to the bottom of the reaction tank.

Important Results

The choice of heat exchangers was limited to the sizes recommended by the Andale Company, but the highest utilization occurred using tubes shorter than the maximum allowable length. Tubes, 5/8 inch in diameter, 12 feet long in a 4 ft. diameter shell gave a utilization of about 9700 KW/ton of P-9 external to the pile. The pressure differential causing the flow is about 45 lbs./sq.ft. Five such exchangers in a horizontal plane above the pile will dissipate 180,000 KW using 20 tons of P-9 external to the pile. Two streams of cooling water were used in each exchanger, entering at 15° C. It appears impractical to increase the cooling water rate above that which keeps the temperature difference constant in each section of the exchanger.

Recommendations

Since utilizations calculated for the thermal syphon (approximately 3,000 KW/ton P-9 total, based on 40 tons P-9 in the pile) are considerably smaller than those reported by Ward and Weills (CT-969) for the case of pumping the slurry (up to 20,000 KW/ton), its use does not appear attractive. However, the order of magnitude of the pressure differential necessary to circulate the slurry in the best case calculated is so small (approximately 40-50 lbs./sq.ft.) that a slowly moving pump, possibly an axial flow type, might considerably improve the utilization without introducing serious erosion difficulties. It is proposed that the possibility of such a system be investigated.

It is further proposed that the possible advantages of elevating the heat exchangers slightly in pump-circulated plants, to remove the heat generated after shutdown, in the event of pump or power failure, be investigated.

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II - GENERAL DISCUSSION

Origin of Problem

The most practical method of removing heat from the homogeneous P-9 pile appears to be that of circulating the slurry through external heat exchangers. The driving force for this circulation may be a pump, the use of which will probably involve difficulties of erosion, stuffing box problem, etc., or a thermal syphon, in which there would be no moving parts.

The operation of the thermal syphon depends upon the change in density of the fluid with temperature. The simplest system may be conceived as a closed loop of piping in a vertical plane, one vertical leg of which is heated and the other cooled. The difference in density results in a pressure differential causing the fluid to flow around the loop. The flow rate depends on the densities involved and on the frictional losses resulting from this flow.

Such a system may be adopted to the homogeneous pile as follows: a pipeline leads from the top of the pile tank to a heat exchanger and returns to the bottom of the tank. The flow rate in this case depends on the geometrical location of the equipment, the temperatures involved and the frictional losses in the lines and equipment pieces. If this type of system could dissipate the required power without a large holdup of the critical materials, an attractive solution to the slurry pumping problems would be furnished.

III - PRESENT CONDITIONS

Previous Work

Some calculations on the performance of a thermal syphon have been made for this pile and the results of these were reported (CT-962) These calculations, however, assumed the density of the slurry in the pile tank to be that at the hot temperature, an optimistic assumption, and considered only vertical heat exchangers. If horizontal exchangers, located on top of the pile tank were used, however, the whole length of the cold "leg" would be at the temperature of maximum density, instead of varying over a range of temperatures, thereby increasing the pressure differential which motivates the flow.

Causes Prompting Investigation

Any mechanical pumping device will eventually wear out and cease to be of any use. Since the pile pumps will be inaccessible for maintenance and repair, it may be seen that the useful life of the pile may depend on the useful life of the pumps. With pumps eliminated, a much longer pile life can be expected. This investigation was prompted by the judgment that elimination of the pumps could be tolerated only if the price in terms of P-9 utilization was acceptable. An extension of the work of J. J. Goett (CT-962) was required to amplify the information available.

IV - INVESTIGATION

System

The system under consideration consisted of the pile tank and a horizontal heat exchanger in the optimum position, viz., directly above the center of the tank. The hot pipeline from the upper part of the tank to the heat exchanger and the cold pipeline from the exchanger to the bottom of the tank were located in the vertical plane determined by the axis of the exchanger and the axis of the pile tank. All calculations were made for this optimum heat exchanger, letting its height above the tank be a variable. The pile tank was taken as a hemisphere 12 feet in diameter, surmounted by a cylinder of the same diameter and six feet high.

Mechanical Energy Balance

When the system described above is in operation, with all temperatures fixed, the rate of flow may be determined by applying the principle of conservation of energy, taking as the "system", one pound mass of slurry and following it around the cycle. Since when the steady state is reached, the two heat effects on this unit of slurry are equal and opposite in sign, and because there is no external work effect, the energy balance for a differential length of path becomes:

$$v\,dp + d\,x + \frac{u\ du}{g} + dF = 0$$

where

v = specific volume of slurry - cu.ft./lb. mass

p = pressure - lb.force/sq.ft.

x = height above datum plane - ft.

u = velocity - ft./sec.

g = conversion factor = $32.2\ \frac{\text{ft. (lb.mass)}}{\text{sec.}^2\ \text{(lb. force)}}$

F = pressure loss due to friction - $\frac{\text{ft.(lb.force)}}{\text{(lb. mass)}}$

Letting ρ = density - $\frac{\text{lb. mass}}{\text{cu. ft.}} = \frac{1}{v}$

this may be rearranged to give

$$dp + \rho\,dx + \rho\,\frac{u\ du}{g} + \rho\,dF = 0$$

Integrating around the loop gives

$$\oint dp + \oint \rho\,dx + \oint \rho\,\frac{u\ du}{g} + \oint \rho\,dF = 0$$

Since p is a point function, $\oint d\,p = 0$, and

$$\oint \rho\, dx + \oint \rho \frac{u\ du}{g} + \oint \rho\, dF = 0$$

Taking as our datum plane the point where the cold pipeline enters the pile tank, we let

H = height of axis of exchanger above datum - ft.

D = height of point where hot pipeline enters tank above datum - ft.

Starting around the loop from a point at the inlet to the hot pipe,

$$\oint \rho\, dx = \rho h\,(H - D) + \rho c(-H) + \bar{\rho}(D) = D\,(\bar{\rho} - \rho h) - H\,(\rho c - \rho h)$$

where ρh = density at hot temperature

ρc = density at cold temperature

$\bar{\rho}$ = average density in pile tank

For the kinetic energy term,

$$\oint \rho \frac{u\ du}{g} = \frac{\rho h}{2\ g}\left[u_t^2 - u_p^2\right] + \frac{\rho c}{2\ g}\left[u_p^2 - u_t^2\right]$$

where u_p = velocity in pile tank

and u_t = velocity in tubes of heat exchanger

[Actually, the u_t^2s in the above equation are slightly different because of the change in density of the fluid as it passes through the tubes. Strictly speaking, another term should be added to account for the change in velocity along the tube length. However, as shown later, this whole expression is of negligible importance.]

Now u_p is essentially zero, so we may write,

$$\oint \rho \frac{u\ du}{g} = -\frac{u_t^2}{2\ g}\left(\rho c - \rho h\right)$$

Since tube velocities encountered are small and the density difference is small and frictional losses are of the order of $\rho \frac{u^2}{2\ g}$, this whole term has been discarded.

The frictional term $\oint \rho \, dF$ we may represent by $\Sigma \rho_n F_n$, the summation of the product of each frictional loss and the corresponding density at that point. The frictional losses may be enumerated as follows:-

1.) Friction in hot pipeline

2.) Expansion - hot pipe to header of exchanger

3.) Contraction - header to tubes

4.) Friction in tubes (average density used here)

5.) Expansion - tubes to header

6.) Contraction - header to cold pipe

7.) Friction in cold pipeline

8.) Expansion - cold pipe - pile tank

9.) Contraction - pile tank to hot pipeline

The mechanical energy balance, therefore, becomes:

$$\Sigma \rho_n F_n = H(\rho_c - \rho_h) - D(\bar{\rho} - \rho_h).$$

The procedure used consisted in arbitrarily fixing the rate of flow, temperatures, and pipe sizes, and then finding the value of H, which satisfied the energy balance. The value of D is fixed by the pile tank dimensions. Curve A of Figure I (p.14) shows the extrapolated values of the density of the slurry as a function of temperature.

The method used to calculate $\bar{\rho}$ is shown in the Appendix.

Heat Transfer Calculation

Because this study was intended to show how to remove large quantities of heat in a small exchanger (for holdup reasons) and to cool to a low temperature (for circulations driving force), two streams of cooling water were used in each exchanger, each flowing countercurrent to the slurry. (This arrangement is, therefore, essentially two exchangers in series. See discussion by Ward and Weills, CT-969). An expression was derived to give the quantity of heat removed as a function of the inlet temperatures, flow rates, and transfer coefficients for a given exchanger. The expression is as follows:

$$Q = B(t_1 - T_1)\left(2 - \frac{B}{C_s W_s}\right)$$

in which

$$B = \frac{1 - a}{\frac{1}{C_w \; W_w} - \frac{a}{C_s \; W_s}}$$

in which $$a = \exp \frac{U_o \; A_o}{2} \left[\frac{1}{C_s \; W_s} - \frac{1}{C_w \; W_w} \right]$$

$$Q = \text{P.C.U./hour} \left[\frac{Q}{1900} = \text{KW} \right]$$

t_1 = slurry inlet temperature °C.

T_1 = cooling water inlet temperature °C.

U_o = overall heat transfer coefficient based on outside area - P.C.U./hr. - sq. ft. - °C.

A_o = Heat transfer area based on outside of tubes - sq. ft.

C = specific heat - P.C.U./lb. - °C.

W = Rate of flow - lbs./hr. (for cooling water, W is same in each section of exchanger).

subscripts

s = slurry

w = cooling water

This expression may be considerably simplified if the condition be imposed that the rise in temperature of the cooling water in each section of the exchanger be equal to the drop in temperature of the slurry in that section; i.e. $C_s \; W_s = C_w \; W_w$. With this simplification, the heat removal may be written as:-

$$Q = \frac{(t_1 - T_1) \; C_s W_s U_o A_o (U_o A_o + 4 \; C_s W_s)}{(U_o A_o + 2 \; C_s W_s)^2}$$

The effect of this limitation may be observed in the following table, calculated for 5/8" tubes, slurry velocity in the tubes of 2 ft./sec., $t_1 = 120°$ C. and $T_1 = 15°$ C.

Cooling Water Rate G P M	KW
1000	40,600
2960 *	57,000
3500	57,500
4000	59,000
5000	60,500
6000	61,500

* In this case $C_s W_s = C_w W_w$

It is seen that increasing the cooling water rate over that in the simplified case causes little change in the heat removed. However, higher rates will give lower exit temperatures for the slurry, and will result in a slightly higher utilization. For the purposes of this calculation, the simplified equations were used. It was also assumed that whatever the cooling water rate, the baffle spacing could be such as to provide the proper shell-side velocity.

The size of the exchanger was taken, in most cases, as the maximum size recommended by the Andale Company. These figures are given in the following table.

Tube Size in.	Maximum Tube Length - ft.	Maximum No. of tubes	Gauge	Diameter of Shell in.
1/4	12	1650	18	25
3/8	16	1850	16	25
5/8	16	2650	14	47
3/4	16	2100	14	47
7/8	16	1600	14	47
1	20	1250	14	47
1 1/4	20	760	14	47

Heat Transfer Coefficients

Standard correlations for the calculation of heat transfer coefficients were used, extrapolating the physical properties of the slurry from the available data. The cooling water velocity on the shell side was taken as the maximum recommended by heat exchanger manufacturers. A shell-side dirt film coefficient of 2000 was also employed. From this information, overall heat transfer coefficients were calculated for each exchanger as a function of the tube velocity and the temperature range of the slurry. Calculations of heats removed and cold-slurry temperatures then were made for the heat exchangers of size given in table above. The results for the larger sizes of tubes are given in Figure II. The effect of decreasing the tube length, L, (to decrease holdup) for a given size of tubes is shown in Figure III.

Calculation of Frictional Losses

Standard formulae for the pressure drop in pipes, tubes, elbows, expansions, and contractions were used. (See McAdams, Heat Transmission; Perry, Chemical Engineers' Handbook; Chapters on Fluid Flow.) Only the turbulent flow cases were explored.

Calculation of Holdup

For a given heat exchanger, a slurry velocity in the tubes was selected, giving the total flow rate, the heat removed, and fixing the temperatures and densities. D, the vertical distance between the points where the inlet and exit pipes intersect the tank was taken in all cases as 11 feet. For each of several pipe sizes, the frictional losses were totaled and the required values of H determined. The total length of pipe and the internal volume of the heat exchanger then allowed the calculation of the amount of slurry holdup for each case. (The heads of the heat exchanger were assumed to be cylinders whose diameter was that of the shell and whose length was 6 inches.) P-9 holdup was calculated, and also the utilization, defined as kilowatts per ton P-9 external to the pile. Pipe sizes used were standard sizes of the smallest wall thicknesses. An optimum pipe size was determined for each flow rate. A complete list of the results of these calculations is given in the following table. The latter part of the table gives the results for 5/8" tubes of shorter length than the maximum recommended. It will be noted that some improvement in utilization resulted from decreasing the tube length. These figures, as mentioned before, are for a single heat exchanger placed in the optimum position over the tank.

Typical Arrangement

Since each of the values in the following table is for the optimum exchanger, so far as location is concerned, it is of interest to know the utilization for a group of exchangers which will dissipate a fair quantity of power. Such a scheme might be visualized as sketched in Figure IV (LZ-A-123) (p. 17) Here five exchangers are placed in a row about 20 feet above the bottom of the tank. For example, let each contain the maximum number of 12 foot long 5/8" tubes. Take a slurry velocity in the tubes of 1.25 ft./sec. and use 18 inch pipe for the lines. The middle exchanger will hold up 3.7 tons, the two on each side 3.8 tons each, and the outer two 4.25 tons each. The total holdup is about 20 tons, the total power about 180,000 KW, giving a utilization of 9000 KW per ton of P-9 external to the pile. Assuming 40 tons inside the pile, the utilization based on the total P-9 is then 3000 KW/ton.

Tube Size In.	No. of Tubes	Tube Length Feet	Velocity in Tubes ft./sec.	Power Output KW	External T. °C.	Pipe Size In.	Velocity in Pipe ft./sec.	Required ht. of Exch. Ft.	External P-9 Holdup Tons	Utilization KW per External Ton P-9
1	1250	20	1.0	42,000	50	16	3.65	31.0	6.8	6200
						18	2.85	21.2	5.9	7140
						20	2.3	16.5	6.1	6930
			1.5	57,000	56	18	4.2	42.1	8.2	6950
						20	3.4	30.4	7.7	7400
						24	2.35	20.5	7.9	7200
						30	1.5	15.5	8.8	6520
			2.0	69,500	62	20	4.4	52	10.6	6600
						24	3.0	32.3	10.0	6950
						30	1.93	23.5	11.5	6050
3/4	2100	16	1.0	38,100	43	14	4.0	34.5	4.6	8220
						16	3.0	22.5	4.4	8730
						18	2.35	17.2	4.4	8660
			1.5	52,300	49	16	4.5	46.8	6.4	8150
						18	3.55	32.6	6.0	8720
						20	2.85	25.4	5.9	8830
						24	1.95	19.4	6.5	8100
			2.0	64,500	55	20	3.80	40.8	8.0	8060
						24	2.60	27.2	7.9	8170
						30	1.68	23.6	10.1	6390
5/8	2650	16	1.0	32,600	36	12	3.82	32.8	3.7	8770
						14	3.10	24.7	3.6	9120
						16	2.33	18.4	3.6	9090
			1.5	45,700	41	14	4.6	49.7	5.1	8960
						16	3.5	33.9	4.8	9430
						18	2.75	26.6	4.9	9370
			2.0	57,000	46	16	4.7	59.6	7.0	8090
						18	3.65	43.3	6.7	8500
						20	2.95	36.2	6.9	8270
3/8	1850	16	1.0	7,500	23	5	4.35	65.6	1.05	7150
						6	3.02	32.5	0.84	8910
						8	1.68	19.3	0.87	8660
			1.5	10,900	26	6	4.5	83.5	1.53	7130
						8	2.55	37.1	1.25	8710
						10	1.58	29.0	1.44	7600
[redacted]		12	1.5	[redacted]		[redacted]	3.55	79.7	[redacted]	[redacted]
						[redacted]	2.25	43.2	[redacted]	[redacted]
						[redacted]	1.55	35.8	[redacted]	[redacted]
[redacted]										
						[redacted]	3.1	24.2	[redacted]	[redacted]
5/8	[redacted]	12	1.0	29,700	44	16	2.33	17.9	3.1	9610
						18	1.82	14.4	3.2	9250
			1.25	35,600	47	14	3.9	36.2	3.9	9120
						16	2.9	24.6	3.6	9760
						18	2.28	19.3	3.7	9690
			1.5	41,000	50	16	3.5	33.6	4.4	9470
						18	2.75	25.8	4.3	9500
						20	2.2	21.3	4.5	9200
		8	0.75	20,000	51	12	2.77	20.2	2.2	9000
						14	2.3	16.2	2.2	9150
			1.0	25,200	55	14	3.1	24.8	2.8	9160
						16	2.33	17.5	2.6	9580
						18	1.82	13.9	2.6	9500
			1.5	33,900	62	18	2.75	25.3	3.9	8710
						20	2.2	20.2	3.9	8760
						24	1.5	16.0	4.4	7800

V - RESULTS OF INVESTIGATION

Discussion

Inspection of Figure II shows that, although 5/8" tube exchangers remove the least heat per exchanger, the exit temperature is lowest, thereby giving the greatest differential to induce flow. In this exchanger also, holdup is less than in those containing larger tubes. The net result is that the utilization is better for the 5/8" tube exchanger, as shown in the table of results. Smaller tubes, 3/8" or 1/4", as shown in table are not attractive; chiefly because of large pressure drops encountered.

Figure III shows that decreasing the tube length causes less than a proportional change in the heat removed and a decrease in holdup; but, on the other hand, shorter tubes deliver the slurry at higher temperatures. These opposing effects cause a maximum utilization to be reached at a tube length of about 12 feet, as shown in the tables.

To increase the power output of the pile over the value operable with the five exchangers shown in Figure IV, a second layer of exchangers may be placed above the first. The height of this row would be about 27 feet from the bottom of the tank. Using 18 inch pipe again, we see from the table that the power output of the middle unit would be near 40,000 KW. Thus the second row would give 200,000 KW more. The additional holdup, if the same proportion may be applied to the second row as was found in the first, would be about 23 tons.

Therefore:	Total power	380,000 KW
	Total holdup	43 tons (external)
	Utilization	8,850 KW per ton

The utilization decreases as more exchangers are added.

Inspection of the exit temperature curves of Figure II shows that the larger the tube size, the higher the exit temperature for the exchangers of maximum size (see Table of Andale size recommendations). The reason for this is that the smaller exchangers take less flow at a given tube velocity and that they have a larger heat transfer area available. The fact that the curve for 1" tubes lies below that for 7/8" tubes is attributed to the longer tube lengths permissible in the 1" and 1 1/4" sizes.

The fact that the power output lines cross for some tube sizes is attributed to a combination of several factors. Take, for example, the 3/4" and the 7/8" tube lines; the tube lengths are equal, and the power output lines cross at a tube velocity of about 1.5 ft./sec. At velocities below this figure, more power is removed from the 7/8" tube exchanger, and the opposite is true above this velocity. Flow is greater through the larger, and heat transfer area is greater in the smaller. The combination of these factors with the mean temperature difference

from slurry to cooling water results in an intersection in the output curves. At low velocities, the greater flow and higher temperature differences appear to over-compensate for less area, while at higher velocities, the reverse is true.

Conclusions

The maximum utilization of P-9, based on the external holdup, occurs for the case of a heat exchanger containing 5/8" tubes, each 12 feet long, with two cooling water streams.

Power outputs as high as about 180,000 KW may be handled with about 20 tons of P-9 external to the pile. This utilization (9000 KW/ton) may be increased to approximately 9700 KW/ton if the power output is decreased to 35,000 KW, and is decreased to 8800 KW/ton if the power output is raised to about 380,000 KW for the arrangements described. Small pressure differentials of about 45 lbs./sq.ft. are sufficient to circulate the slurry for this arrangement.

Such a system cools the slurry from 120° C. to about 60° C. The cooling water enters at 15° C. The average temperature of the exit water is 45° C. An increase in power output of only about 7% can be obtained by doubling the cooling water rate. This would change the cold slurry temperature by only a few degrees and would improve utilization correspondingly. This increase in cooling water rate, therefore, does not appear to be a practical way of obtaining substantial increases in utilization.

The fact that considerable power can be removed in this way offers a possible safeguard in the pump-circulated plant in that thermal syphoning might well remove the power generated after shutdown in the event of pump failure.

APPENDIX

Calculation of $\bar{\rho}$

$\bar{\rho}$, the mean effective density in the pile tank was calculated after making the following assumptions:-

1.) Rate of heat production varies sinusoidally along axis of tank.

2.) Density-temperature relationship will eliminate radial temperature gradients.

3.) The cross section of the tank is constant.

4.) The slurry leaves the pile at 120° C.

If x is the height measured from the bottom of the tank, and h is the total depth, then the temperature rise dt in a unit volume of slurry in passing through dx will be proportional to the heat generated in the element dx during the time of passage.

$dt = B \sin \frac{\pi x}{h} dx$, where B is a constant embracing the velocity, the density, and the specific heat of the slurry.

Integrating from t = 120 and x = h, to t and x gives

$$120 - t = B \frac{h}{\pi} \left(1 + \cos \frac{\pi x}{h}\right)$$

If the slurry enters the pile at $t = t_2$, (at $x = 0$), then

$$B \frac{h}{\pi} = \frac{120 - t_2}{2}, \text{ and}$$

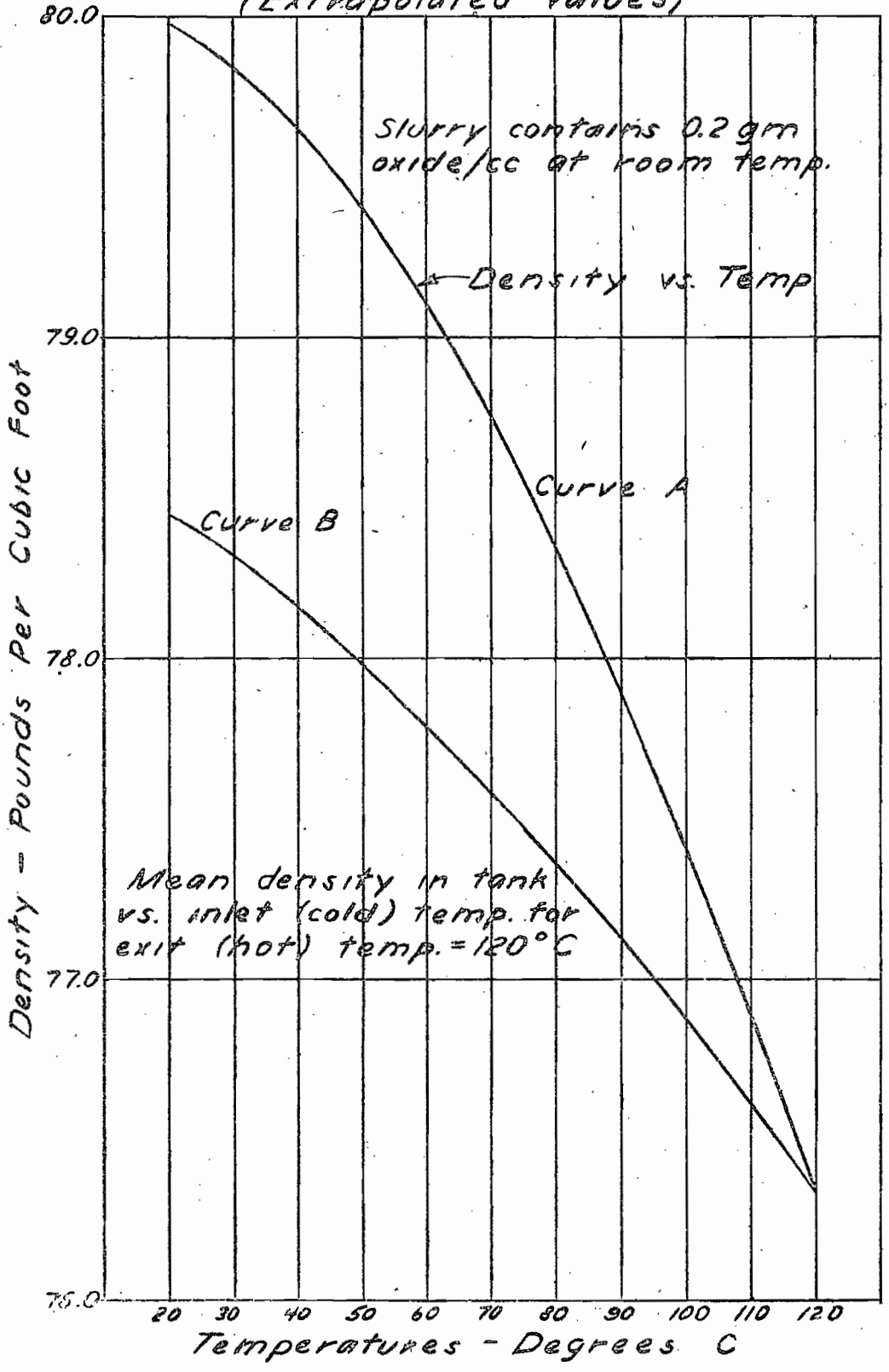
$$t = 120 - \frac{120 - t_2}{2} \left(1 + \cos \frac{\pi x}{h}\right)$$

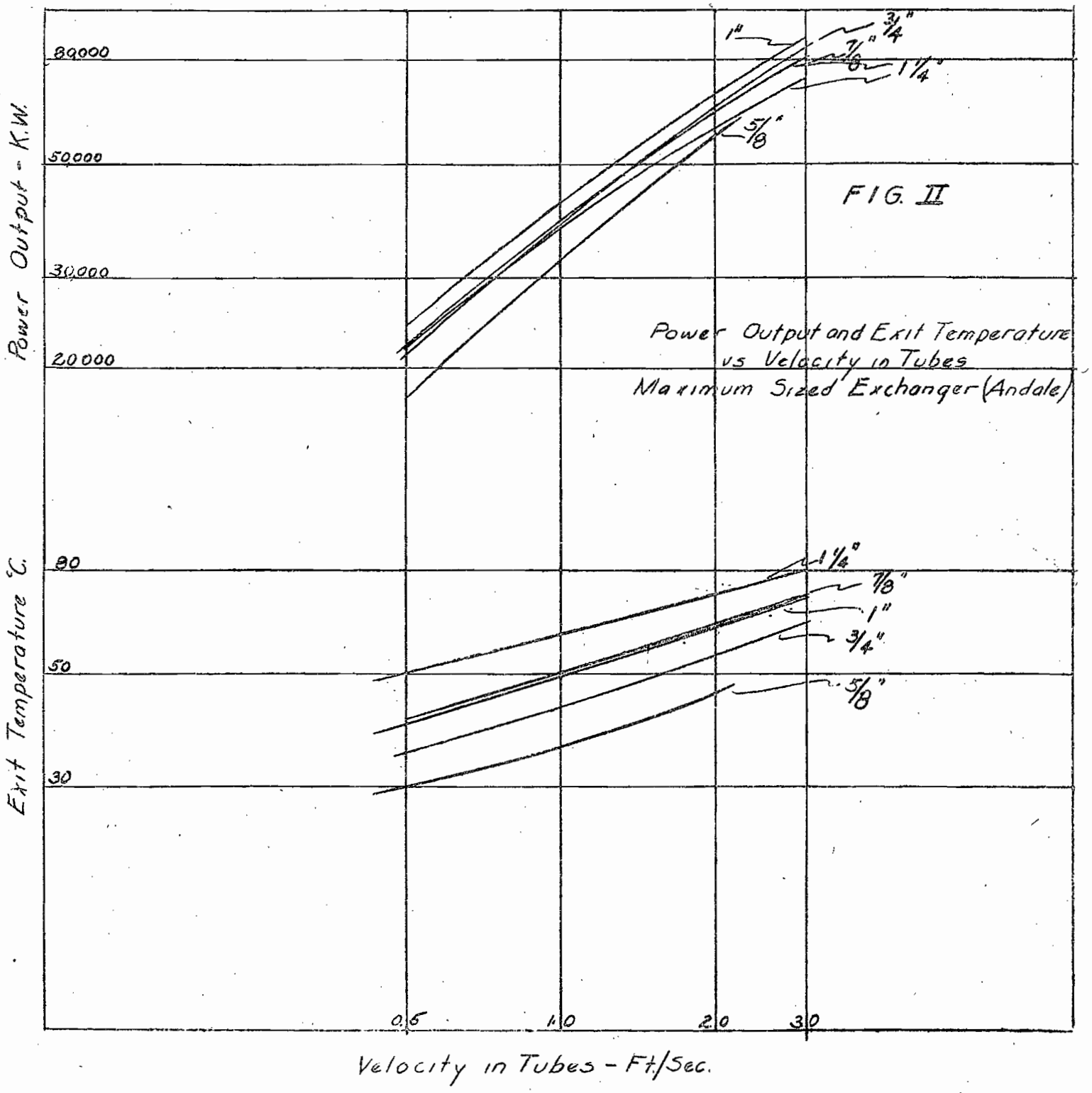
Using this equation, t vs. $\frac{x}{h}$ plots were made and from these ρ vs. $\frac{x}{h}$ plots for several values of t_2. For each t_2, graphical integration gave

$$\bar{\rho} = \frac{\int_0^h \rho \, dx}{h}$$

ρ was then plotted against t_2. This is shown as Curve B of Figure I.

Figure 1
Slurry Density vs. Temperature
(Extrapolated Values)




Power Output - K.W.
80000
50000
30000
20000
1"
3/4"
7/8"
1 1/4"
5/8"
FIG. II
Power Output and Exit Temperature
vs Velocity in Tubes
Maximum Sized Exchanger (Andale)
Exit Temperature °C.
80
50
30
1 1/4"
7/8"
1"
3/4"
5/8"
0.5
1.0
2.0
3.0
Velocity in Tubes - Ft./Sec.

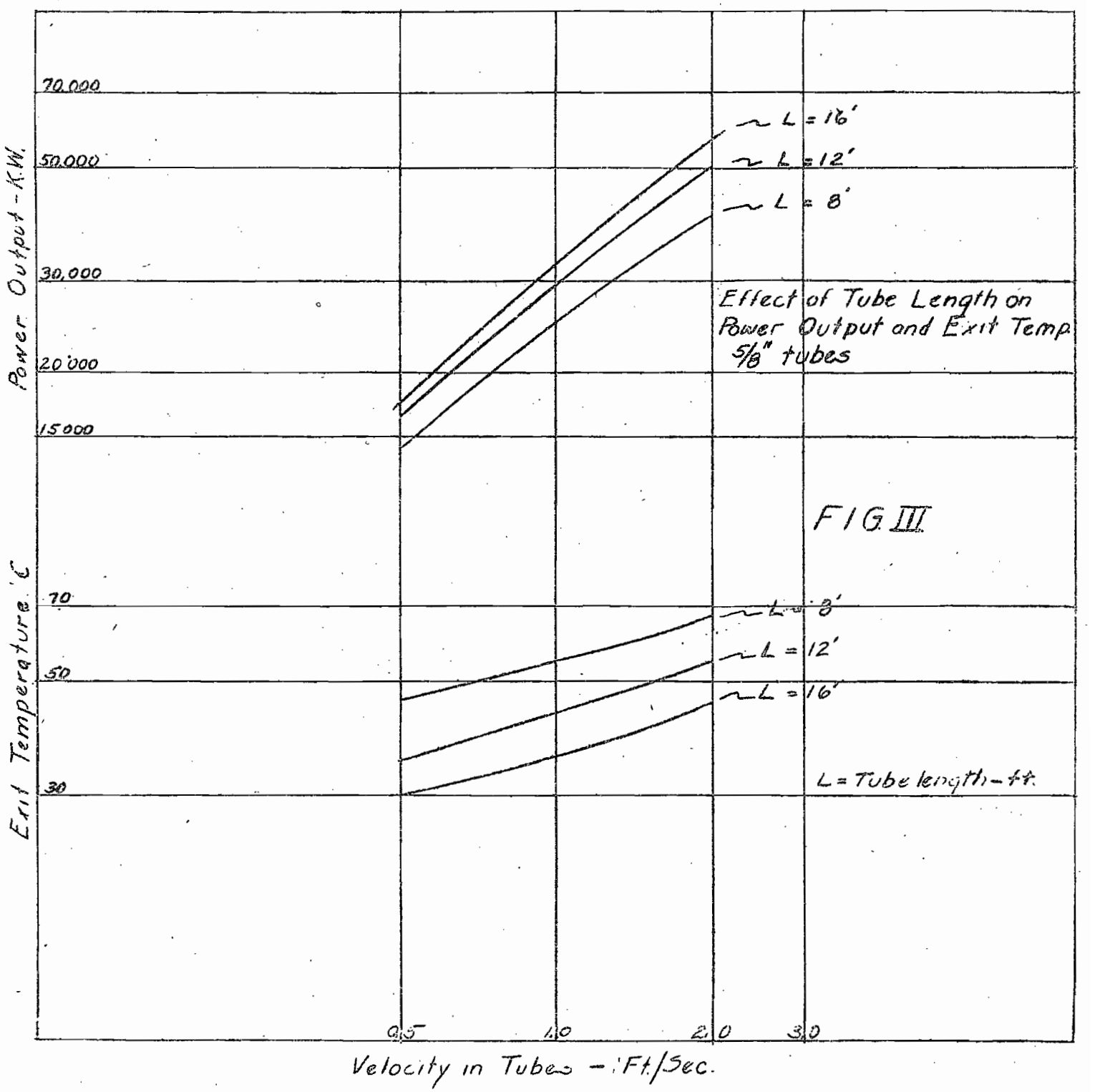

70,000
50,000
30,000
20,000
15000
Power Output - K.W.
~ L = 16'
~ L = 12'
~ L = 8'
Effect of Tube Length on
Power Output and Exit Temp
5/8" tubes
FIG III
70
50
30
Exit Temperature °C
~ L = 8'
~ L = 12'
~ L = 16'
L = Tube length - ft.
0.5
1.0
2.0
3.0
Velocity in Tubes - Ft./Sec.

Figure 4

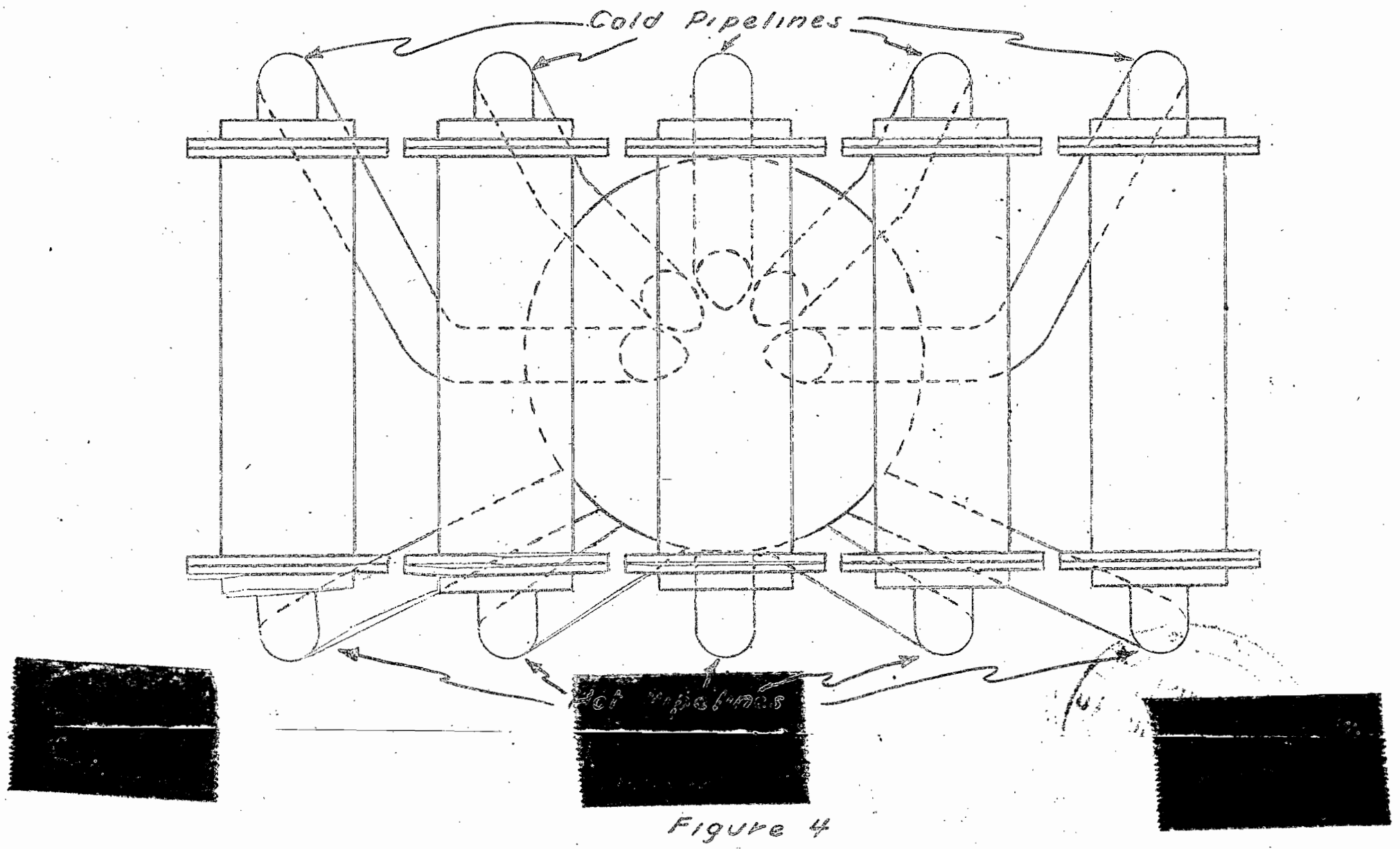
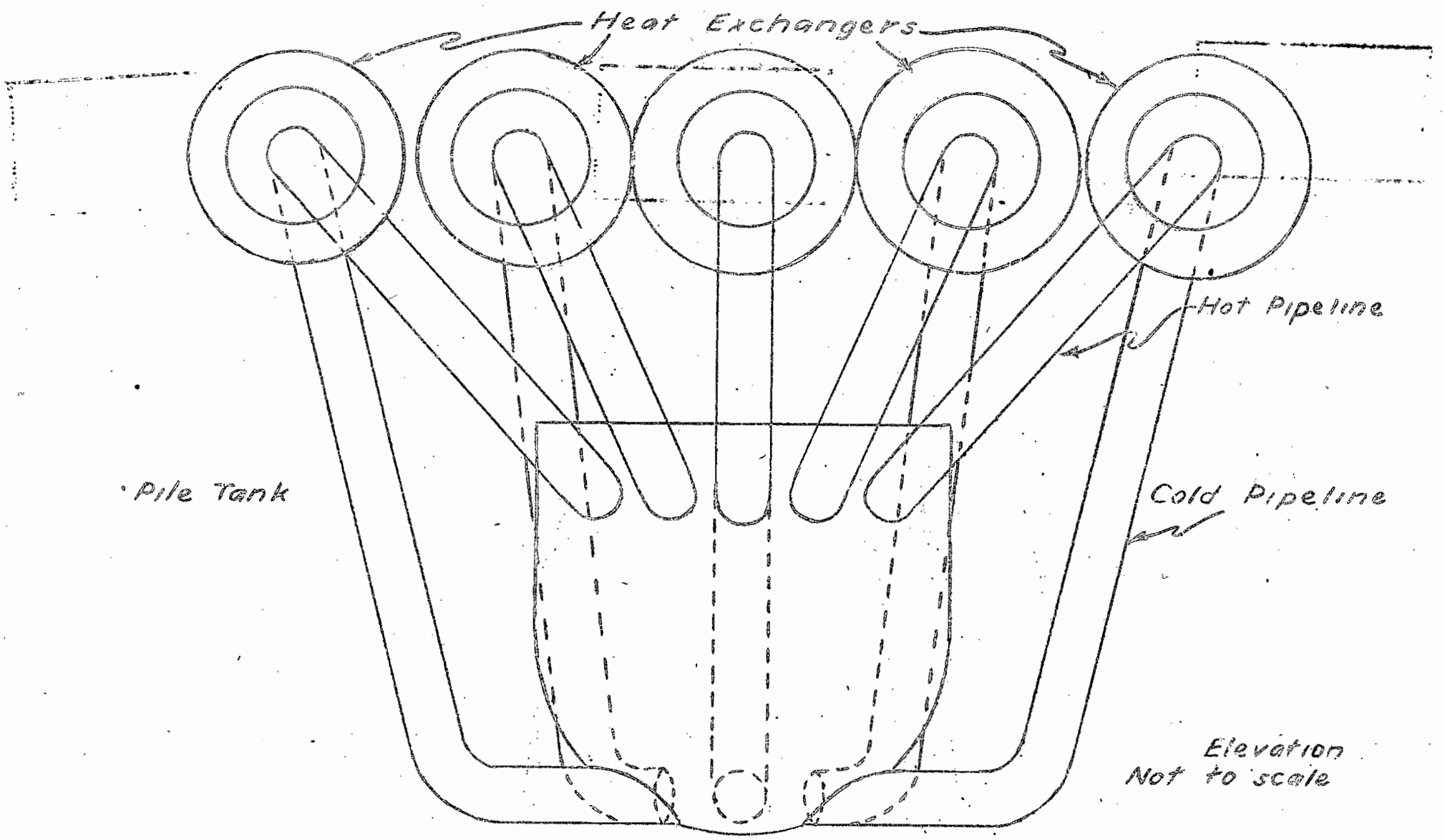


Figure 4



Sketch of Typical Arrangment For Thermal Sylphon

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