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TEMPERATURES ON THE SURFACE OF A SLUG JACKET

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November 1, 1944

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SUMMARY

At Hanford, it is proposed to operate a chain-reacting, graphite-moderated, water-cooled uranium pile. On account of the corrosive action of water on uranium, it was necessary to enclose the uranium cylinders in aluminum jackets. Preliminary data were available which indicated that aluminum jacket corrosion might be influenced by the water temperature. Therefore, in order to specify safe operating conditions, it was desirable to know the temperature at any position on the aluminum jacket of a slug in the tube. Because of the experimental difficulties encountered in the measurement of the correct surface temperature, the problem has been attacked primarily from a mathematical standpoint by members of the Physics Division, although some experimental data were also obtained through the combined efforts of the Physics and Technical Divisions. Mathematical formulae have been developed for most of the proposed Hanford designs. The purpose of this report was to evaluate and summarize the theoretical and experimental information for calculation of aluminum jacket surface temperatures for the design now installed at Hanford. Also, the summarized results were to be put into a form suitable for use in routine calculations.

As a result of this survey, the aluminum surface temperature for the Hanford tube and slug design may be calculated within 15°C. by employing routine methods and certain simple factors and equations contained in this report. The factors have been assembled from mathematical analyses but agree closely with the existing experimental data. The equations are of the usual type used in heat transfer calculations.

Until adequate experimental data are available, it is recommended that the factors and formulae presented in this report be employed in the calculation of the jacket surface temperatures. Experimental work is needed to substantiate the formulae and factors obtained. Also, additional work is needed to determine the surface temperatures necessary to cause early failure of the slug jacket by corrosion, erosion, or cavitation.

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GENERAL DISCUSSION

When it was decided to employ a water-cooled pile, the problem existed of protecting the uranium slugs from water corrosion. Of all the coating materials and methods tested, aluminum jackets on the slugs proved to be least unfavorable. One of the problems, then, was to investigate the effect of temperature on the corrosion rate of the aluminum. Also another condition that might be troublesome during pile operation was the possibility that, if surface temperatures became sufficiently high to allow local boiling, the quantity of water in the tube could be reduced to one-half that under normal conditions. This condition would result in unstable operation of the pile because the change in reproduction factor caused by the reduced mass of water in one tube is appreciable; it corresponds to the change obtained with the control rods.

The two methods available for the determination of the temperature on the surface of the jacket were mathematical analysis and experiment. The experimental difficulties of measuring the surface temperature without disturbing the high heat flux and the water flow pattern have not been completely mastered. The existing methods to determine the surface temperatures are indirect and result in data which are inconclusive. The mathematical treatments represent the application of some of the best analytical methods available. In the reports of the Physics Division are presented exact mathematical formulae for most of the proposed Hanford designs.

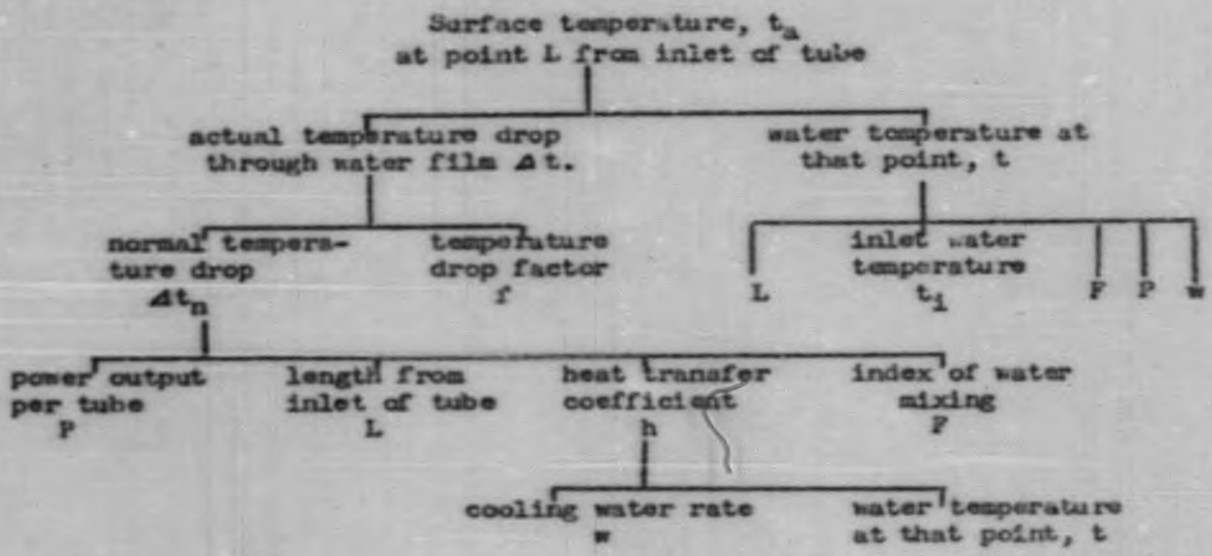
The purpose of this report was to reduce all the information pertinent to the calculation of the jacket temperatures to a form suitable for routine utilization. By restricting this report to consideration of only the existing Hanford tube and slug design, it was possible to simplify greatly the methods of calculation.

Although this report is limited to the Hanford design, the procedures outlined can be used for various inlet water temperatures, water rates, and power output. No consideration was given to the effects produced by such conditions as film formation, cavitation, or for local changes in water velocity.

INVESTIGATION

In order to determine the surface temperature, both mathematical analyses and experimental investigations were made. Since few experimental data were available, emphasis was placed on the mathematical approach.

The variables affecting the surface temperature mathematically were isolated. The general relationships between the variables are shown on the following diagram. The items directly under each factor influence this factor directly.



Of the seven variables listed in the diagram, the operating conditions fix three; inlet water temperature, t_i; cooling water rate, w; and power output of the tube, P. Designation of the location of the point to be analyzed fixes the fourth variable, L, the distance from the inlet of the tube.

Three variables are left to be calculated or estimated. The factor, f, applied to the normal temperature drop to give the actual temperature drop, is the term obtained from the work done by the Physics Department on the effect of a certain condition (such as the presence of a rib) on the actual temperature drop. If two conditions occur at the same point (example: Coating defect and the presence of a rib) the product of the factors describing each condition is the new factor. Comparison with experimental data (Exhibit G) (Exhibit H), indicates that the factors are sufficiently accurate. The factors are listed in Tables II and III. These factors, with two exceptions, are dependent only on the geometry of the slugs and the tube. For minor changes in dimensions, the original Physics Division reports should be consulted. Any major changes in the geometry would require extensive mathematical work to provide new factors.

The heat transfer coefficient, h, is calculated in this report by use of the correlation of Sieder and Tate (13). Comparison with the experimental data (Exhibit J) shows a maximum difference of 25%. The data are not of sufficiently good quality, however, to justify the use of another correlation. The maximum error in the calculation of the slug surface temperature can be expected to originate in the evaluation of the heat transfer coefficient, h.

The use of F, the index of the degree of mixing, comes from consideration of water velocities not in the direction of flow which show that the water in the annulus must be only partially mixed. An examination of Figure 3 shows that one-half of the slug is contained in the lower half of the view (corresponding to one-half of the heat generation). The presence of the ribs, however, makes the available area for water flow less than one-half of the annular area. With the total water flow distributed approximately evenly over the available annular area, this means that one-half of the heat load must be handled by less than one-half of the water. The symbol F represents the ratio of the heat fraction to the water fraction. For perfect mixing, the heat fraction is unity; and, therefore, F equals one. Except for perfect mixing, F on the lower side of the tube is greater than one. For the upper part of the tube, F is less than one.

Three degress of mixing have been investigated as indicated in Exhibit C. Of these the first case seems to be the most likely in view of the experimental date (12). Case I (Exhibit C) assumes perfect mixing of θ and $\phi/2$. (See Figure 3). This means a maximum mixing distance 0.566 inches or 1.44 cm. If there is perfect mixing between θ and $\phi/2$, then there will be perfect mixing in the entire lower half of the tube, or α region. The upper half of the tube, or β region is all at the same temperature because there are no discontinuities in the water stream. No mixing is assumed between the α and β streams. The α heat fraction is the same as the β fraction and equals one-half. The β water fraction is 0.525; the α water fraction is 0.475. Therefore, F_α equals 1.05; F_β equals 0.954.

In the immediate vicinity of the ribs the water velocity may be lower. This would mean a lowered heat transfer coefficient and, therefore, a higher surface temperature at that region. That the temperature near the rib is higher has been determined experimentally by Kratz, but there is no experimental evidence to indicate whether the condition is due to lower velocity, poor mixing, or both.

An estimation was made of a possible degree of velocity distribution due to the difference in the upper and lower paths for mixing Case I, with α and β temperature distribution. (See Figure 3). At the inlet, when the temperatures of the two streams are equal, their velocities are also equal. (The hydraulic radius of each path is equal. The extra wetted-periphery due to the ribs is compensated for by the slight eccentricity upward due to the height of the ribs). As the lower stream becomes hotter, it tends to have greater veolcity. At 19 ft. from the entrance, the increase of α velocity over β belocity is 1.02. Therefore, some water from the cold stream must flow into the hotter section. This has two desirable effects: (a) to lower the temperature of the hot stream (b) to promote mixing.

The heat flux per unit area of the slug surface is assumed not to be appreciably changed by the change in surface temperatures around the slug. The major resistance to heat flow is in the tuballoy and the maximum change in heat flux from the slug due to changes in surface temperature will be less than 5%. Neglect of this factor will give a conservative temperature difference.

No attempt has been made to account for solid film accumulating on the surface of the aluminum. This situation has been thoroughly studied at CMX, and at the present time it is believed that conditions can be maintained such that iron, chromium and aluminum films of thickness sufficient to interfere with heat transfer and pressure drop may be prevented.

The experimental method used to determine jacket surface temperatures (Appendix G, Cabell's date) was to cut a slot in the surface of the aluminum. A thermocouple was imbedded in the metal and the leads were taken out along the slot which was then filled with type metal to present a smooth surface to the flowing water. The temperature read on the thermocouple was then corrected for the thickness of metal between the tip of the thermocouple and the outside surface of the coating. The temperature of the water at that point could not be measured because the introduction of any temperature measuring device could completely change the flow patterns in the thin annulus. The mathematical evaluation necessary in order to interpret the experimental data involves nearly as many assumptions as a mathematical evaluation of the entire problem. In view of the corrections applied to the data, the results are subject to large errors. The use, then, of the available data to furnish reliable information on slug surface temperatures, even at normal operation, is open to question and to doubt.

RESULTS

This survey shows that the aluminum jacket surface temperature for the Hanford tube and slug design may be calculated, by routine methods, by use of certain simple factors and equations. The factors have been compiled from the extensive and careful mathematical analyses made by the Physics Division for each case. Limiting the factors to only those which apply to regular Hanford slugs and tubes allows a great simplification in their presentation. Each factor describes a certain condition. If two conditions occur at the same point (example: coating defect over a rib) the product of the factors describing each condition is the new factor. This method of treatment has been suggested by Karush and Young. The factor for the temperature rise above the rib is the only factor that has been checked by experimental data. The experimental data check the calculated values very closely. (See Exhibits G and H).

When operation is very steady and there is little variation in power, water rate, and inlet water temperature, a plot such as Figure 4 can be made. Knowing the product of the factors that apply to the jacket surface or slug surface in question, the metal temperature at any length can be read directly from the chart. However, it should be emphasized that this type of graph is good only at a given power level, water rate, inlet water temperature, and degree of mixing.

The factors themselves, with two exceptions, are dependent only on the geometry of the slugs and the tube. Figures 1, 2 and 3 clearly define the geometry for which the factors apply. The factors are independent of power, water rate, temperature, or degree of mixing.

In order to calculate the slug surface temperature, it is necessary to know:

L - the distance of the point from the inlet of the active portion of the tube, ft.

t_i - the inlet temperature of the cooling water, °C.

v - the cooling water rate, gal./(min.)(tube).

(The following equations are developed using gal./(min)(tube); however, in the results, Table III, and in the derivations, Exhibits D and E, give the forms for velocity in ft./sec. per tube and the mass flow rate, lb./(hr.)(tube).)

P - the power output of the tube, kilowatts.

F - a degree of mixing for the cooling water. An optimistic assumption is Case I, using ∞ mixing; other values are analyzed and given in Exhibit C.

f - temperature drop factors, given in Table II.

The following equations are derived in the Appendix (Exhibits D, E, and F) from fundamental equations.

To calculate the temperature of the local cooling water:

$$t = t_i + \frac{1.91\ (F)\ (P)\ \left(1 - \cos \frac{\pi L}{23.3}\right)}{v} \qquad (1)$$

Using Figure 5, a plot of $\left(1 - \cos \frac{\pi L}{23.3}\right)$ versus L, and the specified con-conditions, the temperature (t) of the cooling water at that point can be readily calculated.

Since heat transfer equations are of the form $q = hA\,\Delta t$, it is necessary to find the heat transfer coefficient at this point. The equation of Sieder and Tate (13), applied to the Hanford slugs and tube design reduces to

$$h = 721\ (v)^{0.8}\ (0.367 + 0.00639\ t) \qquad (2)$$

Figure 6 gives a plot of the last term of the equation: (0.367 + 0.00639 t) versus t. The rate, in gallons per minute, is already specified. The heat transfer coefficient is expressed as B.t.u./(hr.)(sq.ft.)(°F.).

For this particular system, the equation of $q = hA\,\Delta t$, when solved for Δt, becomes:

The sine curve describing the heat generation along the length of a tube is shown in Figure 10.

DISCUSSION OF RESULTS

The correct surface temperature is obtained as the sum of the water temperature plus the temperature difference between the surface and the water. The water temperature is a function of inlet water temperature, water rate, the heat input up to point in question, and the mixing of the water around the annulus (Equation 1). The temperature difference is a function of the rate and physical properties of the water passing the surface, heat flux at that point, the design of the end cap (Figures 1 and 2), imperfections in the bond, eccentricity of slug, and rib contact.

Karush (5) showed that the rib may cause an increase in surface-to-water temperature difference. For the size of rib shown in Figure 3, the maximum factor of 1.56 is for the case of no heat transfer in the area above the rib. The minimum factor, 1.0 to 1.05, is that obtained by assuming a good contact of aluminum on aluminum. Kratz (8) has found experimentally that this condition may occur. Comparison with experimental data (Kratz, Cabell) shows a factor which corresponds to a water film of about 0.001 inch in thickness.

Murray (10) has shown that displacing the slug from the center of the tube will increase the slug surface temperature. The value given in Table II is due to eccentricity because the ribs do not place the slugs in the center of the tube. Tests by Kratz indicate that this correction may be too high (14).

A number of investigators (4),(11) have calculated the increase in temperature of the surface adjacent to an imperfection in the bonding layer. The factor given ($f = 1.2$) corresponds to the maximum permissible diameter of imperfection as determined by the frost test. Other values are given in Figure 11.

The temperatures around the end of the slug have been studied by a number of investigators (3), (6), (7), (9). The factors given are for the insulated end cap as used on the unbonded slug; and for the solid end cap as used on the aluminum-silicon bonded slug. Experimental check by the electrical heat-sink and resistance methods has been obtained for the solid end cap by a collaborating agency.

In addition to the particular conditions due to the rib, coating defect, and effect, Wilkins' effect (hot ring near end of slug), eccentricity, and mixing effects, there are possible effects resulting from non-uniformity of the slug diameter, from curved surfaces at the end of the slug coating, and from slug warping.

$$\Delta t_n = \frac{339\ (F)\ (P)\ \sin \frac{\pi L}{23.3}}{h} \qquad (3)$$

Figure 7 gives a plot of $\sin \frac{\pi L}{23.3}$ versus L. The heat transfer coefficient, h, can be calculated from equation (2), and the normal temperature difference (Δt_n) obtained from equation (3).

The final step in the calculation of the actual surface jacket temperature is to apply the factors to the normal temperature drop. The actual temperature drop is the product of the normal temperature drop and all the factors pertinent to the description of the point.

$$t_s = t + (f_1)(f_2)(f_3)(\ldots.)\ \Delta t_n \qquad (4)$$

The surface temperature t_s is fully defined. It is at a certain length, L, from the inlet, and at a certain position on the jacket.

Table I shows the effect of varying any of the conditions of the system. As a basis of comparison, Condition 1 was chosen as representing a set of possible operating conditions. Condition 2 shows the rise in surface temperature due to an increase in the factor. Condition 3 shows the corresponding values if the power input is reduced one-half. The normal temperature difference does not decrease a proportional amount because the heat transfer coefficient is lower. Condition 4 shows the effect of reducing the cooling water rate. The water temperature is higher, and the heat transfer coefficient is lower on account of the lower velocity through the tube. The result is a higher temperature difference which when added to the high water temperature gives a much higher metal temperature. An increase in the inlet water temperature, Condition 5, does not give an equal rise in the metal temperature because of the improved heat transfer coefficients. Condition 6 shows the effect of a different estimate for the amount of mixing. (Condition 1 was for ∞ mixing; Condition 6 is for perfect mixing.) The effect of a more favorable, lower, F (heat fraction to water fraction) is a lowering of the water temperature and of the normal temperature difference, and, therefore, a lowering of the jacket temperature.

Figure 8 shows the modified cosine curves of heat transfer coefficient, h, versus L for the conditions of 346 KW/tube, and a water rate of 22 gal./min. The family of curves was obtained by using various assumptions for the degree of mixing. At fixed conditions of flow and heat generation, h decreases as F decreases.

Figure 9 shows the modified sine curve of normal Δt versus length, L. The peak of the curve comes before the center of the tube. The highest metal temperature, however, (see Figure 4) does not come until after the center of the tube, on account of the rapid rise of the water temperature at the center of the tube. The higher the factor, the nearer the maximum metal temperature will be to the center of the tube. As the factor approaches low values, the maximum temperature is nearer the exit of the tube.

The variation in slug diameter, which would occur chiefly at the welded ends, may result in slightly higher temperatures in cases where small slugs are leeward of large slugs. The end temperature of the small slug may be increased slightly; however, maximum temperatures are further along the slug surface, hence this is of minor consideration.

The maximum temperature effect expected by warping of the slug could be predicted by the eccentricity effect as described by Murray (10).

Curved surfaces at the end of the slug, warping, and variation in diameter of the slug, if carried to extreme, may have an effect on the water pressure at the slug in addition to the small temperature effects. Such curved surfaces may result in cavitation. That is, the water velocity or vortexes may be such that a partial reduction in static pressure on the surface of the slug may result, and vaporization occur over very small areas. This caviatation would result in very rapid removal of aluminum metal near this area. This sort of hazard has not been analyzed mathematically nor determined experimentally. The chance that cavitation will occur becomes more important as the surface temperatures approach the vaporizing temperature of the cooling water. No attempt has been made to determine experimentally the conditions necessary to obtain cavitation; and there has been no evidence of this condition existing under proposed operation conditions.

CONCLUSIONS

The factors for estimating the maximum temperature difference between the water and metal surface are believed to include all considerations accurately, but the temperature difference as calculated from the heat transfer coefficient will result in a maximum error of 15°C. in estimating surface temperatures. There is no experimental evidence that the temperature need be known with greater degree of accuracy for the present proposed maximum temperature rise of the water and the maximum flow rates. It is likely that the maximum surface temperature can be increased 30 to 40°C. (up to 135°C.) before this factor of uncertainty may become of importance, or that surface temperatures become important.

NOMENCLATURE

A - Area, sq.ft.
D - Hydraulic diameter of water passage, ft.
F - Ratio of heat fraction to water fraction
G - Mass velocity, lb./(hr.)(sq.ft.)
L - Length of tube, ft.
L' - Distance along tube from last known point, ft.
P - Total power output of tube, kilowatts
a, a', c - Constants
c_p - Specific heat of water, B.t.u./(lb.)(°F.)
f - Temperature drop factor
h - Heat transfer coefficient, B.t.u./(hr.)(sq.ft.)(°F.)
k - Thermal conductivity of water, B.t.u./(hr.)(sq.ft.)(°F./ft.)
q - Rate of heat transfer, B.t.u./(hr.)(ft. of length)
r - Radius of slug, in.
t - Temperature, °C.
u - Velocity, ft./sec.
v - Volume rate of flow, gal./min.
w - Weight rate of flow, lb./hr.
θ, θ', ϕ - angles, radians
α - Refers to a certain amount of mixing in lower half of tube. (See Exhibit C)
β - Refers to upper half of tube (See Exhibit C)
Δt - Coating-to-water temperature difference, °C.
Δt_a - Temperature drop across aluminum jacket, °C.
$\Delta' t$ - Temperature difference along length of tube, °C.
μ - Viscosity of water, lb./(hr.)(ft.)
μ' - Viscosity of water, cp.
Heat fraction - amount of heat in given sector divided by total heat generated.
Water fraction - amount of water in given sector divided by total water flow.

Subscripts
a - surface of aluminum jacket
i - inlet
m - surface of uranium
n - normal
r - within slug
s - steam
w - water
θ, θ', ϕ, α, β - regions to which term refers (See Figure 3.)

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APPENDIX

TABLE I

Variation in Jacket Surface Temperature with a variation in operating conditions or mixing

	Distance from inlet end of active portion of tube.	Inlet water temp.	Cooling water rate	Total power generation of tube	Degree of mixing	Water temp. Eq.(1)	Heat transfer coefficient Eq.(2)	Normal jacket-to water temp. drop. Eq.(3)	Temp. drop factor	Temp. of jacket surface
Condition	L	t_1	v	(P)	F	t	h	Δt_n	f	t_s
	(ft.)	(°C.)	gal./min.	kilowatts		(°C.)	B.t.u./(hr.)(sq.ft.)(°F.)	(°C.)		(°C.)
1	12	5	22	346	1.05	38.2	5240	23.6	1	62
2	12	5	22	346	1.05	38.2	5240	23.6	2	86
3	12	5	22	173	1.05	21.6	4340	14.2	1	36
4	12	5	11	346	1.05	71.4	4040	30.5	1	102
5	12	10	22	346	1.05	43.2	5510	22.2	1	65
6	12	5	22	346	1.00	36.6	5170	22.7	1	59

TABLE II

Temperature-drop Factors to be Applied to the Normal Jacket-to-water Temperature Drop for Various Conditions

Location of Jacket Surface and Assumed Condition A. Factors for temperature, t_m, at surface of aluminum jacket	Factor	Reference
Above rib. No heat flow to rib.	1.56	(5)
Above rib. Insulating film of water 0.001 inch in thickness between rib and jacket	1.0-1.22	(5)
Above rib. Good Al-Al contact between slug and rib.	1.0-1.05	(5)
Coating Defect. For a coating defect between the jacket and the slug. No heat flow across the defect. See Figure 11 for all values. Value here is for defect diameter of 0.4 in.	1.2	(4), (11)
Top. Slug is displaced upward due to ribs; therefore, top of tube is hotter than normal.	1.07	(10)
Insulated End Cap. At point on surface at end of slug and beginning of cap. (Position 3, Figure 1.)	$0.56\left(1+e^{-0.9\sqrt{\frac{h}{2160}}}\right)$	(3),(6),(9)
Insulated End Cap. At point on surface at end of cap. (Position 4, Figure 1).	$1.12e^{-0.45\sqrt{\frac{h}{2160}}}$	(3),(6),(9)
Insulated End Cap. Hot ring near end of slug. (Wilkins effect. Position 5, Figure 1).	1.12	(9)
Solid End Cap. At center of end of slug at uranium-aluminum interface. (Position 1, Figure 2).	2.63	(7), (9)
Solid End Cap. At center of end of end cap. (Position 2, Figure 2).	1.75	(7), (9)
Solid End Cap. At point on surface at end of slug and beginning of cap. (Position 3, Figure 2).	0.73	(7), (9)
Solid End Cap. At outside of end of cap. (Position 4, Figure 2).	0.48	(7), (9)
Solid End Cap. Hot ring near end of slug. (Wilkins effect. Position 5, Figure 2).	1.05	(9)

TABLE II (cont'd.)

Temperature-drop Factors to be Applied to the Normal Jacket-to-water Temperature Drop for Various Conditions

B. Factors for obtaining temperature, t_m, at surface of slug (under the jacket)	Factor	Reference
Coating Defect. For a coating defect between the jacket and the slug. No heat flow across the defect.		
Diameter of defect = 0.16 in.	2.04	(4),(11)
0.32	4.08	
0.39	5.10	
0.47	6.12	
Solid End Cap. At center of slug at uranium-aluminum interface (Position 1, Figure 2)	2.63	(7),(9)
Insulated End Cap. Temperatures are the same as if the end cap were not there. Use formula for calculation of temperatures within the slug.		

TABLE III

Equations for Calculating the Slug or Aluminum Jacket Temperatures.
See Nomenclature, p. 13.

A. Calculation of the Aluminum Jacket Surface Temperature, t_a, for use on Hanford Slug and Jacket Design. (Fig. 1-3).

$$t = t_i + \frac{1.91\ (F)\ (P)\ (1 - \cos \frac{\pi L}{23.3})}{v} \quad (1)$$

or

$$t = t_i + \frac{1.82\ (F)\ (P)\ (1 - \cos \frac{\pi L}{23.3})}{u} \quad (5)$$

or or

$$t = t_i + \frac{950\ (F)\ (P)\ (1 - \cos \frac{\pi L}{23.3})}{w} \quad (6)$$

$$h = 721\ (v)^{0.8}(0.367 + 0.00639\ t) \quad (2)$$

or

$$h = 750\ (u)^{0.8}(0.367 + 0.00639\ t) \quad (7)$$

or

$$h = 5.05\ (w)^{0.8}(0.367 + 0.00639\ t) \quad (8)$$

$$\Delta t_n = \frac{339\ (F)(P)\ \sin \frac{\pi L}{23.3}}{h} \quad (3)$$

$$t_a = t + (f_1)(f_2)(\ldots)(\Delta t_n) \quad (4)$$

B. Calculation of Temperatures within the Slug, t_r

$$0.0546\ (t_r - t_m) + 0.0000245(t_r^2 - t_m^2) = 0.0364\ (P)(\sin \frac{\pi L}{23.3}) - 43.6\ r^2$$

C. Relation of Slug Surface Temperature, t_m to Aluminum Jacket Surface temperatures, t_a.

$$t_m = t_a + \Delta t_a$$

At a power output of 346 kw per tube and no coating defect between jacket and slug:

$$\Delta t_a = 4 \sin \frac{\pi L}{23.3}$$

This means that t_m is, as a maximum, 4°C. higher than t_a if there is no coating defect at that point.

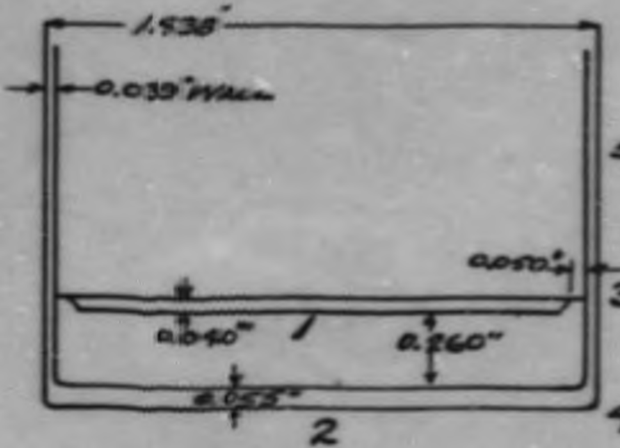


FIGURE 1: UNBONDED SLUG, INSULATED END CAP DESIGN.

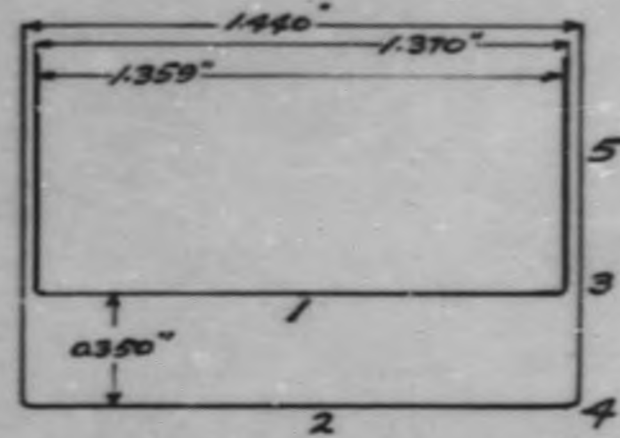


FIGURE 2: AL-SI BONDED SLUG, SOLID END CAP DESIGN.

$\theta = \theta' = 0.258$ RADIANS

$\theta + \frac{\phi}{2} = \frac{\pi}{4}$ RADIANS

FIGURE 3: CROSS SECTION OF HANFORD TUBE AND SLUG

FIGURE-4

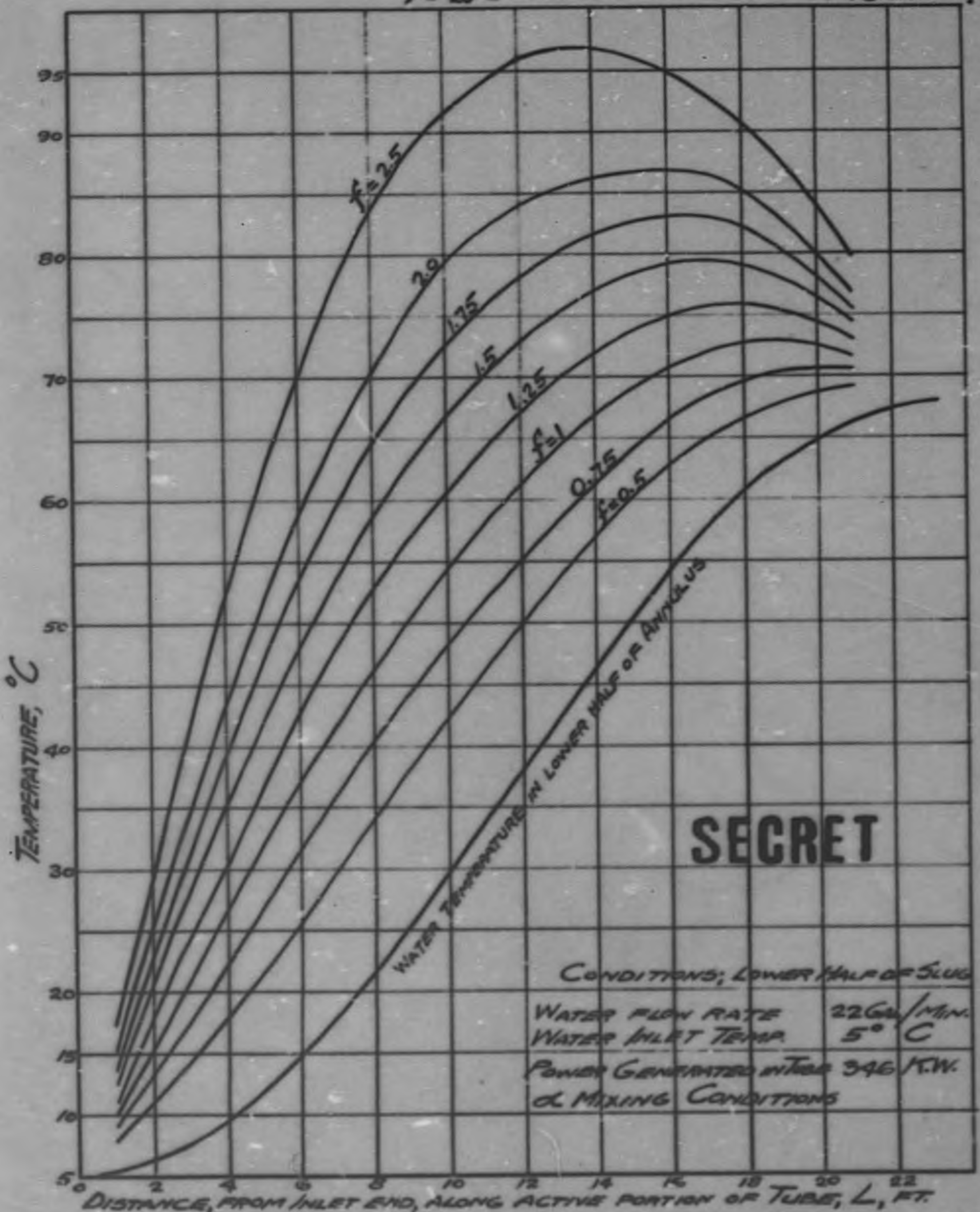


FIGURE 4: JACKET SURFACE TEMPERATURES FOR VARIOUS FACTORS

FIGURE-5

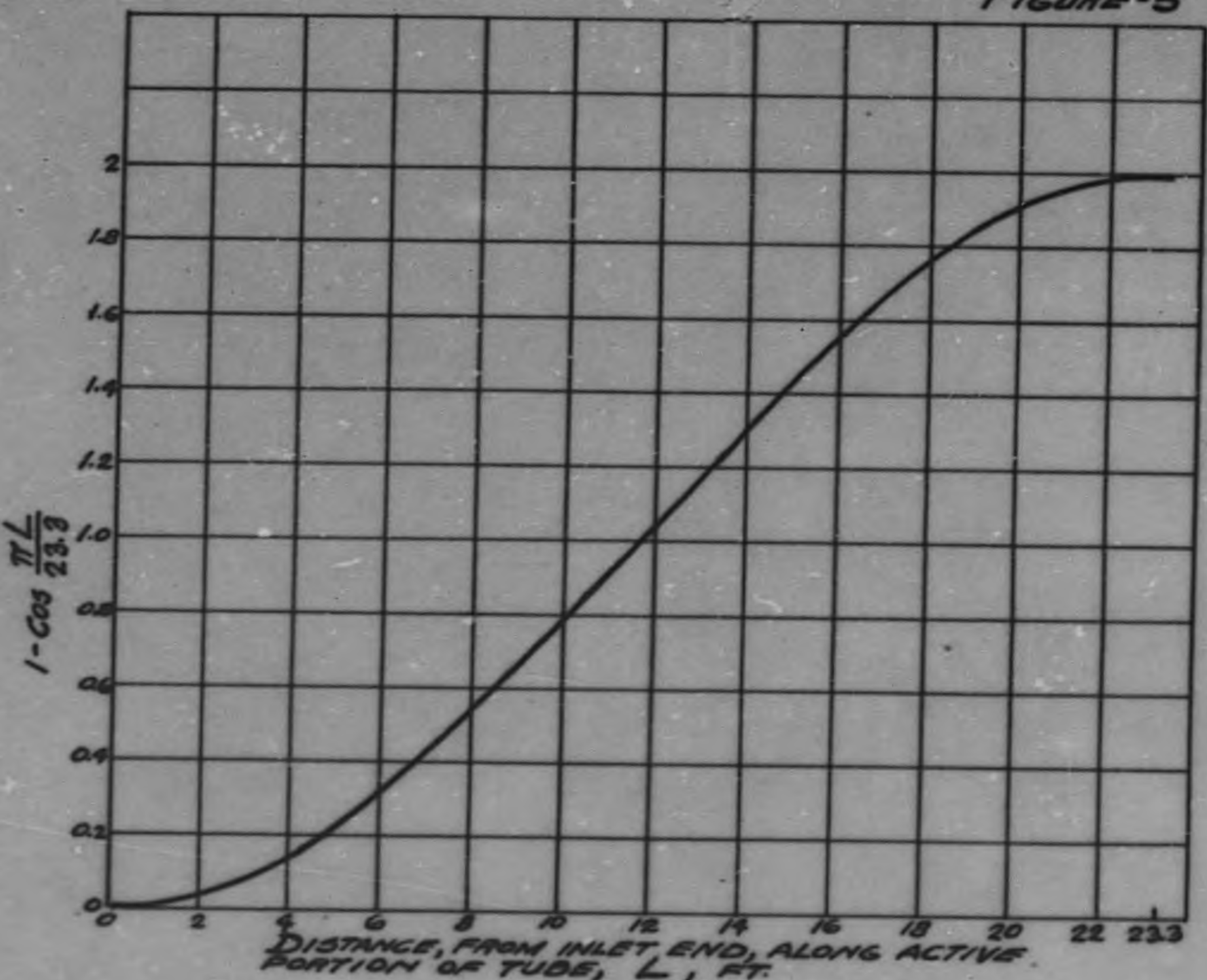


FIGURE 5: EVALUATION OF TERM $\left(1 - \cos \frac{\pi L}{23.3}\right)$ VS L, FOR USE IN EQUATIONS 1,5,6.

SECRET

FIGURE - 6

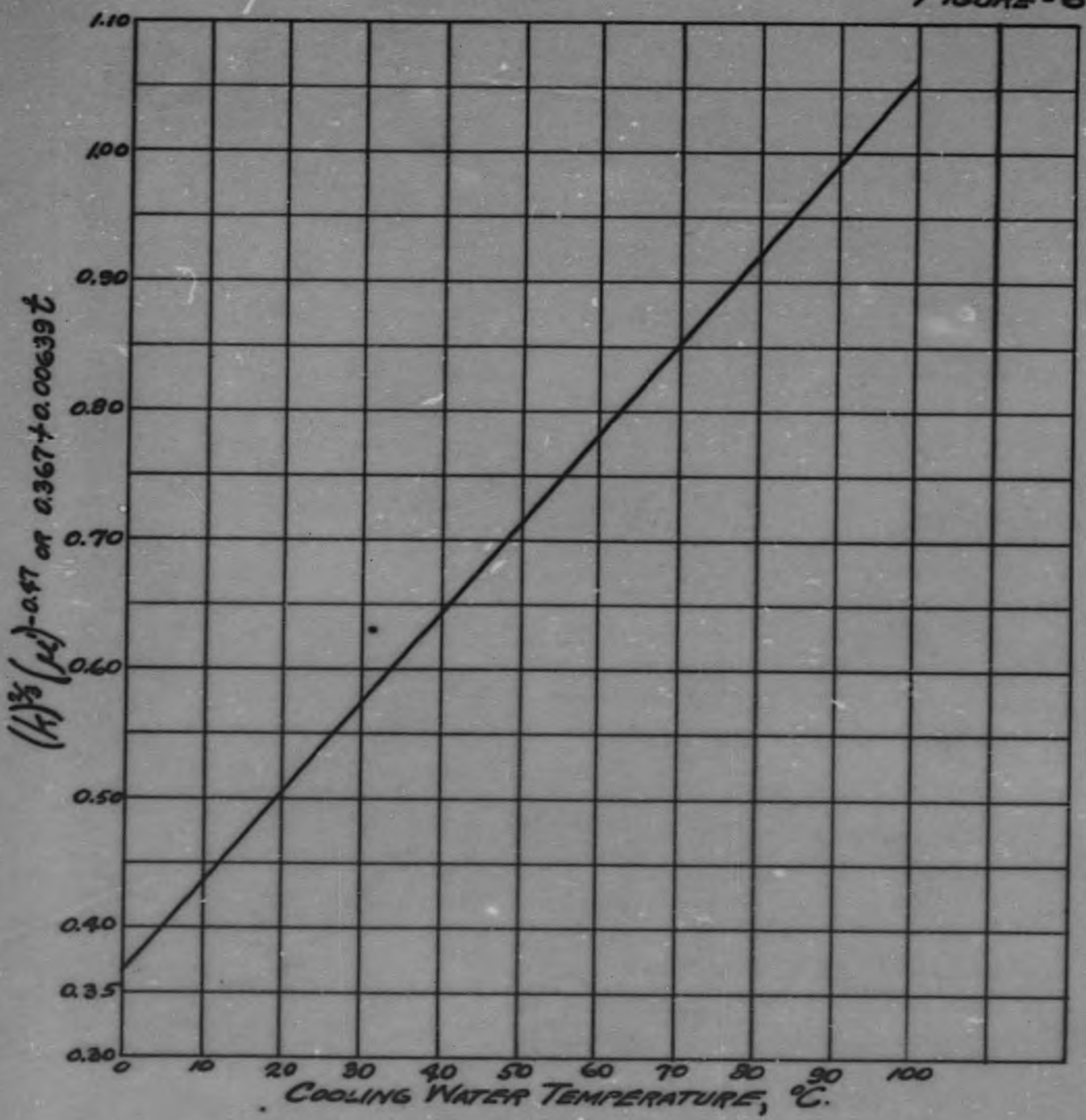


FIGURE 6: EVALUATION OF TERM (0.367 + 0.00639 t) VS t, FOR USE IN EQUATIONS 2, 7, 8.

FIGURE - 7

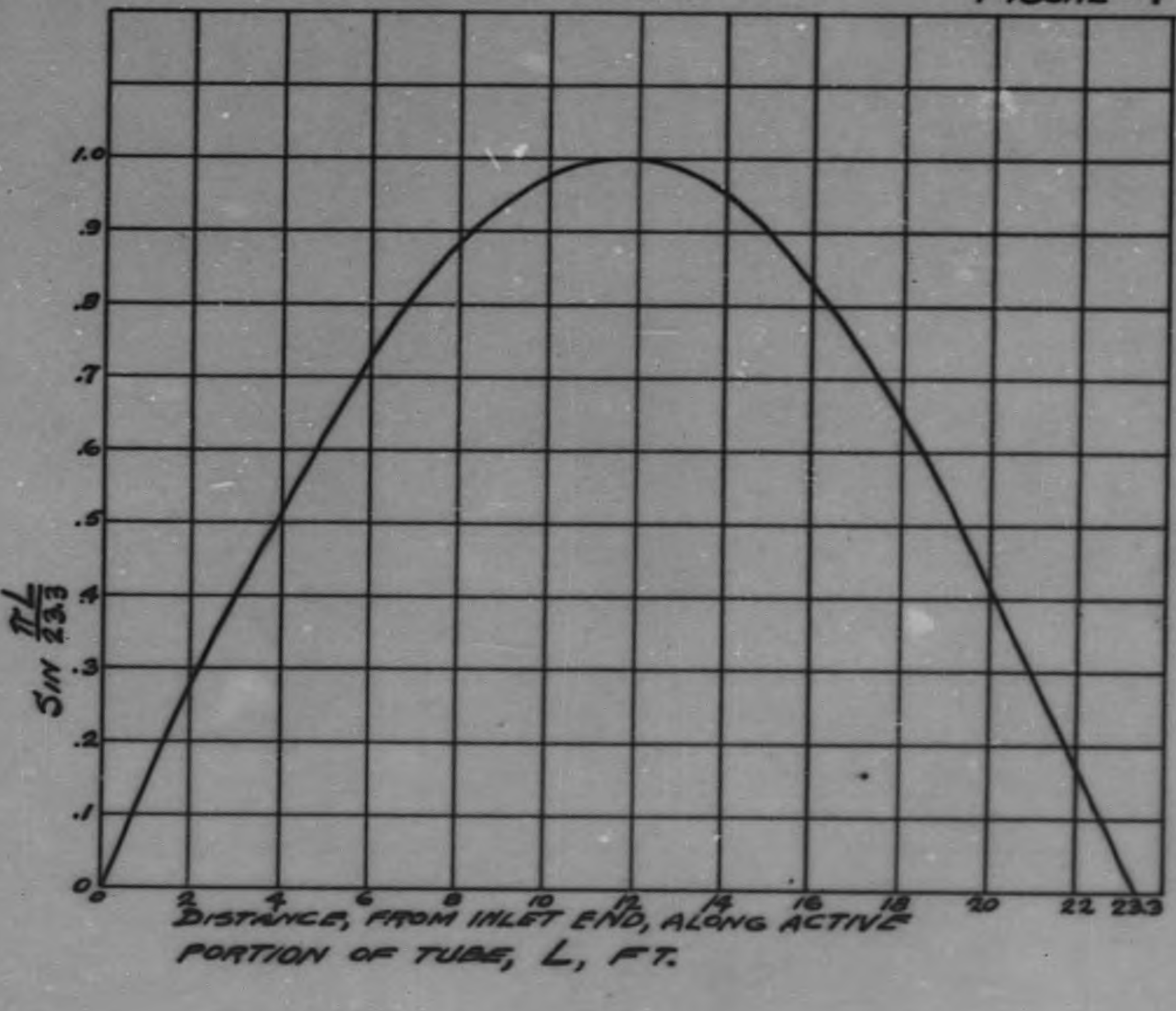


FIGURE 7: EVALUATION OF TERM $\left(\sin \frac{\pi L}{23.3}\right)$ VS L FOR USE IN EQUATION 3.

FIGURE-8.

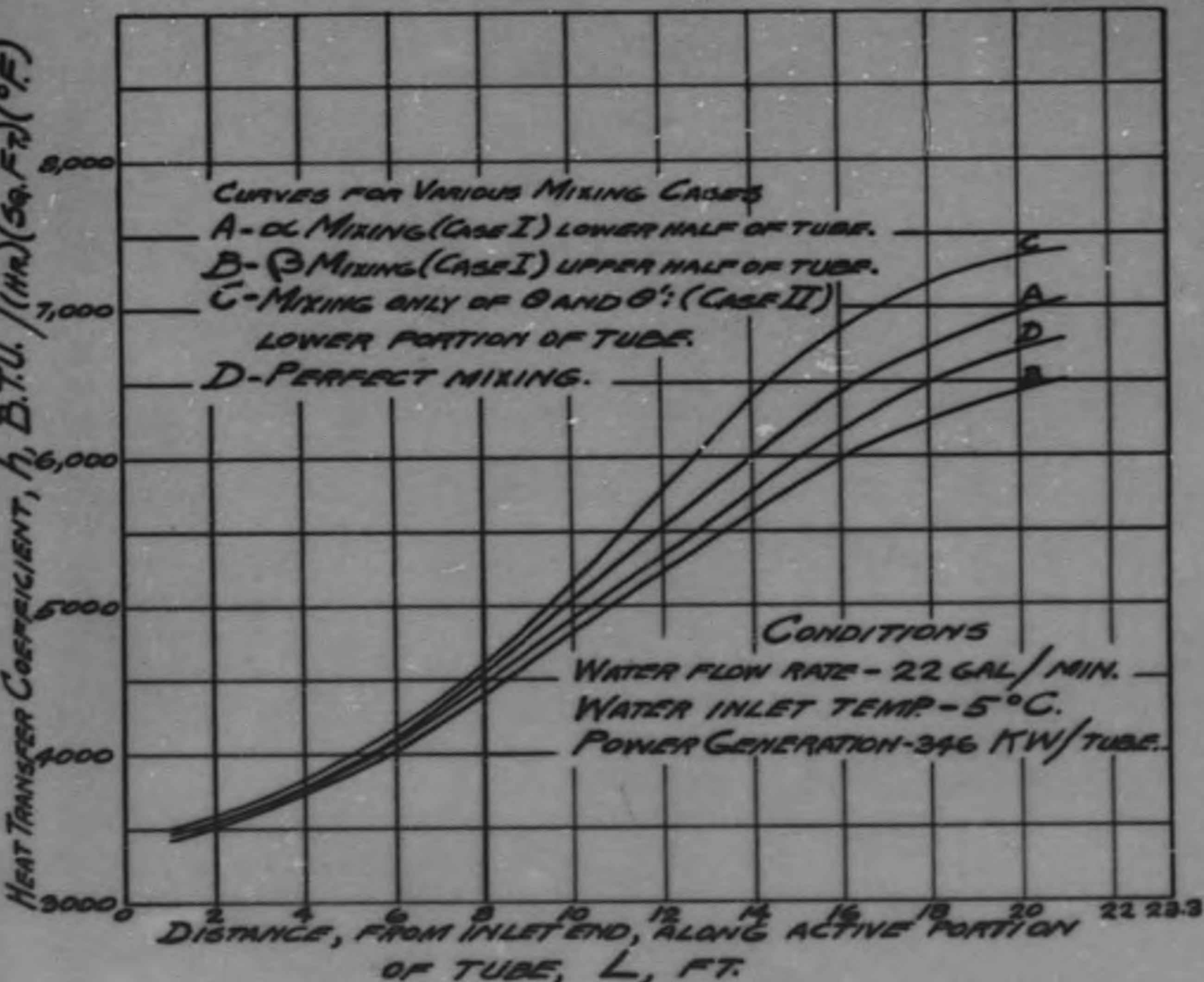


FIGURE 8: CALCULATED HEAT TRANSFER COEFFICIENT ALONG THE TUBE (SIEDER AND TATE)

FIGURE-9

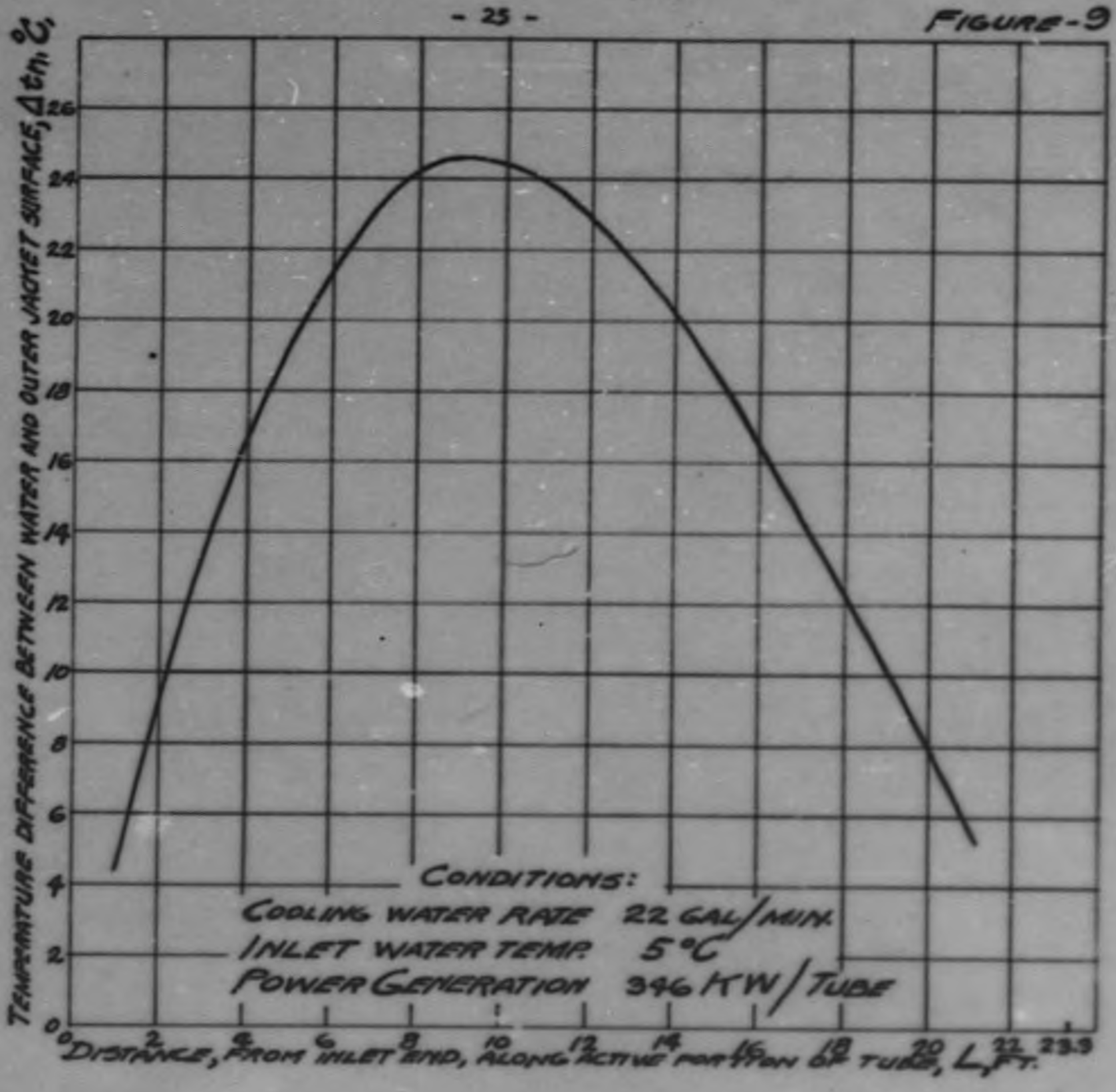


FIGURE 9: CALCULATED NORMAL TEMPERATURE DIFFERENCE BETWEEN OUTER JACKET SURFACE AND COOLING WATER.

FIGURE - 10

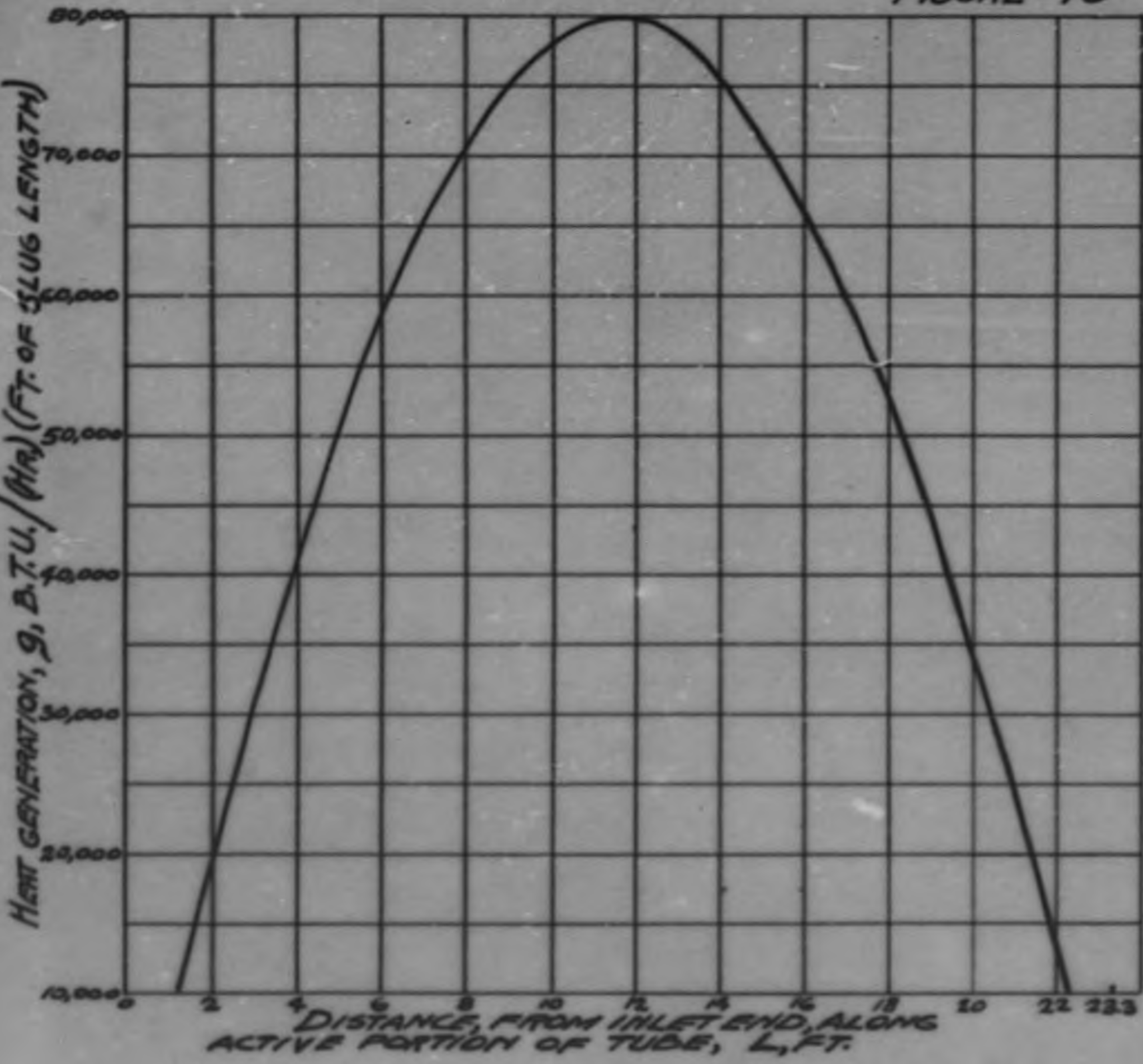


FIGURE 10: HEAT GENERATION PER FOOT OF SLUG LENGTH ALONG A TUBE. TOTAL POWER OUTPUT IN TUBE, 346 KW.

FIGURE - 11

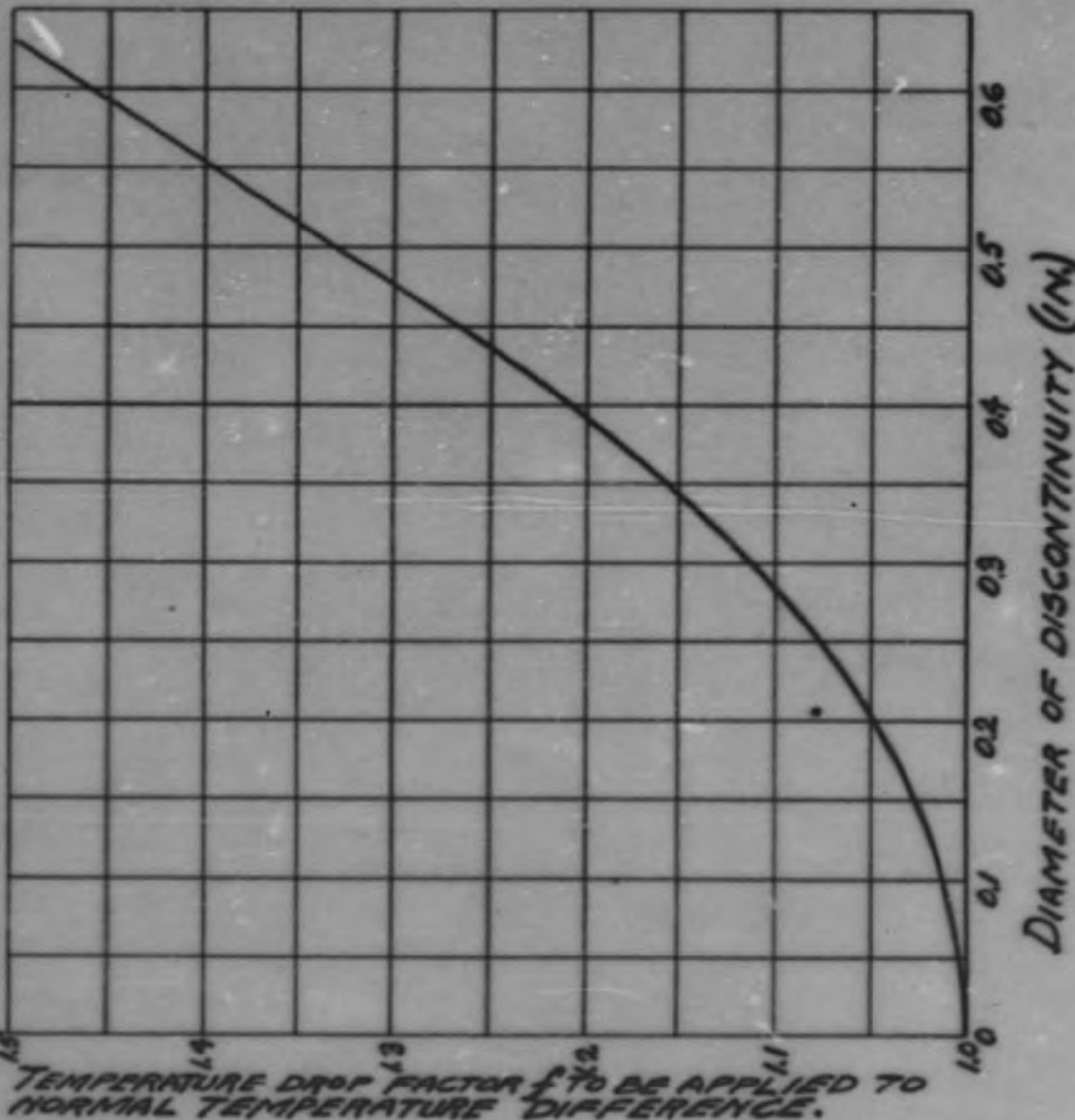


FIGURE 11: TEMPERATURE DROP FACTOR FOR CIRCULAR DISCONTINUITIES BETWEEN ALUMINUM JACKET AND SLUG.

FIGURE-12

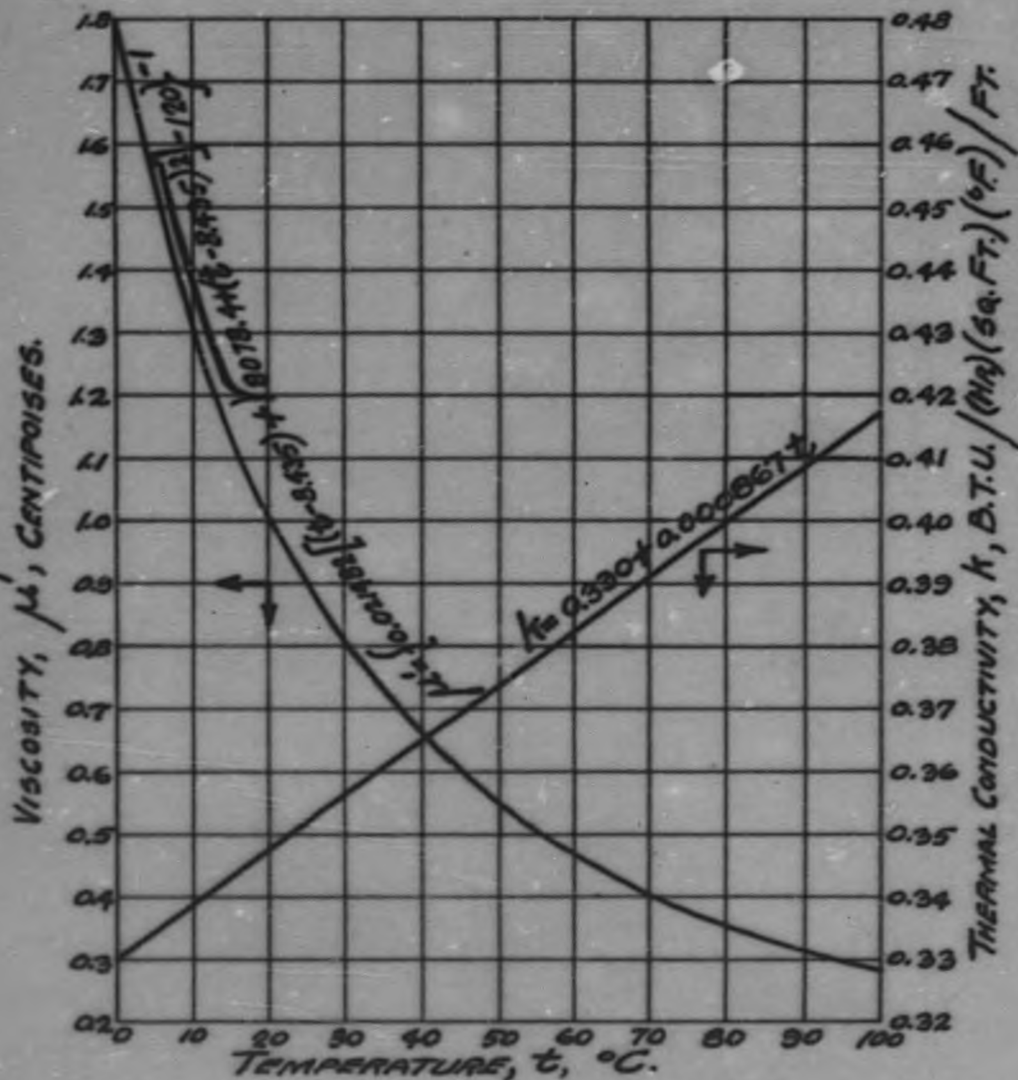


FIGURE 12: VISCOSITY AND THERMAL CONDUCTIVITY OF WATER AS A FUNCTION OF TEMPERATURE.

Exhibit A.

Sample Calculation of the Aluminum Surface Temperature.

Twelve feet from the entrance of the tube, for the worst condition possible:

the aluminum surface of the jacket near the rib at hot ring near end of slug with a defect, having a diameter of 1 cm. under the coating at the rib; power output of tube, 346 KW; cooling water rate 22 gal./min.; ∞ mixing; inlet water at 5°C.

rib $f = 1.56$)
hot ring $f = 1.12$) $f_1 \times f_2 \times f_3 = 2.1 = f$
defect $f = 1.2$)

$$t = t_1 + \frac{1.91\,(F)(P)(1 - \cos \frac{\pi L}{23.3})}{v} \qquad (1)$$

using Figure 5, L = 12 ft.

$$1 - \cos \frac{\pi L}{23.3} = 1.31$$

F = 1.05 (Case I, ∞ mixing. See Exhibit F)

$$t = 5 + \frac{1.91 \times 1.05 \times 346 \times 1.31}{22} = 38.2^\circ C.$$

$$h = 721\,(v)^{0.8}(0.367 + 0.00639\,t) \qquad (2)$$

using Figure 6 at t = 38°C.
$0.367 + 0.00639\,t = 0.610$

$$h = 721(22)^{0.8}(0.610)$$
$$= 5240$$

$$\Delta t_n = \frac{339\,(F)(P)\,\sin \frac{\pi L}{23.3}}{h} \qquad (3)$$

using Figure 7, at L = 12.

$$\sin \frac{\pi L}{23.3} = 0.999$$

$$\Delta t_n = \frac{339 \times 1.05 \times 346 \times 0.999}{5240} = 23.6^\circ C.$$

$$t_a = t + (f_1)(f_2)(f_3)\Delta t_n$$
$$= 38.2 + (1.56)(1.12)(1.2)(23.6)$$
$$= 38.2 + (2.1)(23.6) = 88^\circ C.$$

Using Graphical Method
From Figure 4, at L = 12, f = 2.1, $t_a = 87^\circ C.$

Exhibit B.

Sample Calculation of Temperature within Slug.

Temperature at center of slug at center of tube.

$r = 0$

$L = \frac{23.3}{2}$

$$0.0364\ (P)\ \sin \frac{\pi L}{23.3} - 43.6\ (r) = .0546\ (t_r - t_m) + 0.0000245(t_r^2 - t_m^2)$$

from Figure 7, $\sin \frac{\pi L}{23.3} = 1.0$ at $L = \frac{23.3}{2}$

$$0.0364\ (346)(1) - 43.6 \times 0 = 0.0546\ (t_r - 100) + 0.0000245\ (t_r^2 - 10{,}000)$$

$$t_r = 295^{\circ}C.$$

Exhibit C.

Calculation for Different Degrees of Mixing (See Figure 3.)

Perfect mixing:

max. mixing distance = 2.26 in.
heat fraction = 1
water fraction = 1
F = 1

Case I Perfect mixing of θ area with $\frac{\phi}{2}$ area. No mixing between α and δ regions.

$$\text{max. mixing distance} = 0.720\left(\theta + \frac{\phi}{2}\right) = 0.566 \text{ in.} = 1.44 \text{ cm.}$$

$$\alpha \text{ heat fraction} = \delta \text{ heat fraction} = 0.5$$

$$\delta \text{ water fraction} = \frac{\frac{\pi \times 1.598 \times 1.611}{8} - \frac{1}{2}(0.72)^2(\pi - 0.019)}{0.365}$$

$$= 0.525$$

$$\alpha \text{ water fraction} = 1 - 0.525 = 0.475$$

$$F_\alpha = \frac{0.500}{0.475} = 1.05 \qquad \text{(bottom)}$$

$$F_\delta = \frac{0.500}{0.525} = 0.954 \qquad \text{(top)}$$

Case II Perfect mixing between θ and θ'

$$\text{max. mixing distance} = 0.720\,(2\theta) = 0.372 \text{ in.} = 0.95 \text{ cm.}$$

$$\theta + \theta' \text{ heat fraction} = \frac{8 \times 0.258}{2\pi} = 0.328$$

$$\delta \text{ heat fraction} = 0.500$$

$$\phi \text{ heat fraction} = 1 - 0.328 - 0.500 = 0.172$$

$$\theta + \theta' = \frac{\frac{8 \times 0.258 \times 0.398}{2\pi} - 0.029}{0.369} = 0.275$$

$$\delta \text{ water fraction} = 0.525$$

$$\phi \text{ water fraction} = 1 - 0.275 - 0.525 = 0.200$$

$$F_{\theta + \theta'} = \frac{0.328}{0.275} = 1.19 \qquad \text{(bottom)}$$

$$F_\phi = \frac{0.172}{0.200} = 0.86 \qquad \text{(bottom)}$$

$$F_\delta = \frac{0.500}{0.525} = 0.95 \qquad \text{(top)}$$

Exhibit C. (cont'd.)

Case III No mixing between θ and θ' and ϕ areas.

max. mixing distance = 0.720 (θ) = 0.186 in. = 0.48 cm.

$\phi + \theta'$ heat fraction $= \frac{2.110}{2\pi} = 0.334$

[illegible] heat fraction = 0.500

θ heat fraction = 1 - 0.334 - 0.500 = 0.166

$\phi + \theta'$ water fraction $= \frac{(2.110)(0.398)}{(2)(\pi)(0.365)} = 0.375$

[illegible] water fraction = 0.525

θ water fraction = 1 - 0.375 - 0.525 = 0.10

$F_{\theta} = \frac{0.166}{0.10} = 1.66$ (bottom)

$F_{\phi + \theta'} = \frac{0.344}{0.375} = 0.89$ (bottom)

$F_{[illegible]} = \frac{0.500}{0.525} = 0.95$ (top)

Exhibit D.

Derivation of Equation for Cooling Water Temperature.

$$t = t_1 + \Delta' t_{0 \rightarrow L}$$

$$= t_1 + \frac{Q}{(w)(c_p)}$$

$$Q = \int_0^L q \, dL$$

$$q = a \sin \frac{\pi L}{23.3}$$

$$\int_0^{23.3} a \sin \frac{\pi L}{23.3} \, dL = P$$

$$P = \frac{46.6 \, a}{\pi}$$

$$a = \frac{\pi P}{46.6}$$

$$q = \frac{\pi P}{46.6} \sin \frac{\pi L}{23.3}$$

$$Q = \int_0^L \frac{\pi P}{46.6} \sin \frac{\pi L}{23.3} \, dL = -\frac{P}{2} \cos \frac{\pi L}{23.3} \Big]_0^L$$

$$= \frac{P}{2} \left(1 - \cos \frac{\pi L}{23.3}\right)$$

$$t = t_1 + \frac{3413 \, (F)(P)\left(1 - \cos \frac{\pi L}{23.3}\right)}{2 \times 1.8 \, (w)}$$

$$t = t_1 + \frac{950 \, (F)(P)\left(1 - \cos \frac{\pi L}{23.3}\right)}{w} \qquad (6)$$

Exhibit B.

Transformation of Heat Transfer Equation of Sieder and Tate (13).

$$\frac{hD}{k} = 0.027 \left(\frac{\mu}{\mu_f}\right)^{0.14} \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{c_p \mu}{k}\right)^{1/3}$$

$$D = \frac{0.151}{12} \qquad D^{-0.2} = 2.4$$

$$G = \frac{(w)\,144}{0.369}$$

$$c_p^{1/3} = 1$$

$$1 < \left(\frac{\mu}{\mu_f}\right)^{0.14} > 1.04 \qquad \mu_f = \text{viscosity of water at film}$$

$$h = (0.027)(2.4)(2.42)^{-0.47}\,(1)\;\; G^{0.8}\, k^{2/3}\;\; \mu'^{-0.47}$$

$$k = 0.330 + 0.000867\, t$$

$$\mu' = \left\{0.021482\left[(t-8.435) + \sqrt{8078.4 - (t-8.435)^2}\right] - 120\right\}^{-1}$$

For plots see Fig. 12.

$$k^{2/3}\, \mu'^{-0.47} = 0.367 + 0.00639\, t$$

$$h = 5.05\,(w)^{0.8}(0.367 + 0.00639\, t) \qquad (8)$$

Exhibit F.

Calculation of Normal Jacket-to-Water Temperature Drop, Δt_n.

$$q = hA\Delta t$$

$$q = \frac{\pi(P)}{46.6} \sin \frac{\pi L}{23.3}$$

$$A = \frac{(\pi)(1.440)(1)}{(12)(P)} = \frac{0.377}{(P)} \text{ sq.ft./ft. of length}$$

$$\Delta t_n = \frac{(3413)(\pi)(F)(P) \sin \frac{\pi L}{23.3}}{(46.6)(0.377)(1.8)(h)}$$

$$\Delta t_n = \frac{339\ (F)(P) \sin \frac{\pi L}{23.3}}{h} \qquad (3)$$

-1

Exhibit G.

Experimental Data of H. Kratz on Rib Temperatures, Compared with Calculated Results

Rib temperature = 35°C.

Wall temperature = 21°C.

Water temperature = 6.6°C.

Brass wall 14 mils thick.

Thermal conductivity of brass = 0.027 cal./(sq.cm.)(°C.)(sec.)/cm.

Tube should be aluminum, 40 mils thick; thermal conductivity = 0.5

Conductivity should be $\frac{0.5 \times 40}{0.27 \times 14}$ = 5.3 times greater.

Difference between rib and wall should be $\frac{14}{5.3}$ = 2.6°

$f = \frac{14.4 + 2.6}{14.4} = 1.18$ corresponds to a layer of water 0.001 inch over rib.

Exhibit H.

Experimental Data of C. P. Cabell on Rib Temperatures, Compared with Calculated Results.

TABLE IV

Results of Cabell's Work on Rib Rise

Rib Rise °C.	Power Kw/ft.	Water Temperature constant at
6.0	19	65°C.
6.5	20	
7.0	21	
7.6	22	
8.1	23	
8.6	24	
9.2	25	

Used 20 mil coating instead of 40 mils; therefore, temperature rise will be $\frac{40}{20}$ = 2 times too large.

(a) Cooling water at 65°C: conditions at 19 ft.
$q = 12.9$ Kw normal $\Delta t = 10°$

Data: Rib rise = 2.8°

$\frac{2.8}{2} = 1.4°$ $\quad f = \frac{10 + 1.4}{10} = 1.14$

(b) Cooling water at 65°C: $\theta - \theta'$ at 17 ft.
$q = 17.5$ Kw normal $\Delta t = 14.5°$

Data: Rib rise = 5.2°

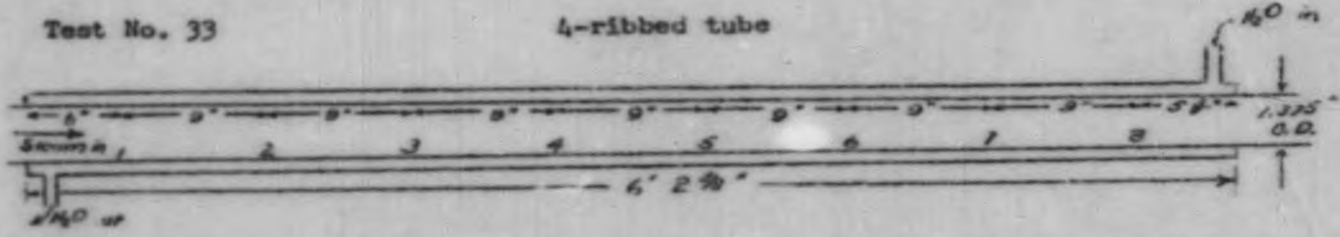
$\frac{5.2}{2} = 2.6$ $\quad f = \frac{14.5 + 2.6}{14.5} = 1.18$

∴ both correspond to about 0.001 inch of water over the rib.

Exhibit J.

Experimental Data of C. P. Cabell on Heat Transfer Coefficients (S.N.1237), Compared with Calculated Results.

Test No. 33 4-ribbed tube



	Steam Pressure	Water Temperature	Water Rate
Inlet	71 p.s.i.	138.55°F.	23.2 gpm
Outlet	65 p.s.i.	191.7°F.	

Total heat transfer area: 2.235 sq. ft.

Cross-section area of annulus: 0.3353 sq. in.

Equivalent diameter: 0.146 in.

Outside diameter of tube: 1.375 in.

Wall thickness of tube: 3/16 in. of Al.

TABLE V

Table of Data from Test on Heat Transfer Coefficients

Position	Surface Temp. °F.	Smoothed Value °F.
1	227.8	221
2	215.3	215
3	205.7	210
4	213.0	205
5	208.4	201
6	195.0	197
7	195.8	193
8	188.9	189

Exhibit J. (cont'd)

TABLE VI

Results of Analysis of Data on Heat Transfer Coefficients

Position	Steam h B.t.u./(hr.)(sq.ft.)	Water h B.t.u./(hr.)(sq.ft.)	Water h by Sieder & Tate	Water* t °F.	Metal t °F.
1	5700	7350	8650	178.5	221
2	4200	6600		173.0	215
3	3600	6200	8150	168.0	210
4	3230	6000		163.0	205
5	2980	5700		158.0	201
6	2800	5500		153.5	197
7	2660	5350	7150	149.0	193
(a) 8	2530	4200		144.5	189
(b) 8	2470	4800		141.0	189

*Assumed mixing Case II, $\Theta + \Theta'$ at each rib.

$F = 0.827$ between ribs.

(a) calculated from direction of position 1.
(b) calculated from direction of position 8.

Deviation of heat transfer coefficient, as predicted by Sieder and Tate equation, from values calculated from data.

Position	%Deviation
1	15
3	24
7	25

Method of calculation of heat transfer coefficients - basis:

$q = h_w A(t_s - t_w)$ Heat balances at any section

$q = U A(t_s - t_w)$

The water temperature may be expressed as a function of the heat input.

$t_w = c + aq$.   for imperfect mixing $t_w = c + a'q$

$a = 2.625 \times 10^{-6} L'$ $a' = 2.625 \times 10^{-6} L'F$

Exhibit J. (cont'd.)

There is a relationship between U and h_w.

$$\frac{1}{U} = \frac{1}{h_s} + \frac{1}{h_m} + \frac{1}{h_w} = \gamma + \frac{1}{h_w} \qquad \text{(definition of } \gamma\text{)}$$

$$U = \frac{h_w}{\gamma h_w + 1}$$

t_s is nearly constant; therefore, treat it as linear.

h_s is allowed to vary as $L^{-1/3}$, therefore, γ is not a constant.

$$\gamma = 1.31 \times 10^{-4} + \frac{1}{h_s}$$

Combining all these equations in[illegible]orking equations:

(1) $$\frac{t_s - c - a'q}{1 + \gamma h_w} = t_a - c - a'q$$

Know: t_s, t_a, a'

Assume: F, h_w, γ

Solve for: q

(2) Check: $q = h_w(t_a - c - a'q)$

(3) Final check: the sum of the inlet water temperature minus the exit water temperature corrected for mixing.

$$\sum a'q = 53 \text{ F}$$

The assumptions are revised until all three of these equations are consistent.

END