# *The Computation of Natural Circulation in Large Boilers*

THESIS

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*June 1948*

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*To*

MY LOVING PARENTS

*this work is respectfully dedicated*

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*Contents*

*Acknowledgments*

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Mr. John Blizard, Consulting Engineer and Director of Research; Mr. Martin Frisch, Vice President in Charge of Engineering; and Mr. Henry H. Hemenway, Mr. John W. Cartinhour, Mr. Robert A. Lorenzini, Mr. George W. Tidd, Mr. Leonard K. Swenson, Mr. John Vail, and Mr. Philip Kass—all of the Research Department.

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# *Introduction*

At present, many large capacity boilers are being built or are in the preliminary stages of design. At some phase of the design stage, the circulation characteristics of the unit must be determined in order to properly apportion riser and downcomer circuits. The simplest method of doing this would be to extrapolate the existing experimental data on similar units. However, boilers are usually built to a customer's specifications and in general no two units are made identical for the same set of operating conditions. Thus, in extrapolating, it is very possible to introduce serious faults in design because of the inapplicability of the extrapolated data. The technical data available in the literature contains many cases of overheating and subsequent failure of boiler tubes due to faulty and inadequate circulation.[(47,60,61)*]

The work of Munzinger[(7)] in 1922 has been the basis of many papers written on the subject of the circulation of steam and water in natural circulation boilers. In order to obtain a workable solution, various assumptions as to the rate of heat absorption, friction coefficients, and the state of the mixture of steam and water were made. Since the appearance of this work, many excellent papers have been published on the mechanism of heat transfer and the mechanics of boiling. While much work been done and is being done on this subject, it is still not fully understood and remains a fertile field for further investigation. In order to fully develop the subject, this work has been divided into the following three sections:—

1. A qualitative description of the mechanics of boiling, *i.e.*, the change of water to steam in a vertical tube according to the latest available data.

2. The mathematical development of all the necessary equations needed to compute the circulation characteristics of a natural circulation boiler. This section also includes curves that have been developed to facilitate circulation computations.

3. A large "Twin Furnace" unit on which circulation tests were made is completely analyzed by means of both the equations and curves of Section II. In addition, two topics of interest are discussed in this section.

A comprehensive bibliography is to be found at the end of this work. It is hoped that this bibliography will be of use for further investigation in this field.

* Numbers in parentheses refer to the bibliography at the end of this work.

# I *Analysis of the Change from Water to Steam in a Vertical Tube*

In this section, a description of the phenomena that occur in a vertical tube in which the liquid flows upward and is simultaneously being heated and evaporated is given. Most of the experimental work on this subject has been performed at or near atmospheric pressure, and very little data is available at the pressures and rates of heating usually found in large modern boilers. The analysis given here follows closely that given in the paper by Lewis and Robertson.[62] Fig. 1 shows the findings of these authors and others. The vertical scale on this figure has been very much reduced in comparison to the horizontal scale. All reference to sections *A*, *B*, *C*, *etc*., will be taken as meaning those sections which have been correspondingly marked on Fig. 1.

If the water is assumed to enter the downcomer system feeding this tube at saturation conditions, then it will be sub-cooled an amount corresponding to the increase in pressure due to the weight of the column of liquid above it, upon entering the tube. Thus, up to section *A*, heat must be added to bring this liquid to saturation conditions, and up to this section there will only be liquid in the tube. At the average entrance velocity usually found in large boilers, the Reynolds Number will be greater than 2100 and the flow will be turbulent.[33]

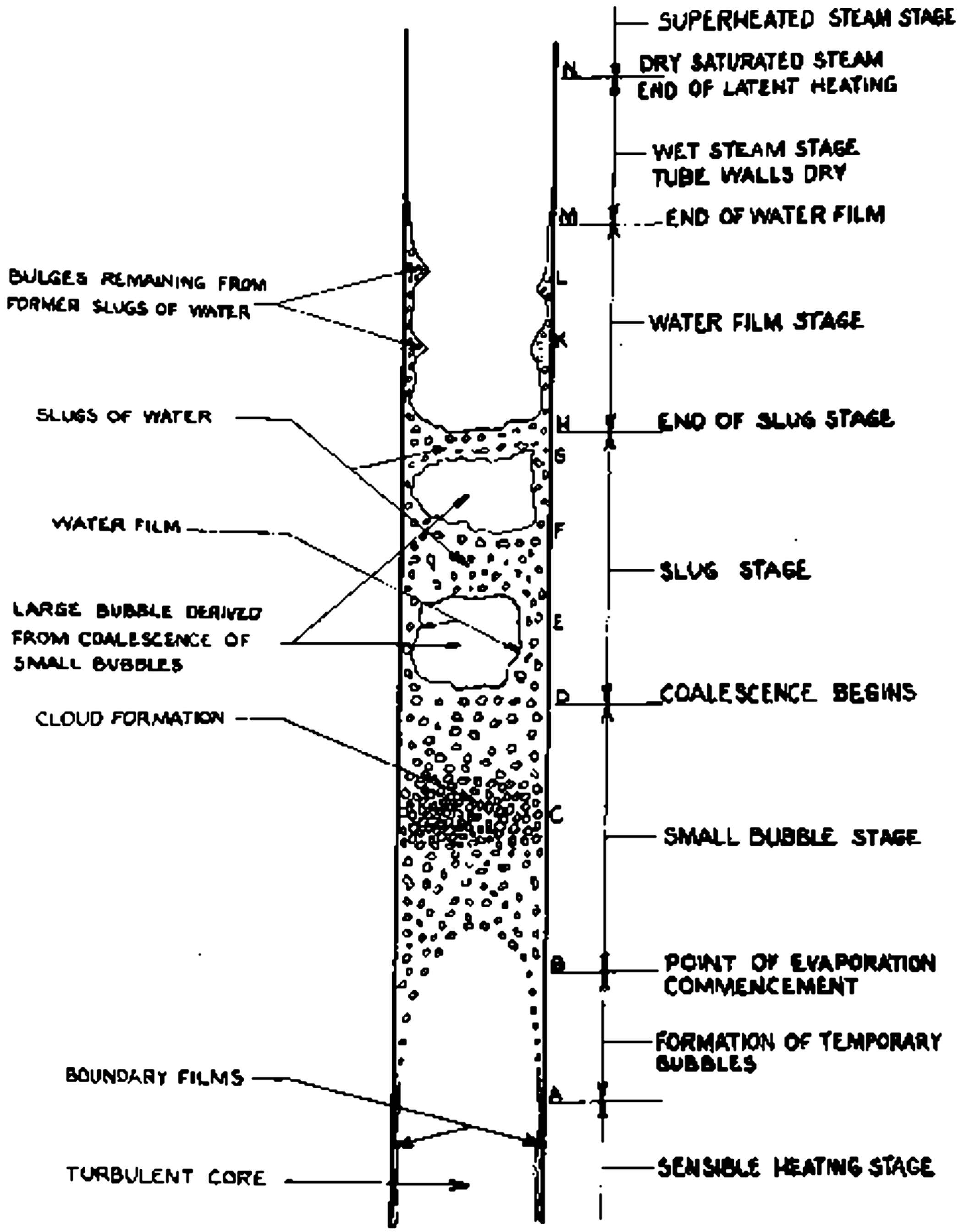



FIG. 1 CHANGE FROM WATER TO STEAM IN VERTICAL TUBE
VERTICAL SCALE DECREASED

For this condition, any cross-section through the tube up to section A will indicate the following: next to the tube wall, there exists a thin layer of fluid moving with laminar or viscous motion, then a transition zone between laminar and turbulent motion, and finally the turbulent core of the main body of the liquid. The existence of these various states of flow has been thoroughly investigated and found to be essentially correct by many investigations.(33, 49, 66, 70, 71, 87) According to McAdams,(33) heat is first transferred from the wall of the tube to and through the laminar film by conduction. Between the laminar and buffer layers, heat is transferred by mechanical mixing due to the eddy formation and also by conduction due to the existence of a radial temperature gradient. The transference of this heat to the turbulent core is accomplished by the mechanisms of turbulent mixing and conduction. In this latter process, the greater portion of the heat transfer is accomplished by the mixing and eddies that occur in the turbulent zone.

Due to the continued heating in the section from A to B, small bubbles of superheated steam will start to form in the water that is in immediate contact with the tube wall. It has been shown in Keenan's work(3) that surface tension causes the pressure in a spherical bubble to be in excess of the pressure of the surrounding liquid and inversely proportional to the radius of the sphere. Thus, a bubble of infinitely small radius would have to have an infinite pressure within it in order to remain in equilibrium. The formation of these bubbles usually starts on the many small curved surfaces of the tube wall or around nuclei of air particles or other impurities in the liquid. Before such bubble formation occurs, it is necessary that the temperature of the fluid adjacent to the wall be considerably above that of saturation conditions. (44) Once the bubble has formed, it is highly unstable, for, as the radius of the bubble increases, the tension in the enveloping sphere becomes unable to balance the excess of pressure within it.

The bubble, once formed on the tube wall or nuclei present, will continue to grow larger in diameter until it either bursts through its enveloping film or is swept off the wall of the tube by the liquid moving past the wall. These bubbles then travel inward toward the center of the tube and the turbulent core that exists there. Because of the radial temperature gradients that exist in any cross-section of the tube, the bubbles then collapse, since the temperature in the core is sufficiently high to enable the bubbles to exist. This process occurs in the section of the tube from $A$ to $B$. In the process of collapsing and subsequent condensation of the vapor in the bubble, the latent heat of vaporization of the vapor is given to the turbulent core and raises its temperature to more nearly that of the saturation temperature. When the core temperature becomes sufficiently high (above section $B$), the bubbles do not collapse. Since the temperature is high enough to maintain the equilibrium of these bubbles, they will be permanent and continue to increase in both size and number. The density of the vapor inside the bubble is less than that of the saturated liquid it displaces and there will be a relative velocity between the bubbles and the main body of the liquid. This relative velocity has been called "slip velocity" by many authors.(1,41, 65) Fig. 7 of Section II of this work shows the value of this slip velocity as a function of tube diameter and absolute pressure. This figure is from the work of Nothman[(1)] who correlated the data of Behringer[(37)] and converted it into English units from the metric units of the original data.

In the section from $B$ to $C$, the formation and growth of the bubbles continues and the distribution of the bubbles throughout the liquid in this section is fairly uniform. At some point in the tube, such as $C$, the distribution is no longer uniform but the bubbles tend to crowd together and form what has been termed "clouds". Interspersed with these clouds are sections in which the number of bubbles in any cross-section of the tube is less than the number in any corresponding section between $B$ and $C$. At some point higher in the tube, $D$, the bubbles in these clouds tend to coalesce and form a single large bubble occupying the entire flow area of the tube. This action displaces some of the liquid in the tube and forms a slug of water between any two such bubbles. This slug of water is pushed ahead by the largest steam bubble behind it and in the section between $D$ and $H$, the tube is occupied by alternate slugs of water and bubbles of steam. In this section, the velocity of the water slugs and steam bubbles is the same. As they continue to travel upward, there is an added evaporation of the water due to the continued heating and also due to the decrease of pressure due to the weight of the column of liquid.

The evaporation that occurs due to the decrease in the static head has been termed "self-steaming"[(62)] or evaporation intensification.[(1)] This self-steaming is essentially an adiabatic evaporation[(62)] and the water in the tube loses an amount of heat equal to the latent heat of vaporization at the pressure at which it occurs. Any steam in the tube will also expand adiabatically due to this decrease of pressure. Nothman[(1)] has computed the effect of this self-steaming at various pressures and found that these effects are usually small and in most cases may be neglected in circulation computations. This is the same conclusion reached by Lewis & Robertson.[(62)]

The continued evaporation in the tube causes the water slugs to become smaller and smaller until they finally cease to exist as such in the tube. At all times up to this stage in the tube, there exists a film of water on the tube wall between the wall and the bubbles of steam in the tube. Thus, above $H$, there are no slugs of water in the tube; only the water film on the tube wall. This film persists from $H$ to $M$, where continued evaporation causes the vapor to become dry and saturated, and finally the vapor becomes superheated. In the section from H to M, the film of water on the tube wall travels at a velocity somewhat less than that of the steam in this section. During the breaking up of the steam bubbles and the final evaporation of the water film on the tube wall, particles of water are entrained in the main body of the vapor flowing in the core of the tube. These particles of water continue to flow upward in the tube with the steam. If the section of the tube above $H$ is not long enough, or the rate of heating is not intense enough, these particles of water will contribute to carry-over.[(62)]

In each of the stages of the process that has been described above, there are various coefficients and rates of heat transfer due to the difference in the properties of the contents of the tube in various vertical sections of the tube. If the rate of heat transmission and the coefficients of heat transfer are assumed constant throughout the entire length of the tube, the various actions that occur in each section of the tube could be readily calculated. This, however, is not the case, and the analysis of the heat transfer is complex.

In modern boilers, it is common to find that 75% or more of the total steam production is produced in the water walls of the unit. These walls are placed such as to receive the direct radiation from the flame in the furnace. The resistance and temperature of the gas film on the outside of the tubes will be relatively constant along the tube length. The other resistances involved are those of the tube metal, the inside water (or steam) film, and the resistance of the main body of fluid in the tube.

McAdams[33] recommends the following equation for the heat transmitted from the inside tube wall to the contents of the tube:

$$\frac{hD}{K} = 0.032\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{c_p\mu}{K}\right)^{0.4} \tag{A}$$

where:

$h$ = Coefficient of heat transfer between fluid and surface—$\frac{\text{Btu}}{\text{hr.}} \times$ ft.$\times$
$D$ = Inside Tube diameter—ft.
$K$ = Thermal conductivity of fluid—$\frac{\text{Btu}}{\text{hr.}} \times$sq.ft. $\times$ °F *per* ft.
$G$ = Mass velocity—$\frac{\text{lb.}}{\text{hr.}}$ *per* sq.ft. of cross section
$C_p$ = Specific heat at constant pressure—$\frac{\text{Btu}}{\text{lb. fluid}}$ *per* °F
$\mu$ = Absolute viscosity of the $\frac{\text{fluid-lb.}}{\text{ft.-hr.}}$

For a velocity of five feet per second and an inside diameter of 0.104 feet, Lewis & Robertson[62] have evaluated this expression at various pressures and their results are given in Table A below.

**Table A**

| Pressure psis | 100 | 300 | 500 | 1000 | 2000 |
|---|---|---|---|---|---|
| Inside tube wall to sat. liquid ($h$) | 1840 | 1050 | 2000 | 1970 | 1910 |
| Inside tube wall to sat. steam | 680 | 802 | 915 | 1170 | 1670 |

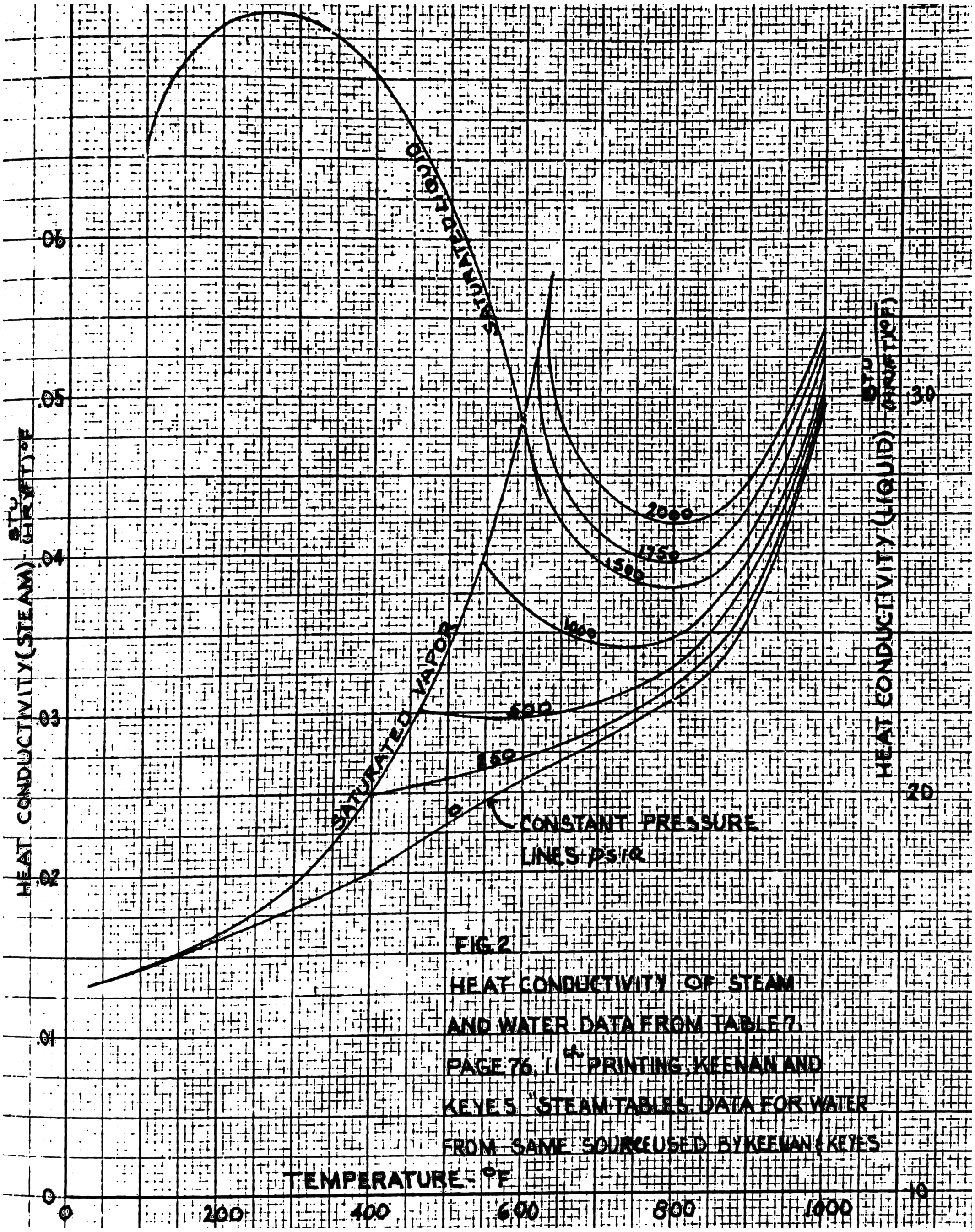



FIG. 2
HEAT CONDUCTIVITY OF STEAM AND WATER DATA FROM TABLE 7, PAGE 76, 11th PRINTING, KEENAN AND KEYES "STEAM TABLES." DATA FOR WATER FROM SAME SOURCE USED BY KEENAN & KEYES

If equation A is evaluated using the data for the physical properties given in the Appendix for the Lewis & Robertson paper it will be found that the values of $h$ given in Table A for inside tube wall to saturated steam are incorrect.

In addition to this, these authors have used the data for the specific heat and thermal conductivity from the earlier printing of the Keenan and Keyes Tables(45) and viscosity data from the work of Hawkins, Solberg, & Potter.(90) Since the appearance of these works, new tests and formulations of these physical properties have been made. The latest and probably most accurate tabulation of these properties appears in the 11th printing of the Keenan & Keyes work, dated 1945. Using this data, the author has plotted Figs. 2 and 5 and recomputed the values in Table A. The results of this computation are given in Table B.

In computing this data, the author used Table 7 of the Keenan & Keyes 11th printing for the thermal conductivity of the steam. Because of what appeared to be a discrepancy in this table, the author communicated with Professor Keyes, and in a letter dated March 11, 1948, Professor Keyes confirmed the fact that three values are incorrect. The corrected values are given below and were used to compute Table B and to plot Fig. 2.

Values for heat conductivity to be corrected in Table 7, Keenan & Keyes 11th printing:

| **Pressure psis (sat.)** | **Heat Conductivity (corr.)** |
|---|---|
| 0 | 13.16 |
| 500 | 30.15 |
| 1750 | 53.18 |

**Table B**

| Pressure psis | 100 | 300 | 500 | 1000 | 2000 |
|---|---|---|---|---|---|
| Inside tube wall to sat. liquid ($h$) | 1755.00 | 1855.00 | 1845.00 | 1810.00 | 1675.00 |
| Inside tube wall to sat. steam | 8.38 | 23.30 | 44.30 | 95.60 | 322.00 |

Since Table B shows that the value of $h$ is less for saturated steam than for the saturated liquid, it follows that the tube wall temperatures in the section containing the saturated steam will be higher than the corresponding wall temperatures in the section carrying saturated liquid. It would therefore be expected that the tube wall temperature in the section above $M$ would increase considerably over that in the sections below this point where a water film is maintained on the tube wall.

In the section from $A$ to $B$, the conditions are very complex and there is very little data available for these conditions. Various attempts have been made to correlate the heat transfer and pressure drop data by use of the Reynolds Analogy,(33) Prandtl Analogy, and combinations of various dimensionless groups. The data that is available indicates that there is a definite increase in heat transfer in this section but the data is so sketchy as to make other than qualitative predictions almost useless.

Thus far, only a vertical tube has been considered. In any boiler, the generating tubes will invariably have sections that are inclined to the vertical at various angles depending on the particular design. If the tube is only slightly inclined to the vertical, it would be expected that the description given above for a vertical tube will be essentially the same for this case. However, as the inclination is made still greater, the steam and water tend to separate, with the water flowing along the bottom of the tube and the steam flowing along the top section. If the tube is heated along the entire outside periphery or along the top, the tube metal temperature will increase in the section where there is steam. The heat transfer across this steam blanket (as it has been frequently termed) takes place by the mechanism of conduction and convection and is considerably smaller than in the section where the tube wall is wetted.

The next logical question that arises is what constitutes good circulation. It would appear that whatever the criterion is, it will be a function of the individual design, the saturation pressure and temperature, the physical properties of the mixture of steam and water, the fraction of steam by volume and by weight in any section of the tube, the amount that the tube is inclined to the vertical, the roughness of the tube surfaces, the tube material, and the temperature in the combustion zone of the furnace. Lewis and Robertson[(62)] set as the criterion of good circulation a vapor-by-weight fraction of 0.20 leaving the tube. The fraction of steam by volume would appear to be a better criterion since, as long as the water film on the tube wall is maintained, the harmful effects of steam blanketing will not occur. Partridge and Hall[(60)] have shown that the inclination of the tube plays a marked role in determining what the optimum ratio of water to steam in the tube should be and what this ratio should be to avoid tube failure in an inclined tube due to overheating.

The author has compared test data on four Twin Furnace units (similar to the one used in Section III of this work) of different capacities and operating pressures. These units have been in satisfactory operation for a number of years and no trouble has been experienced that could in any way be attributed to faulty circulation. In these units, it was found that the fraction of steam by weight leaving the tubes was always less than the criterion of 0.20 set by Lewis and Robertson, and in all cases, the fraction of steam by volume leaving the tubes was less than 0.85. As has been stated above, there can be no one universal criterion of good circulation. Each individual design would have to be tested and the results completely correlated to find the effect of each of the variables, and only then could such a criterion be established. Such a correlation has not as yet been made or, if it has, the results have not as yet been published. The amount of data on natural circulation test data in actual installations is too small to justify the establishing of any such criterion; it would merely be a guess. Therefore, no such criterion of good circulation will be set up in this work. It is hoped that such a correlation will be made in the near future, since the importance of adequate circulation in a natural circulation boiler cannot be over-emphasized if trouble-free operation is to be had in these units.

## II *Derivation of Equations for Natural Circulation Computations*

The equations developed in this section follow the method of Munzinger[7] but have been put in a convenient form advocated by Mr. John Blizard. The following nomenclature will be used throughout except as otherwise noted:

$E$ $=$ Steam produced—$\frac{\text{lbs.}}{\text{hr.}}$
$W$ $=$ Weight of water entering tube—$\frac{\text{lbs.}}{\text{hr.}}$
$q$ $=$ Rate of heat absorption—$\frac{\text{Btu}}{\text{hr.}\times\text{sq.ft.}}$ (total outside surface)
$S$ $=$ Specific volume of saturated steam—$\frac{\text{cu.ft.}}{\text{lb.}}$
$s$ $=$ Specific volume of saturated liquid—$\frac{\text{cu.ft.}}{\text{lb.}}$
$h$ $=$ Latent heat of vaporization—$\frac{\text{Btu}}{\text{lb.}}$
$V_o$ $=$ Velocity entering tube—$\frac{\text{ft.}}{\text{sec.}}$
\$ $=$ Specific volume of mixture at any point in tube—$\frac{\text{cu.ft.}}{\text{lb.}}$
$d$ $=$ Inside diameter of tube—inches
$D$ $=$ Outside diameter of tube—inches
$g$ $=$ Acceleration due to gravity—$\frac{\text{ft.}}{\text{sec}}$ *per* sec.
$f$ $=$ Friction factor
$l$ $=$ Length of tube to point considered—ft.
$L$ $=$ Total length of tube—ft.
$P$ $=$ Density of mixture relative to saturated liquid
$P_0$ $=$ Density of mixture entering tube relative to saturated liquid
$N$ $= \frac{Dq(S-s)}{75d^2h}$ — $\frac{1}{\text{sec.}}$
$\Delta P$ $=$ Static head loss—ft. of saturated liquid
$B$ $=$ Vapor friction by volume
$\lambda$ $=$ Vapor fraction by weight
$A$ $=$ Flow area—sq.ft.
$K$ $= \frac{\text{Flow area of circuit}}{\text{Flow area of } n\text{th. circuit}}$
$X$ $= \frac{NL}{KV_o}$
$SPdl$ $=$ Gravity head
$\bar{P}$ $= \frac{SPdl}{L}$
$\theta$ $=$ Angle circuit makes with horizontal
$R_e$ $=$ Reynolds number
$\mu$ $=$ Viscosity—$\frac{\text{lb.}}{\text{ft.hr.}}$
$K_B$ $=$ Bend loss coefficient

Subscript $_o$ denotes entering conditions of circuit 1. Subscript $_o n$ denotes entering conditions of circuit "n". Subscripts $_{1,2}, \ldots n$ denote properties of circuits $1, 2, \ldots n$.

In order to obtain a workable solution the following assumptions will be made:

1. The velocity of the mixture at any section perpendicular to the axis of the tube is constant.
2. The mixture of steam and water is uniform and homogeneous.
3. The velocity of the steam at any point along the tube is equal to the velocity of the water at the same point.
4. The rate of heat absorption along the heated section of the tube is constant and uniform.
5. Saturated liquid enters the tube.

While all of these assumptions are open to question, they serve as a convenient starting point. The equations developed from these assumptions may be modified later to account for any deviations from the assumed conditions. Fig. 3 shows schematically a simple circuit having equal downcomer and riser flow areas. The total amount of steam evaporated per hour is given by:

$$E = \frac{\pi D l q}{12h} = \frac{\pi D L q}{12h} \tag{1}$$

and the weight of water entering the circuit per hour is given by:

$$W = \frac{V_o \pi d l 2 \times 3600}{144 \times 4 \times s} \tag{2}$$

From equations (1) and (2), the specific volume of the mixture at any point along the tube will be given by equation (3) and the velocity of the mixture at the corresponding point is given by equation (4).

$$\$ = \frac{ES + (W - E)\,s}{W} \tag{3}$$

$$V = \frac{\$}{s} \times V_o \tag{4}$$

By substitution of (1), (2), and (3) into (4), the velocity at any section in the tube can be expressed in terms of the properties of the fluid entering the tube, the rate of heat absorption, the dimensions of the tube, and the velocity entering the tube.

$$V = \left[ \frac{\frac{\pi D L q s}{12h} + \left( \frac{V_o \pi d^2 \times 3600}{4 \times 144 \times s} - \frac{\pi D L q}{12h} \right) s}{\frac{s V_o \pi d^2 \times 3600}{4 \times 144 \times s}} \right] V_o \tag{5}$$

Re-arranging equation (5) yields equation (6).

$$V - V_o = \frac{DqL\,(S - s)}{75d^2h} \tag{6}$$

If we now let the right hand side of equation (6) equal $NL$, where:

$$N = \frac{Dq\,(S - s)}{75d^2h} \tag{7}$$

equation (6) becomes:

$$V - V_0 = NL \tag{8}$$

The substitution of (8) into (4) yields the following relations:

$$\frac{\$}{s} = 1 + \frac{NL}{V_o} \propto P = \frac{1}{1 + \frac{NL}{V_o}} \tag{10,10a}$$

From McAdams[33] pages 111 to 125, the static head loss for length $dl$ along a vertical tube can be written as:

$$-dP = \frac{4f}{d} \times \frac{V^2}{2g} \times dl + \frac{Y}{g} dV + dl \tag{11}$$

Equation (11) is in terms of feet of mixture and it is more convenient to rewrite it in terms of feet of saturated liquid.

$$-dP = \frac{4f}{d}\frac{P}{2g}\left(\frac{V_o}{P}\right)^2 dl + \frac{V_o dv}{g} + Pdl \tag{12}$$

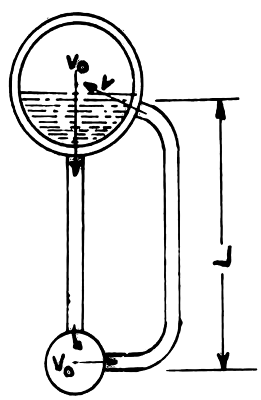



FIG.3

In equations (11) and (12) the first term on the right-hand side represents a friction loss, the second term the loss due to an increase in momentum, and the third term a loss due to the weight of the column of mixture. In addition to these losses, we must also consider an entrance and exit loss and bend losses.

In order to evaluate equation (12), it will be necessary to determine the relation between $f$, $dv$, $P$, and $l$. Thus far equations (10) or (10a) give the relation between three of these variables but as yet no relation has been developed giving the friction factor as a function of the other variables.

For isothermal flow the friction factor has been found to be a function of the Reynolds Number and the relative roughness of the tube.(33, 49, 71, 70) Fig. 4 shows a chart based on the work of Pigott(49) from which the friction factor for isothermal flow can readily be evaluated. For the complex conditions existing when water is being boiled, this relation cannot longer be used since it is not known whether the friction factor remains a function of only the Reynolds Number and the relative roughness of the tube. Even if this functional relation is true under these conditions, it still cannot be used since no data is available to evaluate the viscosity of a mixture of steam and water.

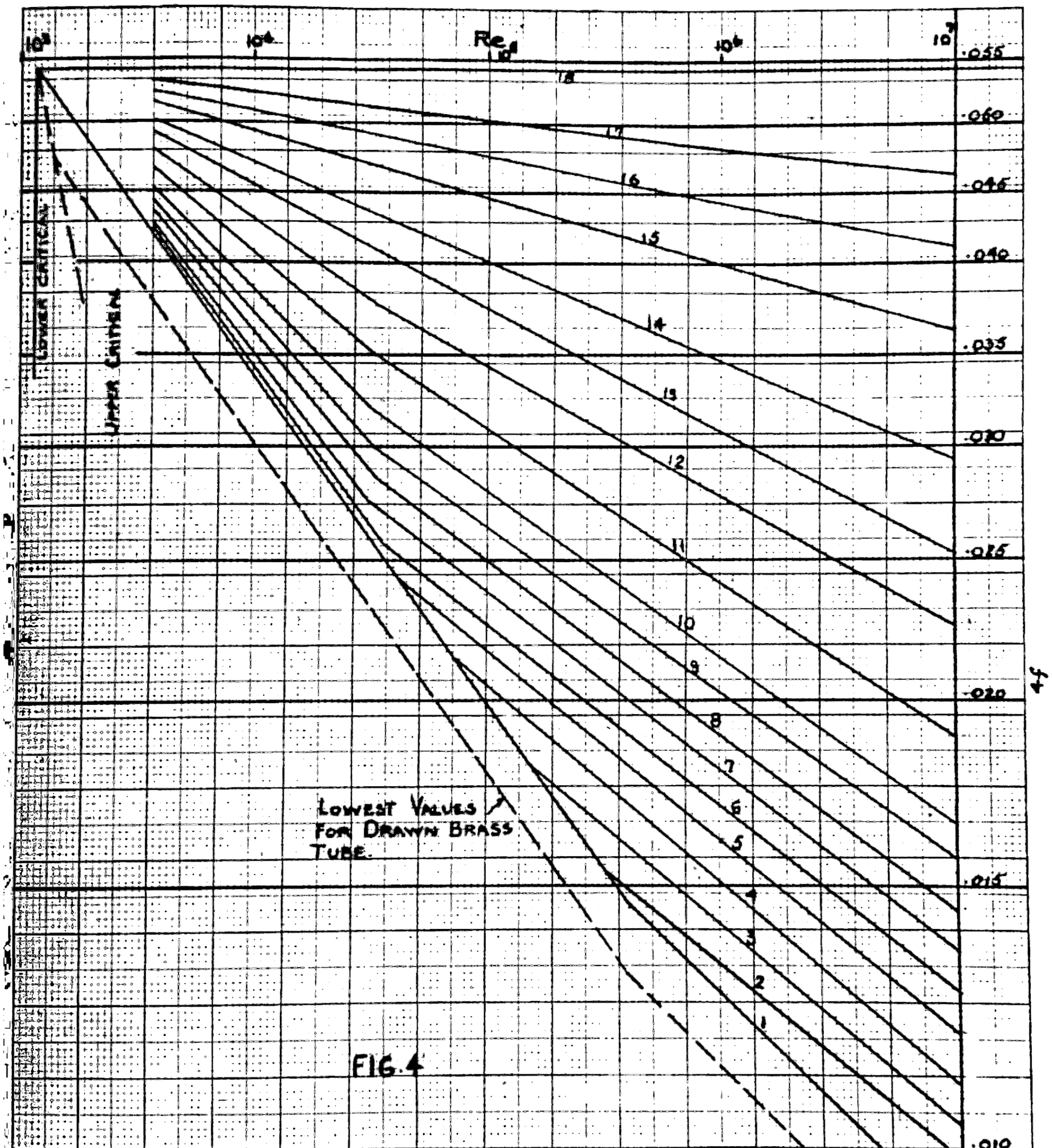



FIG. 4

| CURVE NUMBER | | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| DRAWN BRASS, TIN LEAD OR GLASS | CONDUIT DIAMETER INCHES | 0.35UP | | | | | | | | | | | | | [illegible] | | | | [illegible] |
| STEEL & WROUGHT IRON | | .072 | [illegible] | [illegible] | 6-12 | 4-5 | 2-3 | 1½ | 1-1½ | ¾ | ½ | ⅜ | ¼ | ⅛ | | | | | |
| GALVANIZED IRON | | | | 36 | [illegible] | [illegible] | 3-5 | 2½ | 1½-2 | 1¼ | ¾-1 | ½ | ⅜ | ¼ | | | | | |
| BEST CAST IRON, CEMENT, LIGHT RIVETED SHEET DUCTS | | | | [illegible] | [illegible] | 16 | 5-10 | 3-4 | 2-3 | 1½ | 1¼ | 1 | | | | | | | |
| AVERAGE CAST IRON, ROUGH FORMED CONCRETE | | | | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] | 4-5 | 3 | | | | | | | | | |
| FIRST CLASS BRICK, HEAVY & SPIRAL RIVETED | | | | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] | 8 | 5 | 4 | | | | | | | |

From the data of Boelter & Kepner[94] and their own tests, Dittus and Hildebrand[39] have proposed that Fig. 4 for isothermal flow can be used in the case of two-phase flow if the Reynolds Number is evaluated using the average specific volume of the mixture and the viscosity of the liquid. For the cases that they considered, this method gave an excellent correlation with test data. Martinelli[55,81] has recently proposed a method of correlating isothermal flow with non-isothermal flow and correlated his method with the test data reported in the paper by Davidson, *et al*.[66] If either of these methods is used to evaluate the friction factor in terms of the other variables, the expressions become very complex and very difficult to apply.

Nothman[1] has made a correlation of the expressions for friction factor as reported or proposed by various authorities and he found that the expressions have such a wide divergence that it is justifiable to assume as a first approximation that the value is essentially constant and equal to 0.006. The author has checked this value against the methods proposed by Dittus and Hildebrand and Martinelli for various assumed conditions and using Figs. 4 & 5 found that this value agrees with these methods very well.

Throughout this work, a value of friction factor equal to 0.006 will be used. The justification of this rests in what has been said above and the fact that the simplification is considerable in subsequent developments. It must be fully understood this is an assumption and must be changed if further test data show the need of it.

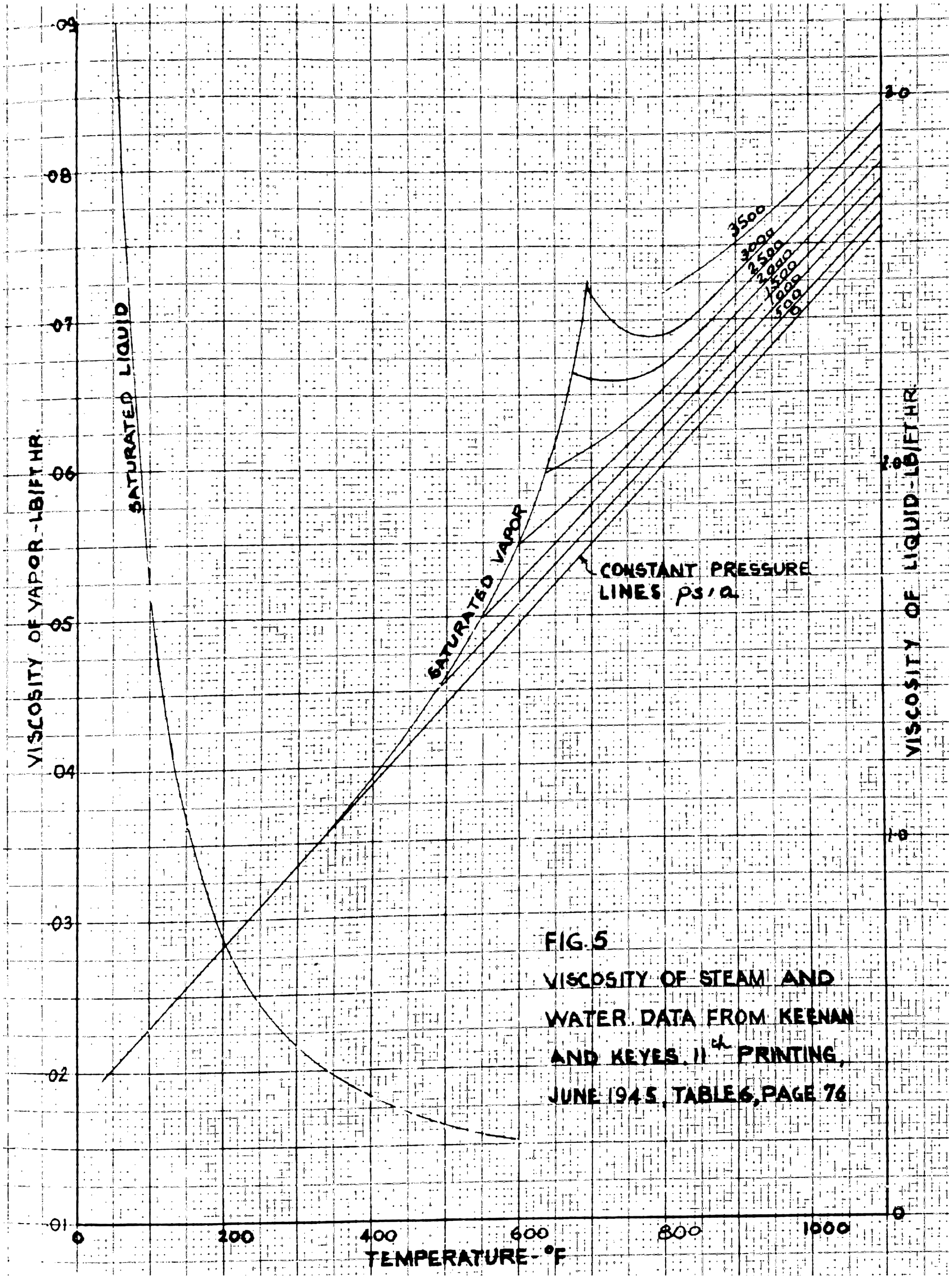



FIG. 5
VISCOSITY OF STEAM AND WATER DATA FROM KEENAN AND KEYES, 11th PRINTING, JUNE 1945, TABLE 6, PAGE 76

Equation (12) Can now be evaluated term by term as follows:

**Friction Loss**

$$-dP = \frac{4f}{2gd} \times \frac{V_o^2}{1+\frac{Nl}{V_o}} \left[1+\frac{Nl}{V_o}\right]^2 dl \tag{13}$$

$$\Delta P = \int_0^L \frac{4f}{2gd} \times V_0^2 \left[1+\frac{Nl}{V_o}\right] dl = 4f\frac{L}{d}\frac{V_o^2}{2g}\left[1+\frac{NL}{2V_o}\right] \tag{14}$$

**Acceleration Loss**

$$\Delta P = \int_0^L \frac{V_o dv}{g} = \frac{V_o}{g}\left(V_2 - V_1\right) = \frac{V_o^2}{zg} \times 2\frac{NL}{V_o} \tag{15}$$

Bend Loss

The loss in static head due to a bend in a pipe may be written as follows:[49]

$$\Delta P = K_B \frac{Y^2}{2g} \tag{16}$$

where $K_B$ is a bend loss coefficient. Fig. 6, taken from Kent,[95] allows the easy evaluation of this bend loss coefficient. It must be remembered that this loss is in addition to the loss due to friction.

**Gravity Head**

$$\text{Gravity Head} = \int_0^L \frac{dl}{1+\frac{Nl}{V_o}} = \frac{V+o}{N}\log_e\left[1+\frac{NL}{V_o}\right] \text{ for a vertical tube} \tag{17}$$

Entrance & Exist Loss

The head available in the downcomer system must also supply the kinetic energy of the liquid entering the riser system, and this is also a loss in available static head. In addition, there is an entrance loss that must be accounted for. The sum of these losses is given by:[49, 62, 65]

$$\Delta P = 1.5\frac{V_o^2}{2g} \tag{18}$$

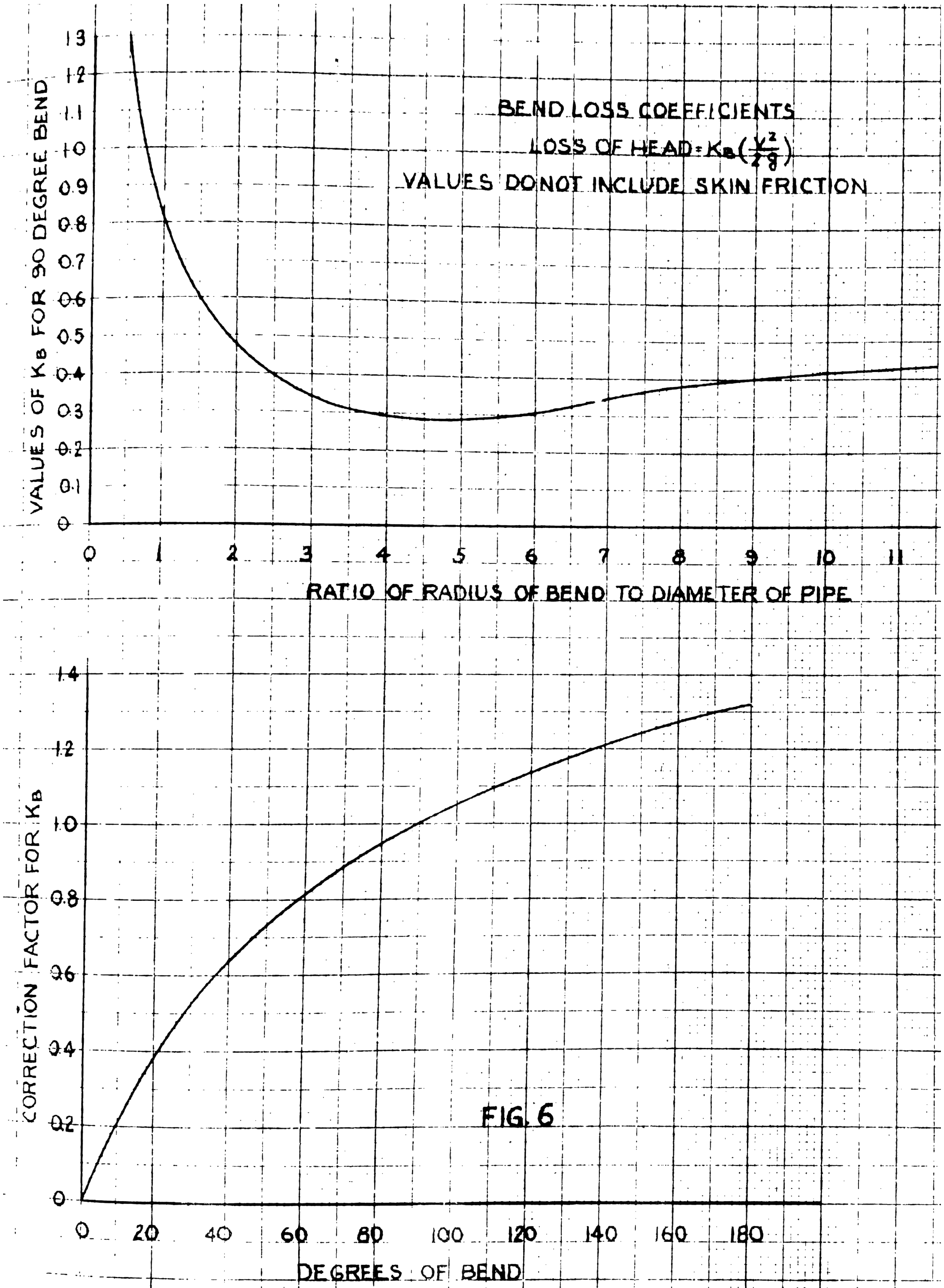



FIG. 6

TAKEN FROM KENT'S HANDBOOK (POWER) PAGE 2-13

**Vapor Fraction by Volume**

From equation (10):

$$\begin{aligned} \frac{s}{\$} &= \frac{1}{1+\frac{NL}{V_o}} = \frac{B\frac{1}{S}+(1-B)\,\frac{1}{s}}{\frac{1}{s}} \\ \text{and}\quad B &= \left(\frac{1}{1+\frac{NL}{V_o}-1}]\right)\frac{1}{\frac{s}{S}-1} = \frac{S}{S-s}\left[\frac{\frac{NL}{V_o}}{1+\frac{NL}{V_o}}\right] \end{aligned} \tag{19}$$

Vapor Fraction by Weight

$$\begin{aligned} \frac{\$}{s} &= 1+\frac{NL}{V_o} = \frac{\lambda S+(1-\lambda)s}{s} \\ \text{and}\quad \lambda &= \left(1+\frac{NL}{V_o}-1\right)\frac{s}{S-s} = \frac{s}{S-s}\left[\frac{NL}{V_o}\right] \end{aligned} \tag{20}$$

The equations developed thus far are for a single vertical tube based on the assumptions made above. If assumption 4 is not correct; that is, the rate of heat absorption along the heated section of the tube is not uniform, the equations developed above for friction loss and Gravity Head are no longer applicable. However, we may readily obtain expressions for any rate of heat absorption from those that have already been derived.

When the total heat absorbed in the tube is the same for any rate of heat absorption, the acceleration loss, entrance and exit loss, vapor fraction by weight, and vapor fraction by volume may all be properly evaluated from the expressions that have been derived for the case of uniform heat absorption.

If $q$ is expressed as follows:

$$q = \phi\,(l) + Z \tag{21}$$

where $\phi$ is a function of length and $Z$ is a constant, $Z$ may be evaluated from equation (22).

$$q_\mu^L = \int_0^L [\phi\,(\lambda) + Z]\,dl; \quad q_\mu = \text{Uniform rate} \tag{22}$$

By substitution of equation (21) into equation (6), the following results:

$$\begin{aligned} P &= \frac{1}{\frac{C(\phi(l)+Z)l}{V_o}+1} \\ \text{Where}\quad C &= \frac{D\,(S-s)}{75d^2h} \end{aligned} \tag{23}$$

**Friction Loss—variable $q$**

Equation (23) substituted in equation (13) allows us to evaluate the friction loss where the rate of heat absorption is a function of the length of the tube.

$$\Delta P = \frac{4f}{2gd} \times V_o^2 \int_0^L \left[ 1 + \int_0^l \frac{C\left(\phi\left(l\right) + Z\right)}{V_o} dl \right] dl \tag{24}$$

**Gravity Head—variable $q$**

The substitution of equation (28) into (17) yields the desired expression for Gravity Head when the rate of heat absorption is a function of the length of the tube.

$$\text{Gravity Head} = \int_0^L ] \frac{dl}{1 + \int_0^l \frac{[C\phi(l) + Z] dl}{V_o}} \tag{25}$$

Mr. John Blizard[96] has evaluated equation (25) for the case of heat transfer by convection and his development is essentially that given below. In this derivation, the densities are actual and not relative values. It follows from equation (25) that the change of specific volume of the mixture with 1 can be written as follows:

$$d\$ = \left[ \frac{c\left(\phi(l) + Z\right) dl}{V_o} + 1 \right] s \tag{25a}$$

For convection, the ratio of heat absorption at any point in the tube $q$ to the rate of heat absorption at the entrance to the tube $q_1$ is given below, where $q$ is the greatest and $b$ is a constant:

a constant: $\dfrac{q}{q_1} = e^{-bx}$

Let $x = \dfrac{l}{L}$; $\quad dx = \dfrac{dl}{L}$

$$\therefore \quad d\$ = c' q_1 e^{-bx}\, dx \quad \text{where } c' \text{ is a constant}$$

$$\$ - \$_{\text{ent}} = \int_0^x c' q_1 e^{-bx}\, dx = \frac{c'}{b} q_1 \left[ e^{-bx} + 1 \right] = \frac{c'}{b} \left[ q_1 - q \right]$$

$$\text{but} \quad \overline{P} = \int_0^1 P\, dx \quad \text{and} \quad P = \frac{1}{v} \quad ; \text{ent.} = \text{entrance}$$

$$\therefore \quad \overline{P} = \int_0^1 \frac{dx}{\$_{\text{ent}} + \frac{c'}{b} q_1 \left[ 1 - e^{-bx} \right]}$$

$$\therefore \quad \overline{P} = P_{\text{ent}} \left[ \frac{1}{1 + \frac{c'}{b} \frac{q_1}{\$_{\text{ent}}}} + \frac{1}{b\left(1 + \frac{c'}{b}\frac{q_1}{b}\right)} \log_e \left\{ \frac{c' q_1}{b \$_{\text{ent}}} \left(1 - e^{-b}\right) + 1 \right\} \right]$$

Since $\quad 1 - e^{-b} = \dfrac{q}{q_1} + 1 \quad$ and $\quad \dfrac{c'}{b} = \dfrac{\$ - \$_{\text{ent}}}{q_1 - q}$

$$\frac{\overline{P}}{P_{\text{ent}}} = \frac{\frac{q_1}{q} - 1}{\log_e \frac{q_1}{q}} \left[ \frac{\log_e \left( \frac{q_1}{q} \frac{\$}{\$_{\text{ent}}} \right)}{\frac{q_1}{q} \frac{\$}{\$_{\text{ent}}} - 1} \right] \tag{25b}$$

If $q_1$ is the least at entrance to the tube, (25b) can be used if the ratios $\frac{q_1}{q}$ and $\frac{\$}{\$_{\text{ent}}}$ are inverted where they appear in equation (25b).

Thus far, the expressions developed have been based upon the assumption that the flow of water and steam is such that there is no relative velocity between the water and the steam. That this is not the case has been shown in Section I of this work. Nothman[(1)] has derived expressions to account for this effect and, using the data of Behringer,[(37)] has found that an error of as much as 25% in Gravity Head can easily be incurred by neglecting this slip. The data of Behringer is shown in Fig. 7. This data was obtained from tests where the water was stationary and the relative velocity of the steam was measured. This is very different from the conditions that prevail in an actual boiler. Since this is the best test data available, this question of slip velocity will not be treated further here. It must be remembered that this is a fault in the analysis given here and must be taken into account when further data becomes available.

In order to facilitate the computations involved in applying the equations developed, a series of graphs has been developed. Table 1, based on the Keenan & Keyes data,[15] gives the thermodynamic properties needed in circulation computations. Fig. 10 shows the relation of $\frac{s}{S-s}$ pressure to pressure absolute.

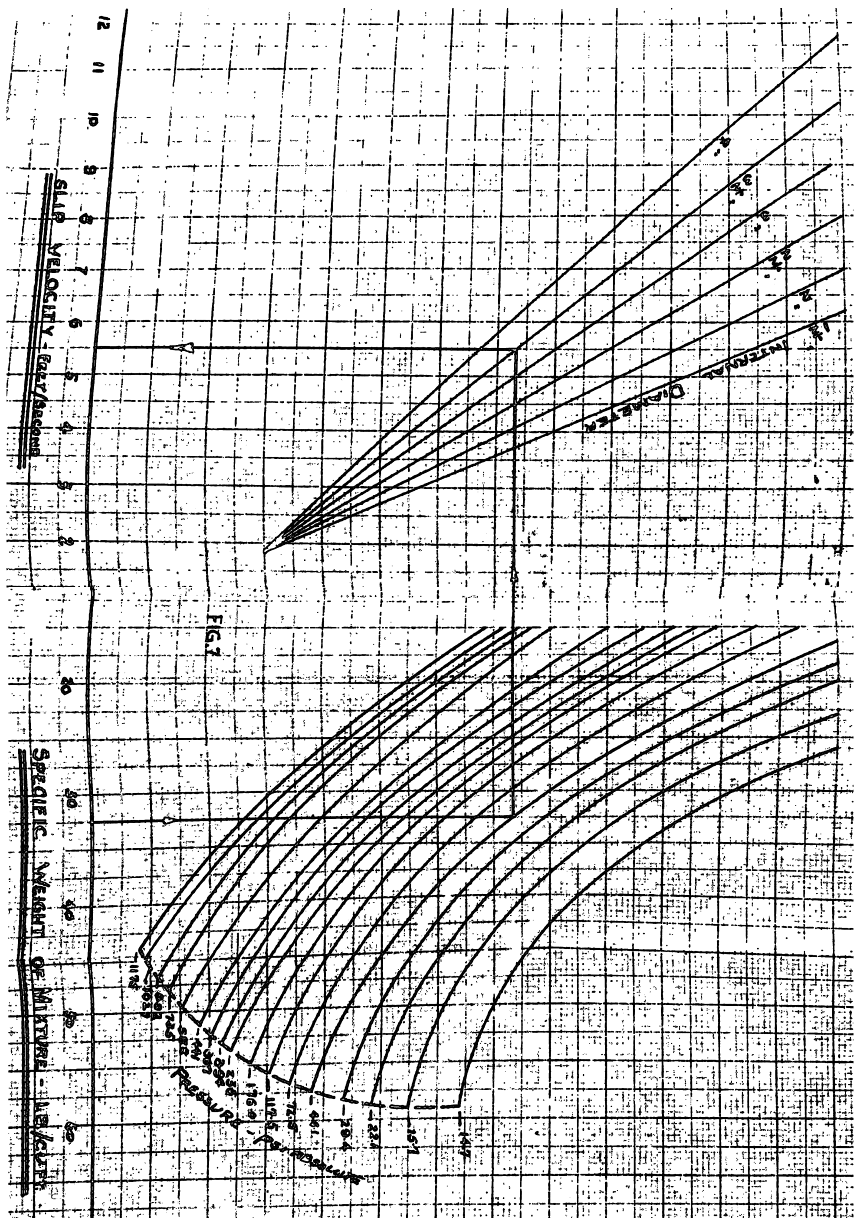

SLIP VELOCITY - FEET/SECOND
SPECIFIC WEIGHT OF MIXTURE - LB/CU FT
INTERNAL DIAMETER
PRESSURE - PSI ABSOLUTE

FIG. 7

Using the data of Table 1, Fig. 11 was plotted giving the value of $N$ as a function of tube dimensions and absolute pressure. This curve is based on a value of a equal to 10,000. For a uniform rate of heat absorption other than 10,000, the value of $N$ read from this curve must be multiplied by the ratio of the actual rate of heat absorption to 10,000.

Figs. 16 to 22 show the total static head loss for a single tube with a uniform rate of heat absorption. The term total static head loss is meant to denote the sum of the Bend loss, Acceleration loss, Friction loss, and Entrance and Exit loss. The assumption was made that there existed one bend in the circuit with a $K$ of 0.3 and that the friction factor is equal to 0.006. With these assumptions, equation (26) is readily arrived at:

$$\Delta P = \frac{{V_o}^2}{2g}\left[\left(.288\frac{L}{d}\right)\left(1+\frac{NL}{2V_o}\right)+\frac{2NL}{V_o}+1.8\right] \tag{26}$$

Thus the total static head loss is a function of $\frac{L}{d}$, $\frac{NL}{V_o}$ and the curves were developed on this basis. The further utility of these curves will be shown later in this section and in Section III of this work.

One of the assumptions made thus far was that saturated liquid entered the tube. Actually, it is possible to have sub-cooled liquid or liquid that has been partially vaporized entering. The section on series circuits allows this to be taken into account.

**Table 1**

| Pressure (psis) | $h$ | $S$ | $s$ | $S - s$ | $\frac{s}{S-s}$ | $\frac{s}{S-s}$ | $\frac{S-s}{h}$ |
|---|---|---|---|---|---|---|---|
| 100 | 888.8 | 4.432 | 0.01774 | 4.4143 | 1.000 | 0.0040 | 0.00498 |
| 125 | 875.4 | 3.587 | 0.01792 | 3.5691 | 1.003 | 0.0050 | 0.00407 |
| 150 | 863.6 | 3.015 | 0.01809 | 2.9969 | 1.005 | 0.0060 | 0.00348 |
| 175 | 850.9 | 2.532 | 0.01827 | 2.5137 | 1.007 | 0.0073 | 0.00299 |
| 180 | 843.0 | 2.288 | 0.01839 | 2.2696 | 1.008 | 0.0077 | 0.00270 |
| 200 | 833.8 | 2.042 | 0.01852 | 2.0237 | 1.009 | 0.0092 | 0.00243 |
| 225 | 815.1 | 1.8436 | 0.01865 | 1.8251 | 1.010 | 0.0102 | 0.00221 |
| 250 | 816.9 | 1.6804 | 0.01878 | 1.6616 | 1.011 | 0.0120 | 0.00204 |
| 275 | 809.0 | 1.5433 | 0.01890 | 1.5244 | 1.013 | 0.0124 | 0.00189 |
| 300 | 794.2 | 1.3260 | 0.01913 | 1.3069 | 1.015 | 0.0147 | 0.00165 |
| 340 | 780.5 | 1.1613 | 0.0193 | 1.1420 | 1.018 | 0.0169 | 0.00146 |
| 400 | 767.4 | 1.032 | 0.0195 | 1.0125 | 1.020 | 0.0193 | 0.00132 |
| 450 | 755.0 | 0.9278 | 0.0197 | 0.9081 | 1.023 | 0.0217 | 0.00120 |
| 500 | 731.6 | 0.7698 | 0.0201 | 0.7497 | 1.029 | 0.0269 | 0.00102 |
| 600 | 709.7 | 0.6554 | 0.0205 | 0.6349 | 1.035 | 0.0323 | 0.000895 |
| 700 | 680.5 | 0.5687 | 0.0209 | 0.5478 | 1.039 | 0.0382 | 0.000795 |
| 800 | 668.8 | 0.5066 | 0.0212 | 0.4794 | 1.045 | 0.0443 | 0.000718 |
| 900 | 649.4 | 0.4456 | 0.0216 | 0.4240 | 1.053 | 0.0510 | 0.000654 |
| 1000 | 630.4 | 0.4001 | 0.0220 | 0.3781 | 1.058 | 0.0582 | 0.000601 |
| 1100 | 611.7 | 0.3619 | 0.0223 | 0.3396 | 1.065 | 0.0658 | 0.000555 |
| 1200 | 593.2 | 0.3293 | 0.0227 | 0.3068 | 1.071 | 0.0740 | 0.000517 |
| 1300 | 574.4 | 0.3012 | 0.0231 | 0.2781 | 1.083 | 0.0835 | 0.000484 |
| 1400 | 556.3 | 0.2765 | 0.0235 | 0.2530 | 1.095 | 0.0930 | 0.000455 |
| 1500 | 539.6 | 0.2551 | 0.0239 | 0.2310 | 1.105 | 0.1035 | 0.000430 |
| 1600 | 519.0 | 0.2354 | 0.0243 | 0.2111 | 1.115 | 0.1151 | 0.000405 |
| 1700 | 501.1 | 0.2197 | 0.0247 | 0.1932 | 1.125 | 0.1275 | 0.000385 |
| 1800 | 481.3 | 0.2021 | 0.0252 | 0.1769 | 1.145 | 0.1429 | 0.000366 |
| 1900 | 463.4 | 0.1878 | 0.0257 | 0.1621 | 1.155 | 0.1585 | 0.000350 |
| 2000 | 444.1 | 0.1746 | 0.0262 | 0.1484 | 1.175 | 0.1761 | 0.000335 |
| 2100 | 426.4 | 0.1625 | 0.0268 | 0.1357 | 1.200 | 0.1975 | 0.000320 |
| 2200 | 408.0 | 0.1513 | 0.0274 | 0.1239 | 1.225 | 0.2210 | 0.000307 |
| 2300 | 403.9 | 0.1415 | 0.0280 | 0.1127 | 1.250 | 0.2490 | 0.000294 |
| 2400 | 388.7 | 0.1407 | 0.0287 | 0.1020 | 1.280 | 0.2810 | 0.000283 |

The data for this table are from the Keenan & Keyes work.[45]

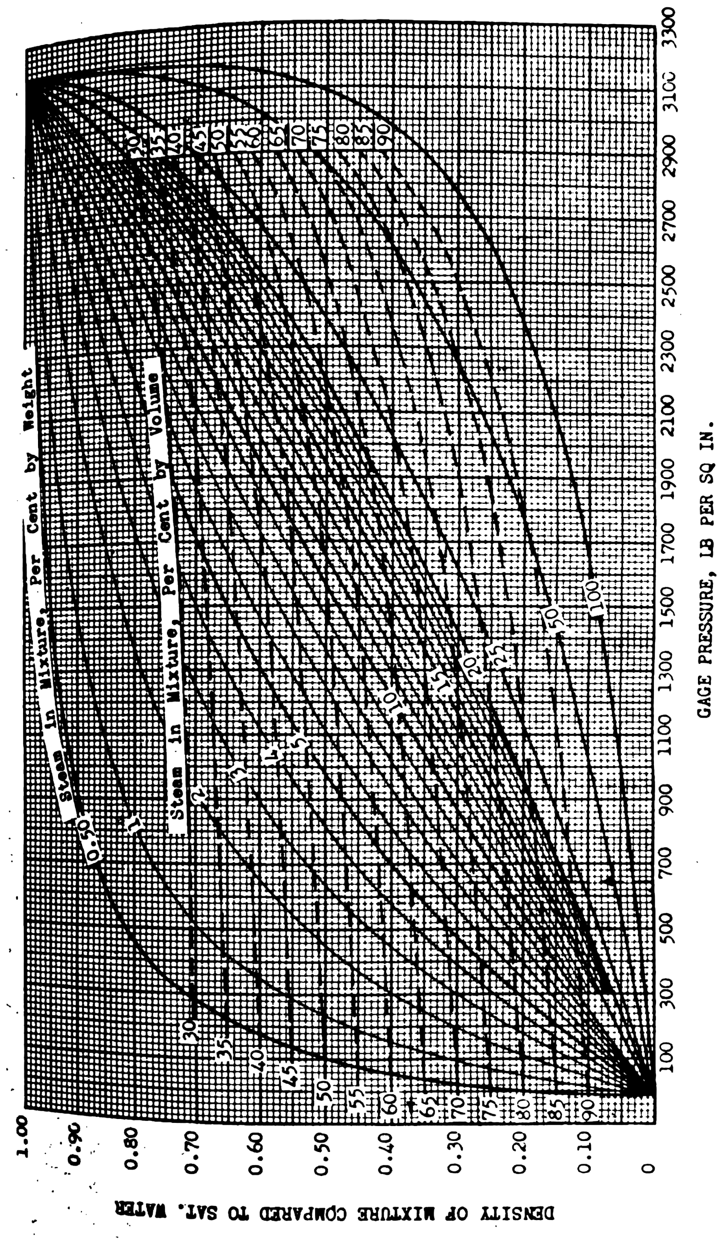



Chart showing relationships for steam-water mixtures

FIG. 8

## Series Circuits

The term “Series Circuits” as used here is meant to distinguish those circuits whose characteristics depend upon what has already happened; *i.e.*, the extent of heating and evaporation that has occurred in the tube. An example of such a circuit is shown schematically in Fig. 9. The tube has a different number of circuits for Gravity Head calculations because a new Gravity Head circuit is created by a change of slope or a change in the intensity of heating along the tube. Notes along the left of Fig. 9 denote circuits and lengths to be used for friction calculations, while notes along the right denote circuits and lengths to be used for Gravity Head circuits.

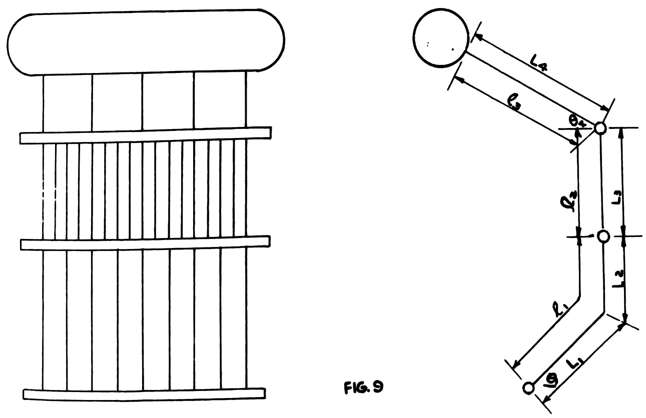


FIG. 9

While all the necessary symbols have already been given, the following symbols will be used in conjunction with the derivations in this section:

Subscript o denotes entering conditions of circuit 1.

Subscript n denotes entering conditions of circuit.

Subscripts $1, 2, \ldots, n$ denote properties of circuits $1, 2, \ldots, n$.

$$K_n = \frac{\text{Flow area of circuit}}{\text{Flow area of circuit } 1} = \frac{A_1}{A_n}$$

$$X = \frac{NL}{KV_o}; X_1 = \frac{NL}{V_o}; X_o = 0$$

All other symbols are as previously given.

The velocity entering the second circuit is:

$$V_{o2} = \frac{V_o}{P_{oz}} \times \frac{A_1}{A_2} = V_o \left[1 + \frac{N_1 L_1}{V_o}\right] K_2 = K_2 V_o \left(1 + X_1\right)$$

and the velocity entering the third circuit is:

$$V_{o3} = \frac{V_{o2}}{P_{o3}} \times \frac{A_2}{a_3} = K_2 V_o \left[1 + X_1\right] \left[1 + \frac{N_2 L_2}{K_2 V_o \left(1 + X_1\right)}\right] \times \frac{A_2}{A_3}$$

Similarly,the velocity entering the $n$th. circuitof $n$ circuits in series may be written as:

$$V_{on} = K_n V_o \left[1 + X_1 + X_2 + \ldots X_{n-1}\right]$$
$$\text{Let} \left[1 + X_1 + X_2 + \ldots X_{n-1}\right] \sum_1^n X_{n-1} \tag{27}$$

$$\therefore V_o n = K_n V_o \left[1 + \sum_1^\pi X_{n-1}\right] \tag{28}$$

Loss in Static Head Due to Friction in the $n$th. Circuit

The general equation of friction loss in any given circuit may be written as follows from equation (13);

$$-dP = \frac{4f \times dl}{2gd} = V^2$$

In terms of the entering velocity and density of such a circuit, this expression becomes:

$$-dP = \frac{4f dl V_o^2 P_o^2 P}{2gdp^2} \qquad \text{ft. of liquid at density } P_o$$

If the entering density of this circuit is taking as unity:

$$-dP = \frac{4f dl V_o^2}{2gdP}$$

The change of relative density $P$ with respect to entering relative density $P_o$ is

$$P_o = \frac{1}{1 + \frac{Nl}{V_o}}$$

If the $n$th. circuit is being considered this becomes:

$$-dP_n = \frac{4f dl V_{on}^2}{2gdn}\left[1 + \frac{Nnl}{V_{on}}\right] \text{—density } P_{on}$$

Using the entering density of circuit 1 as reference, the final differential equation for friction loss becomes:

$$-dP_n = \left(\frac{4f dl V_{on}^2}{2gd_n}\right)\left(\frac{1}{1 + \sum_1^n X_{n-1}}\right)\left(\frac{N_n l_n}{V_{on}}\right) \tag{29}$$

*Or*

$$\Delta P_n = \int_0^{L_n} \left(\frac{4f dl v_{on}^2}{2gd_n}\right)\left(\frac{1}{1 + \sum_1^n X_{n-1}}\right)\left(1 + \frac{N_n l_n}{V_{on}}\right) \tag{30}$$

Note: In the above integral, the lower limit is taken as zero because it is the point of zero of the reference axis of this circuit.

Integrating yields:

$$\Delta P_n = \frac{4fV_{on}^2}{2gd_n}\left[\frac{L_n}{1+\sum_1^n X_{n-1}}\right]\left[1+\frac{N_n L_n}{2V_{on}}\right] \tag{31}$$

Substituting for $V_{on}$ from (28) in (31) yields the desired equation for the static head loss due to friction in the $n$th. circuit.

$$\Delta P_n = \frac{4fK_n^2 L_n V_o^2}{zgd_n}\left[1+\frac{X_n}{2}+\sum_1^n X_{n-1}\right] \tag{32}$$

**Accleration Loss for the *n*th. Circuit**

The general equation of the accleration loss for any circuit is:

$$-dP = \frac{V_o dv}{g} \tag{33}$$

For the $n$th. circuit in terms of the relative density entering the first circuit:

$$\begin{aligned}
\Delta P_n &= \frac{V_{on}}{g}\int_{V_{on}}^{V}\frac{dv}{1+\sum_1^n X_{n-1}} \\
\Delta P_n &= \frac{V_{on}}{g}\left(Y - V_{on}\right)\left(\frac{1}{1+\sum_1^n X_{n-1}}\right) \\
\Delta P_n &= V_{on}^2\left(\frac{1}{1+\sum_1^n X_{n-1}}\right)\frac{N_n L_n}{V_{on}} \\
\Delta P_n &= \frac{V_o^2}{2g}K_n^2 \times 2X_n
\end{aligned} \tag{34}$$

**Entrance and Exit Loss for the *n*th. Circuit**

$$\begin{aligned}
\Delta P &= 1.5\frac{V_o^2}{2g} \qquad \text{For any circuit} \\
\Delta P_n &= \frac{V_{on}^2(1.5)}{2g\left(1+\sum_{i=1}^n X_{i-1}\right)}
\end{aligned} \tag{35}$$

$$\Delta P_n = \frac{1.5V_o^2K_n^2}{2g}\left(1+\sum_1^n X_{n-1}\right) \tag{36}$$

**Bend Loss at Any Point in the Circuit**

The bend loss in velocity heads of liquid flowing at the entrance to the bend can be obtained from Fig. 6. If the bend exists at some point in the nth circuit, the loss can be written in terms of the density entering the first circuit as follows:

$$
\begin{aligned}
V &= K_n V_o \left(1 + \sum_{i=1}^{n} X_{i-1} + \propto X_n\right) \\
\Delta P &= \frac{K_B \left[K_n^2 V_o^2 \left(1 + \sum_{i=1}^{n} X_{i-1} + \propto X_n\right)^2\right]}{2g}
\end{aligned}
\tag{37}
$$

$$
\Delta P_n = \frac{K_B K_n^2 V_o^2 \left(1 + \sum_{i=1}^{n} X_{i-1} + \propto X_n\right)}{2g} \tag{38}
$$

Where $\propto$equals the ratio of length of tube in the *n*th. circuit to the bend, to the toal length of the *n*th. circuit.

Gravity Head of the *n*th. Circuit

$$
-dH = Pdl \tag{39}
$$

The relative density at any point in a circuit with respect to the density entering the circuit is:

$$
P = \frac{1}{1 + \frac{Nl}{V_o}}
$$

For the *n*th. circuit in terms of the relative density entering the first circuit, the Gravity Head expression becomes:

$$
\begin{aligned}
\Delta H_n &= \int_0^{Ln} \frac{dl}{1 + \frac{N_n L_n}{V_{on}}} \\
\Delta H_n &= \frac{V_{on} N_n \log_e \left[1 + \frac{N_n L_n}{V_{on}}\right]}{\left(1 + \sum_{i=1}^{n} X_{i-1}\right)} \\
\text{Gravity Head} = \Delta H_n &= \frac{L_n}{X_n} \log_e \left[1 + \frac{X_n}{1 + \sum_{i=1}^{n} X_{i-1}}\right]
\end{aligned}
\tag{40}
$$

If a circuit makes an angle $\theta$ with the horizontal, the above expression must be multiplied by $\sin\theta$.

If $X_n \to O$, the expression for the Gravity head becomes of the form $\frac{O}{0}$, which is indeterminate. In the limit:

$$X_n \to O_1 \frac{L_n \sin\theta \log_e \left[1 + \frac{X_n}{1+\sum^n X_{n-1}}\right]}{X_n}$$

$$X_n \to O_1 \frac{\frac{d}{dX_n}\left(L_n \sin\theta \log_e \left[1 + \frac{X_n}{1+\sum_{i=1}^n X_{n-1}}\right]\right)}{\frac{d}{dX_n}\left(X_n\right)} \tag{41}$$

$$= L_n \sin\theta \left(\frac{1}{1 + \frac{X_n}{1+\sum_{i=1}^n X_{n-1}}}\right)\left(\frac{1}{1+\sum_{i=1}^n X_{n-1}}\right)$$

Setting $X_n = 0$ in this yields (41).

**Vapor Fraction by Weight Leaving with *n*th circuit**

If the vapor fraction by weight of any circuit is $\lambda$, then the ratio of specific volume of the mixture at any point in the circuit to the specific volume is:

$$\frac{\$}{\$_{\text{entering}}} = 1 + \frac{Nl}{V_o} = \frac{\lambda S + (1-\lambda)s}{s} = 1 + \frac{\lambda}{s}(S - s)$$

$$\lambda = \frac{Nl}{V_o} \times \frac{s}{S-s} \tag{42}$$

For the $n$th. circuit and compensating to the entering density of the first circuit:

$$\lambda_n = \frac{\left[\sum_{i=1}^n \frac{N_n L_n}{V_{on}}\right]}{\left[\frac{1}{1+\sum_{i=1}^n X_{i-1}}\right]} \times \frac{s}{S-s}$$

$$\lambda_n = \frac{s}{S-s} \frac{\left[\frac{\sum_{i=1}^n X_n}{1+\sum_{i=1}^n X_{i-1}}\right]}{\left[\frac{1}{1+\sum_{i=1}^n X_{i-1}}\right]} \tag{43}$$

$$\lambda_n = \frac{s}{S-s}\left[\sum_{i=1}^n X_n\right]$$

Vapor Fraction by Volume Leaving $n$th. Circuit

The ratio of vapor by volume to vapor fraction by weight is:

$$\frac{\frac{B}{\lambda}}{\frac{\frac{S}{s-S}\left(\frac{Nl}{V_o}\right)}{1+\frac{NL}{V_o}}} \frac{s}{S-s} \times \frac{Nl}{V_o}$$

for the $n$th. circuit:

$$B_n = \frac{\frac{S}{S-s}\left[\sum_{i=1}^{n} X_n\right]}{\left[1 + \sum_{i=1}^{n} X_n\right]} \tag{44}$$

The equations developed in this section for Series Circuits enable the computation and prediction of the circulation in any circuit regardless of the number of circuits for Gravity Head or Friction. It is obvious, however, that if there are many such circuits, as is usually the case in a boiler, the entire process becomes tedious and involved. For this purpose, the curves developed for simple circuits may be used.

The sum of the losses in the $n$th. circuit can be written as follows if $K_B$ is taken as 0.3 and $f$ equal to 0.006.

$$\Sigma\Delta P_n = K_n^2 \frac{V_o^2}{2g}\left[.288\frac{L_n}{d_n}\left[1 + \frac{X_n}{2} + \sum_{1}^{n} X_{n-1}\right] + 2X_n + 1.8\left[1 + \sum_{1}^{n} X_{n-1}\right]\right] \tag{45}$$

Comparison of the above expression with equation (26) shows that when $\sum_n^1 X_{n-1} = 0$, the two expressions are equivalent. Therefore, to evaluate the total static head loss for the $n$th. circuit, the value of $\Delta P$ is read from the proper curve at the corresponding value of $n$, and the value of $\Delta P$ at $n = 1$ is multiplied by the value of $X$. The sum of these values is the total static head loss for the $n$th. circuit.

A graphical solution for the Gravity Head of the $n$th. circuit is given in Fig. 14. Thus, at a given value of total head required, it can readily be evaluated.

Should the entering liquid of the first circuit be subcooled or contain any steam, the computations can still be made on the basis of the equations for series circuits by the introduction of a fictitious circuit before the first circuit. This concept and its use presents no difficulties.

The next section gives all of the computations for the prediction of the circulation in a large boiler utilizing all the curves and equations developed so far. The method used is similar to that used in Hydraulics.[49] It consists essentially of evaluating equation (46):

$$\text{Head available} - \text{Downcomer losses} = \text{Gravity Head} + \sum \text{Riser}_{\text{losses}} \tag{46}$$

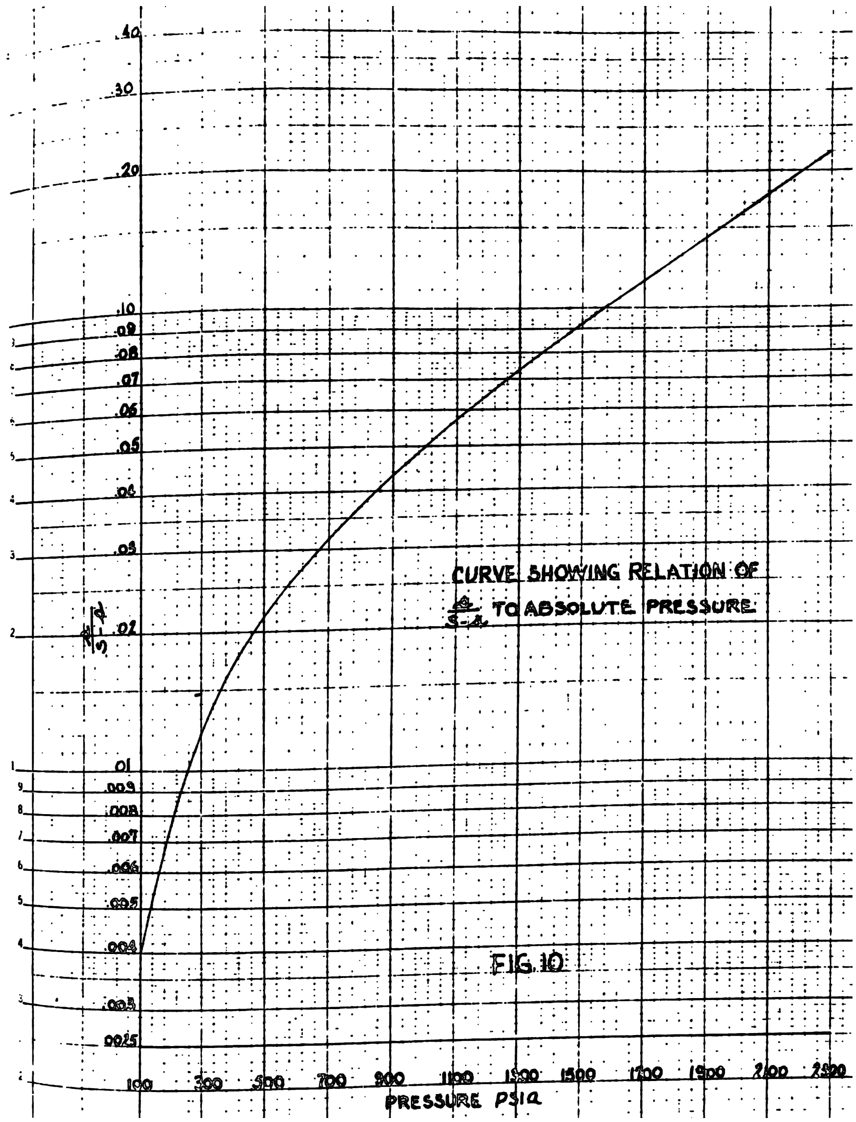

CURVE SHOWING RELATION OF
$\frac{d}{s-d}$ TO ABSOLUTE PRESSURE
PRESSURE PSIA
100 300 500 700 900 1100 1300 1500 1700 1900 2100 2300
.40 .30 .20 .10 .09 .08 .07 .06 .05 .04 .03 .02 .01 .009 .008 .007 .006 .005 .004 .003 .0025

FIG. 10

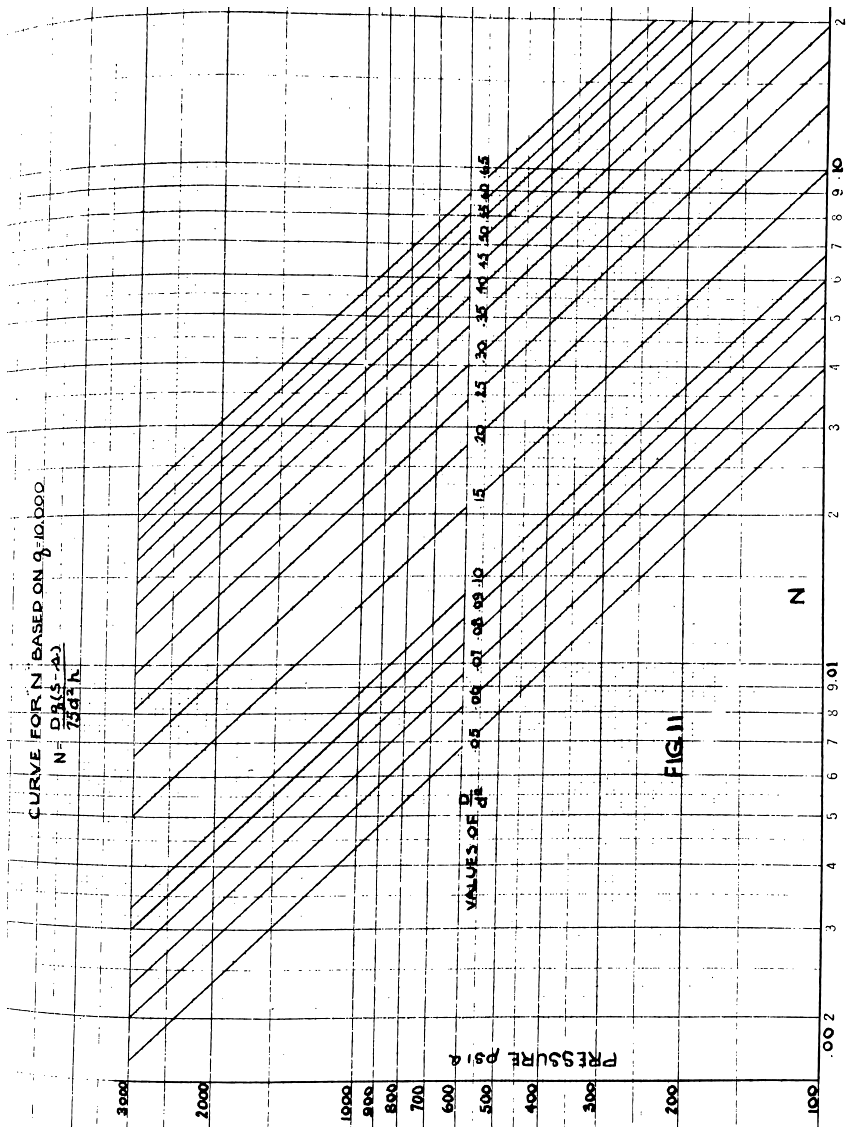
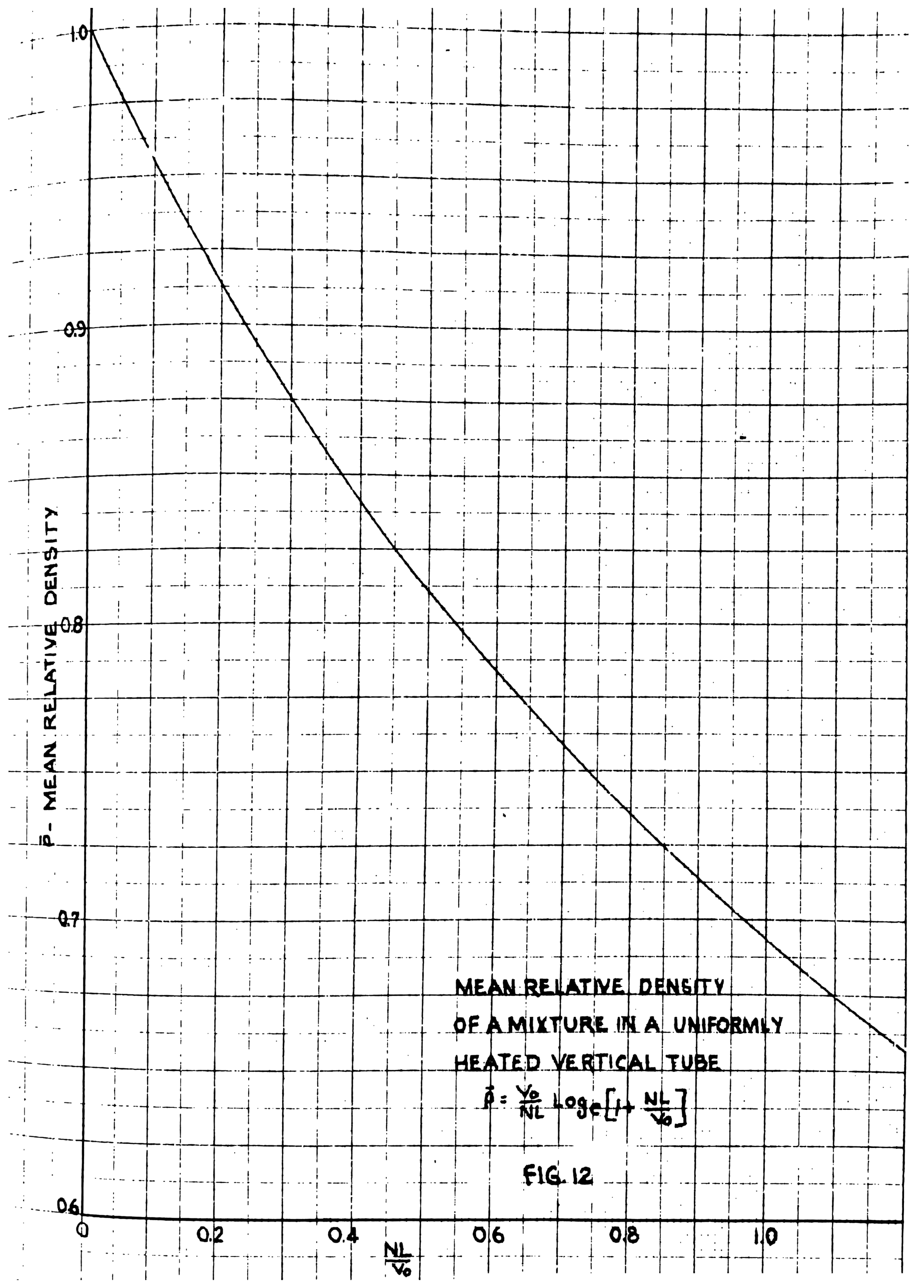
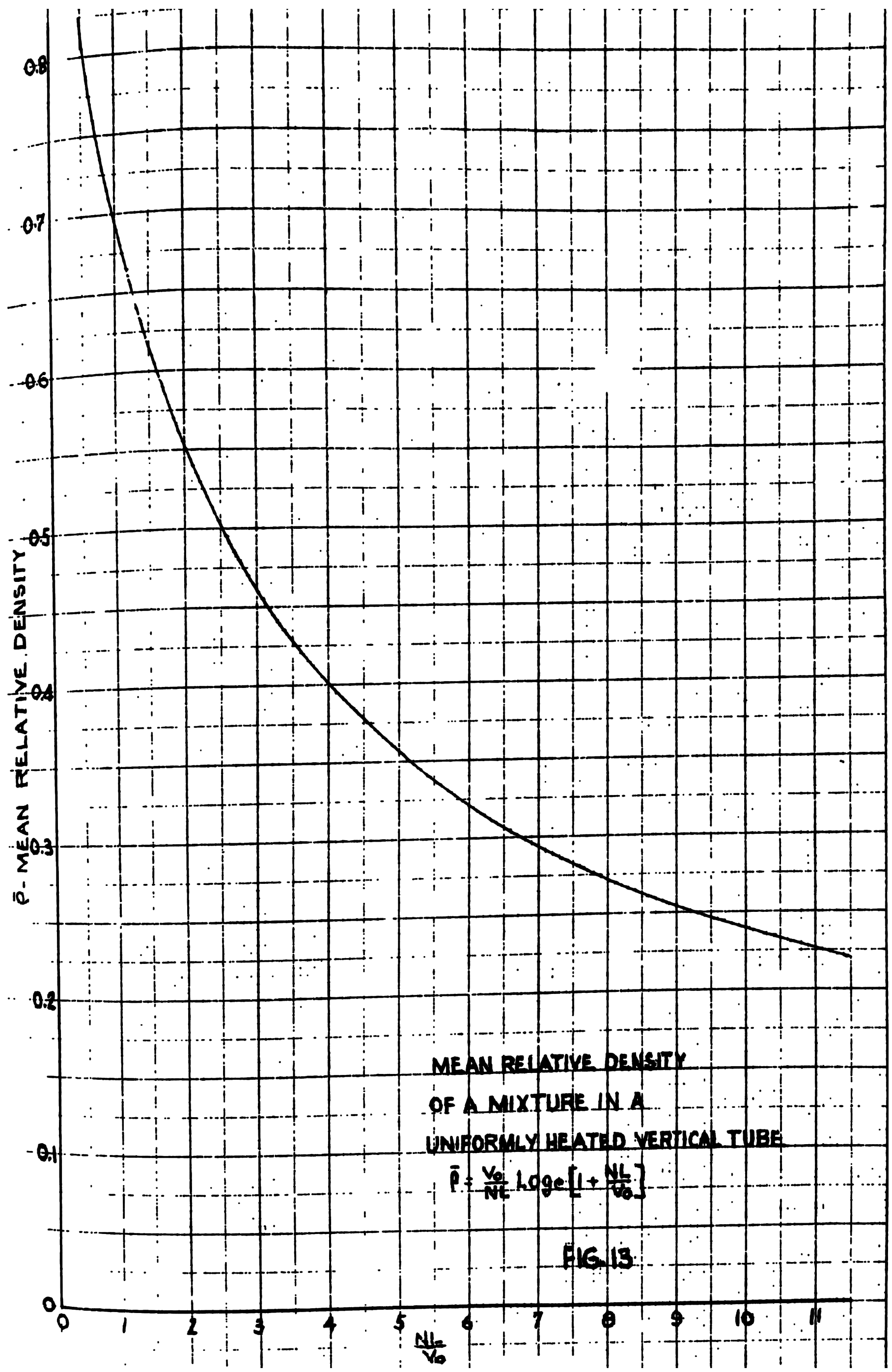



FIG. 11


1.0
0.9
0.8
0.7
0.6
$\bar{\rho}$ - MEAN RELATIVE DENSITY
0
0.2
0.4
0.6
0.8
1.0
$\frac{NL}{V_0}$
MEAN RELATIVE DENSITY
OF A MIXTURE IN A UNIFORMLY
HEATED VERTICAL TUBE
$\bar{\rho} = \frac{V_0}{NL} \log_e\left[1 + \frac{NL}{V_0}\right]$
FIG. 12


MEAN RELATIVE DENSITY
OF A MIXTURE IN A
UNIFORMLY HEATED VERTICAL TUBE
$\bar{\rho} = \frac{V_0}{NL} \log_e \left[1 + \frac{NL}{V_0}\right]$
FIG. 13
$\bar{\rho}$ - MEAN RELATIVE DENSITY
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
1
2
3
4
5
6
7
8
9
10
11
$\frac{NL}{V_0}$

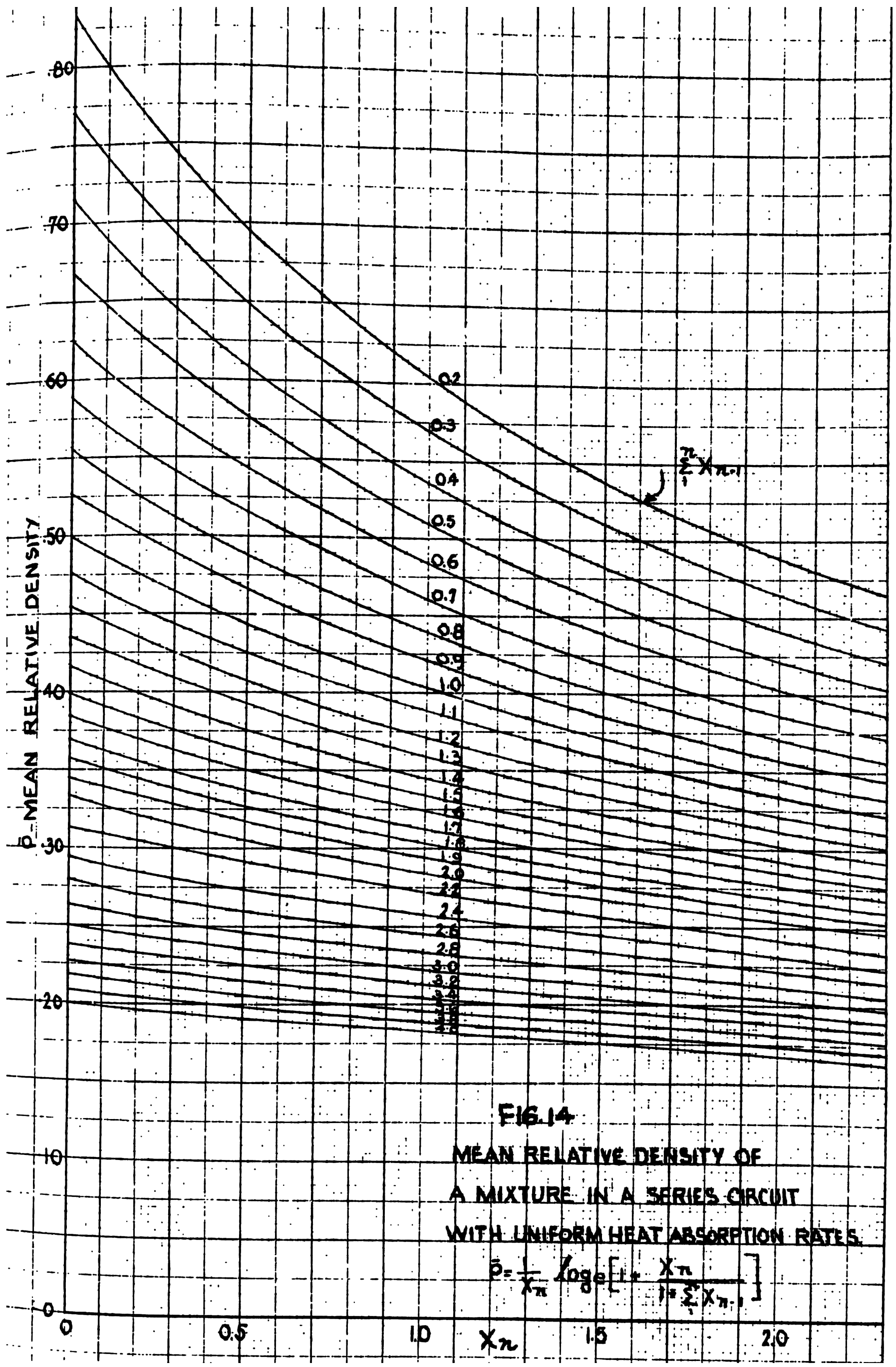
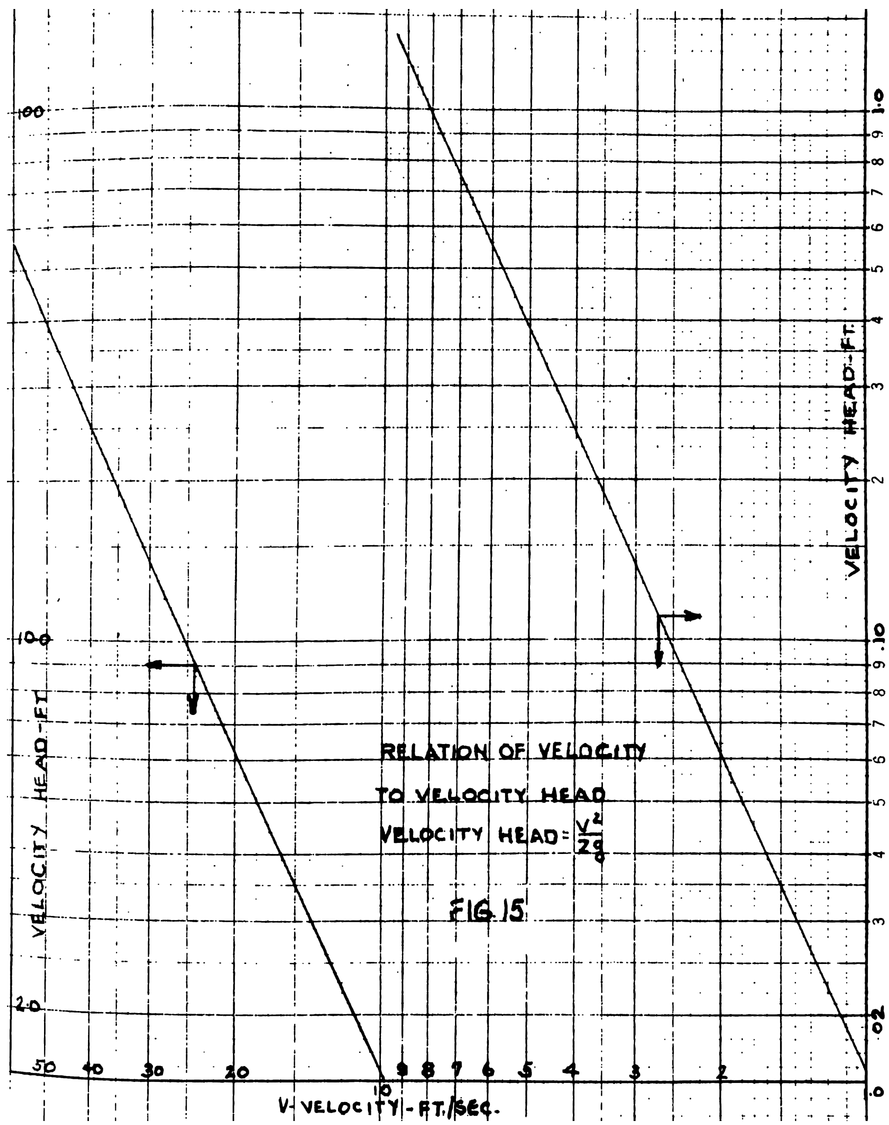



FIG. 14

MEAN RELATIVE DENSITY OF A MIXTURE IN A SERIES CIRCUIT WITH UNIFORM HEAT ABSORPTION RATES.

$$\bar{\rho} = \frac{1}{X_n} \log_e \left[ 1 + \frac{X_n}{1 + \sum_1^n X_{n-1}} \right]$$


RELATION OF VELOCITY
TO VELOCITY HEAD
VELOCITY HEAD = $\frac{V^2}{2g}$
FIG 15
VELOCITY HEAD - FT.
V - VELOCITY - FT./SEC.

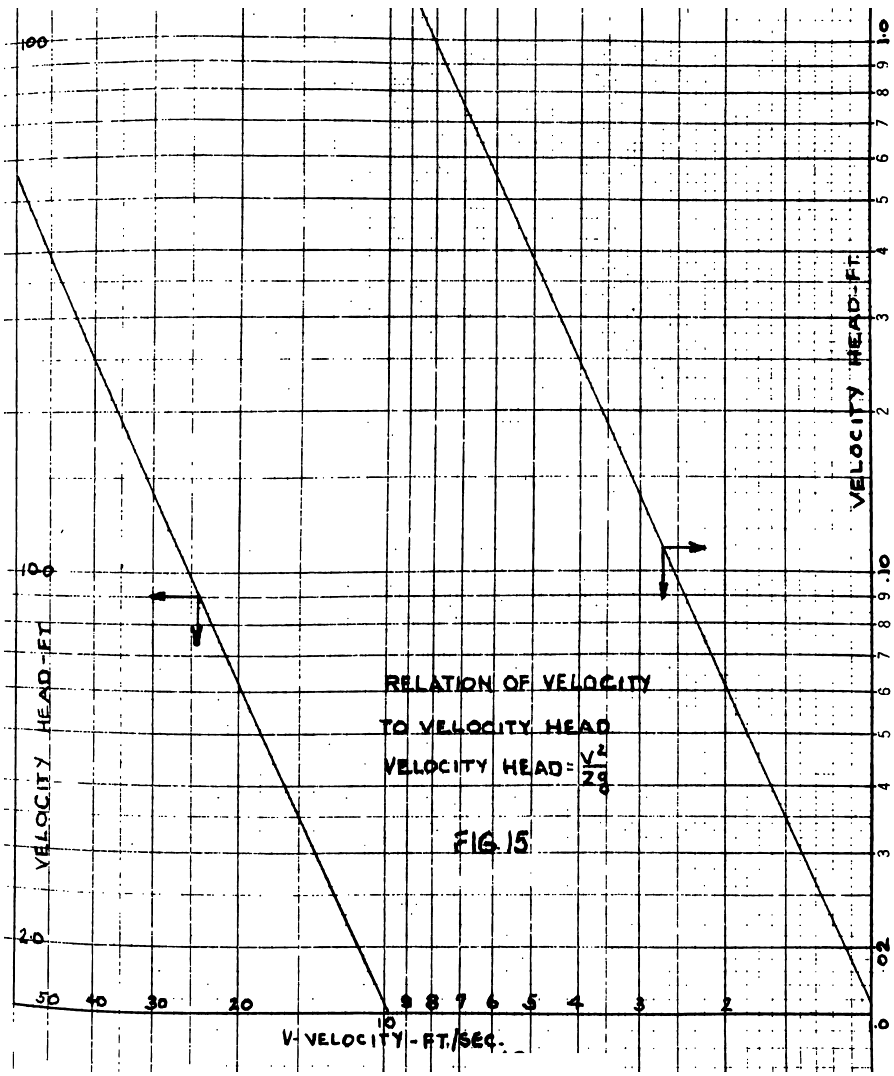

100
10.0
2.0
50
40
30
20
10
9
8
7
6
5
4
3
2
V- VELOCITY - FT./SEC.
VELOCITY HEAD - FT.
VELOCITY HEAD - FT.
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.10
.02
.0
RELATION OF VELOCITY
TO VELOCITY HEAD
VELOCITY HEAD = $\frac{V^2}{2g}$

FIG. 15

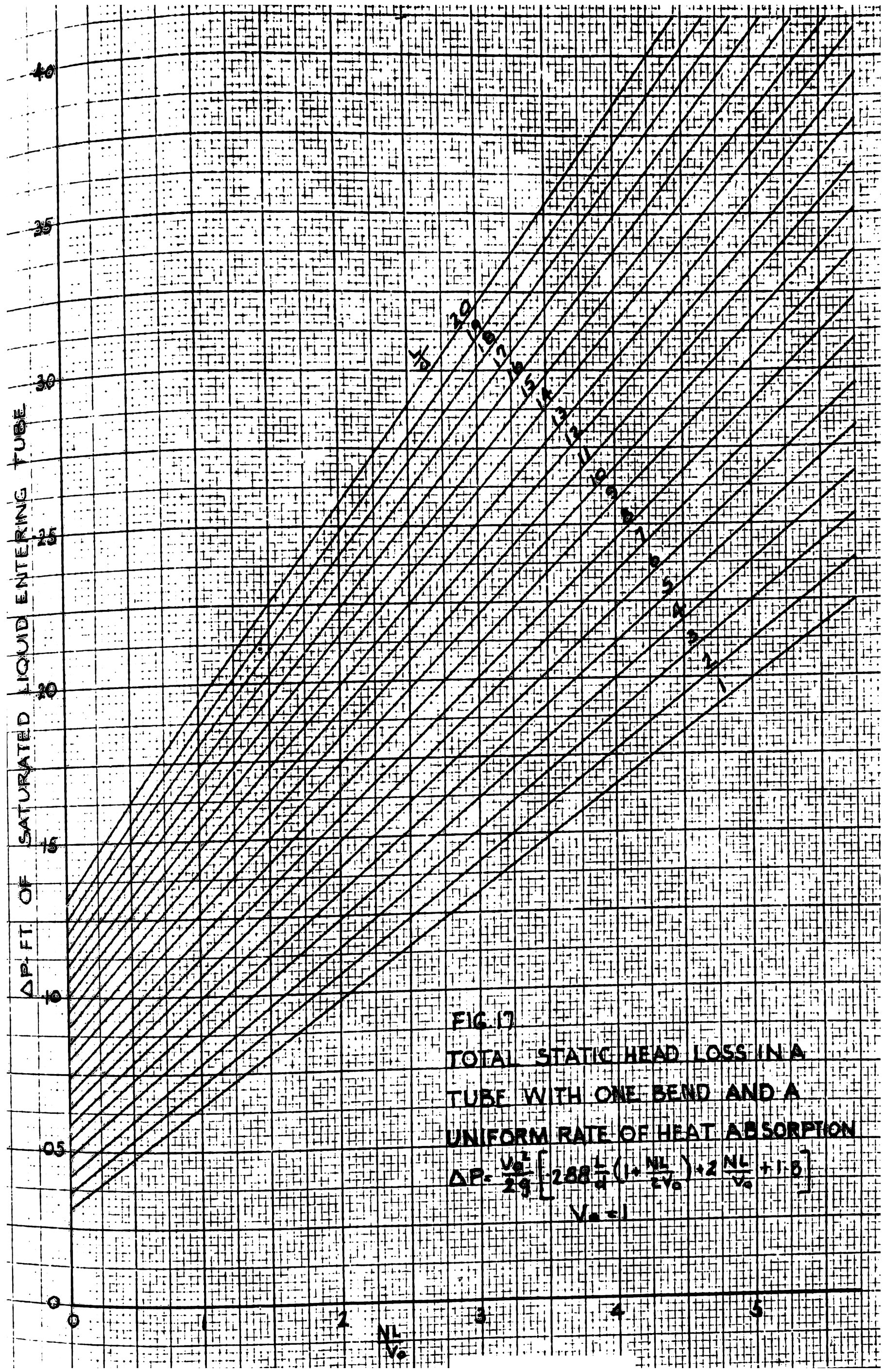



FIG. 17
TOTAL STATIC HEAD LOSS IN A TUBE WITH ONE BEND AND A UNIFORM RATE OF HEAT ABSORPTION

$$\Delta P = \frac{V_o^2}{2g}\left[.288\frac{L}{d}\left(1+\frac{NL}{2V_o}\right)+2\frac{NL}{V_o}+1.8\right]$$

$V_o = 1$

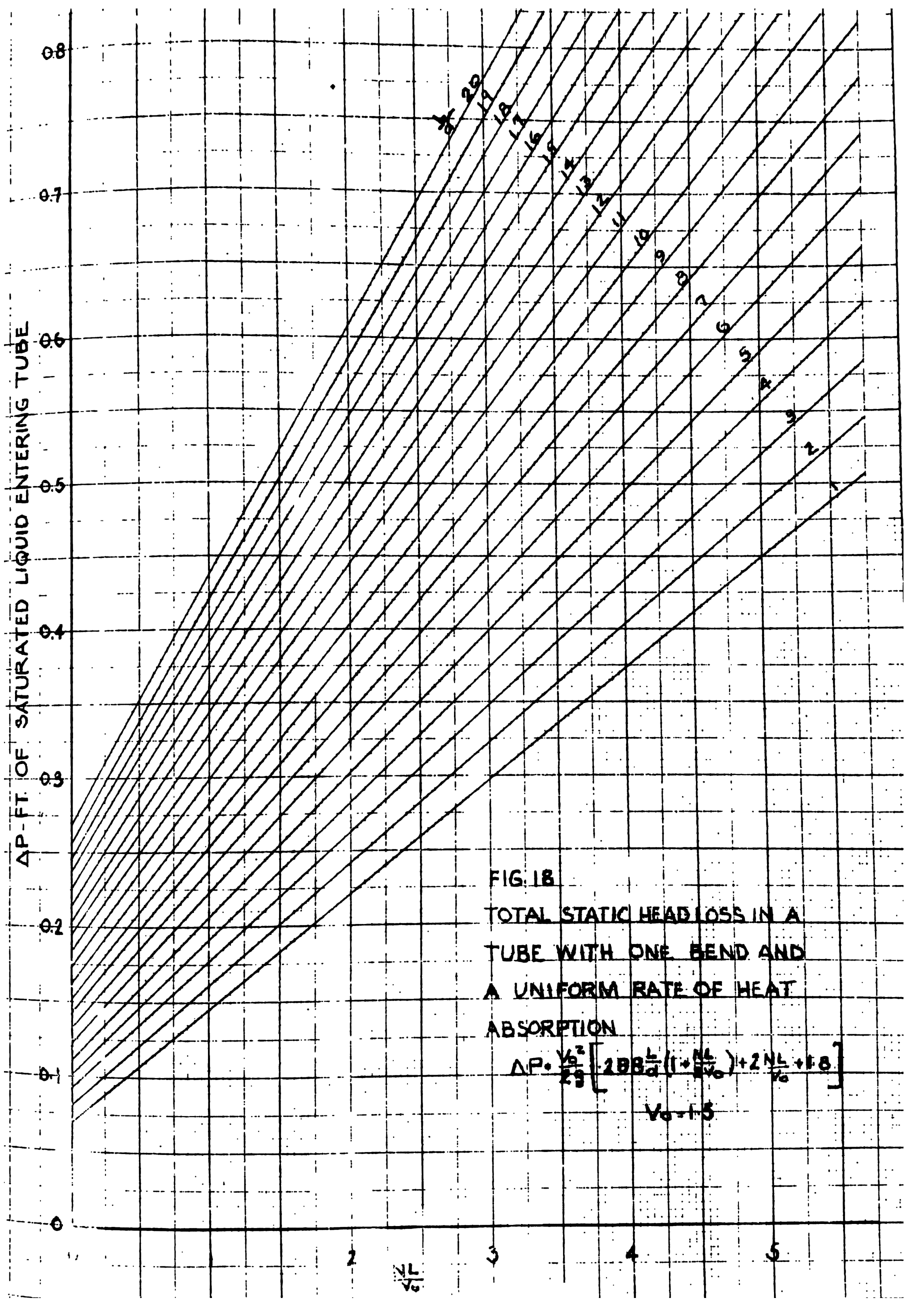



FIG. 18
TOTAL STATIC HEAD LOSS IN A TUBE WITH ONE BEND AND A UNIFORM RATE OF HEAT ABSORPTION

$$\Delta P = \frac{V_o^2}{2g}\left[.188\frac{L}{d}\left(1+\frac{NL}{2V_o}\right)+2\frac{NL}{V_o}+1.8\right]$$

$$V_o = 1.5$$

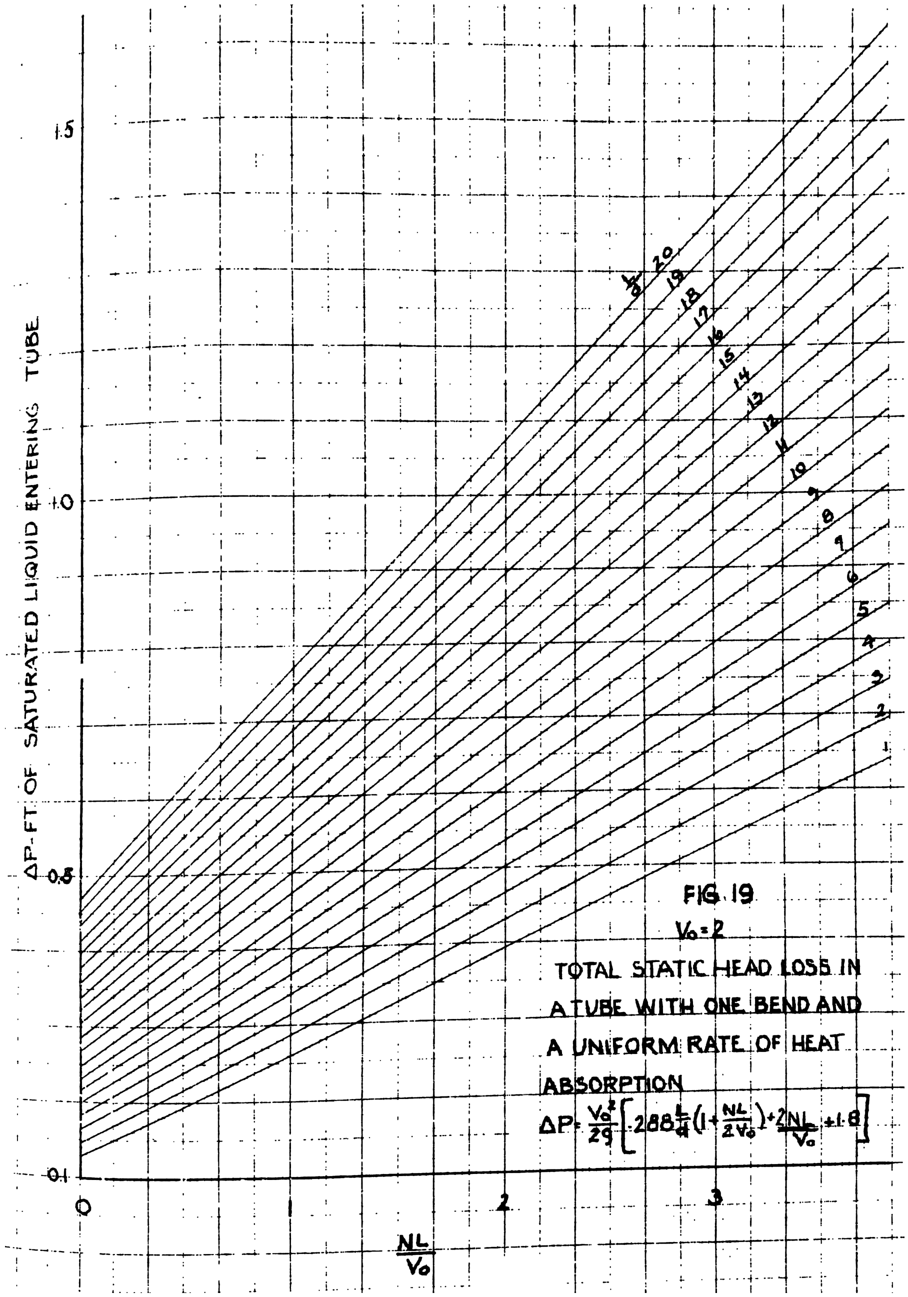
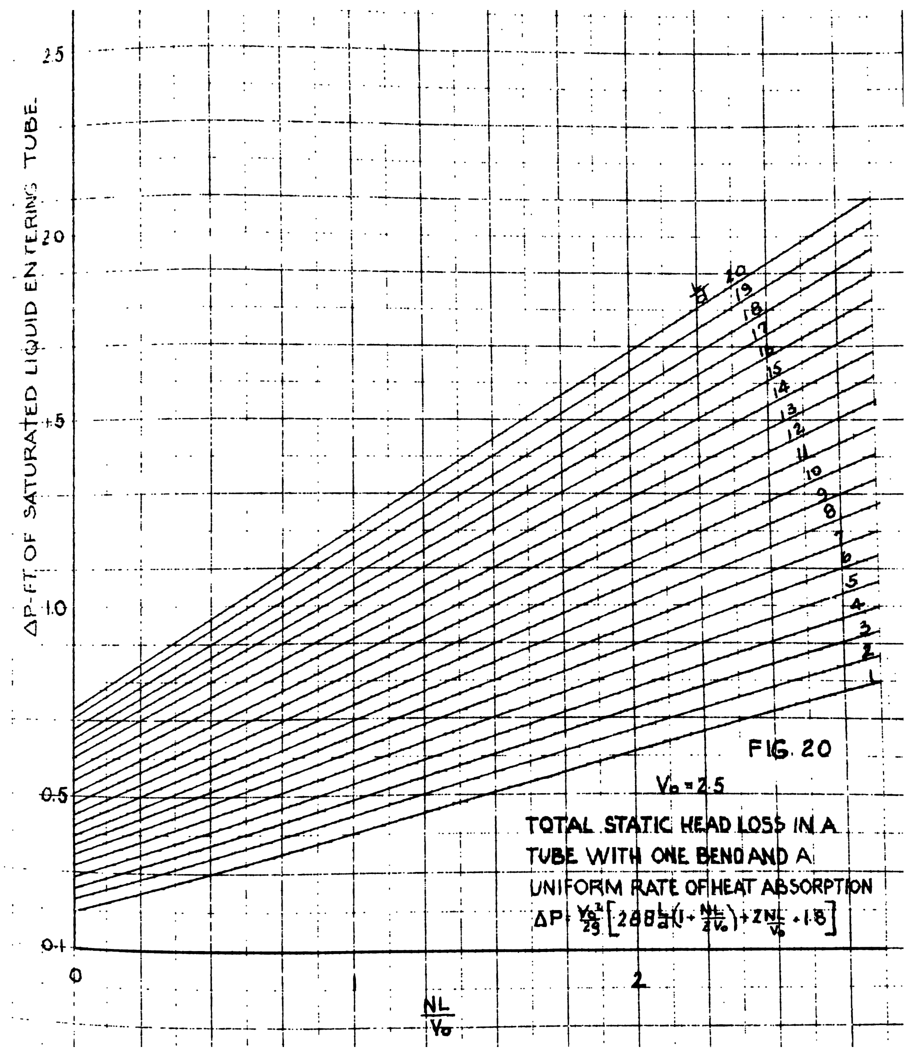



FIG. 19

$V_o = 2$

TOTAL STATIC HEAD LOSS IN A TUBE WITH ONE BEND AND A UNIFORM RATE OF HEAT ABSORPTION

$$\Delta P = \frac{V_o^2}{2g}\left[.288\frac{L}{d}\left(1+\frac{NL}{2V_o}\right)+\frac{2NL}{V_o}+1.8\right]$$


ΔP-FT. OF SATURATED LIQUID ENTERING TUBE.
2.5
2.0
1.5
1.0
0.5
0.1
0
1
2
NL/Vo
FIG. 20
Vo = 2.5
TOTAL STATIC HEAD LOSS IN A TUBE WITH ONE BEND AND A UNIFORM RATE OF HEAT ABSORPTION
ΔP = Vo²/2g [288 L/d (1 + NL/2Vo) + 2NL/Vo + 1.8]

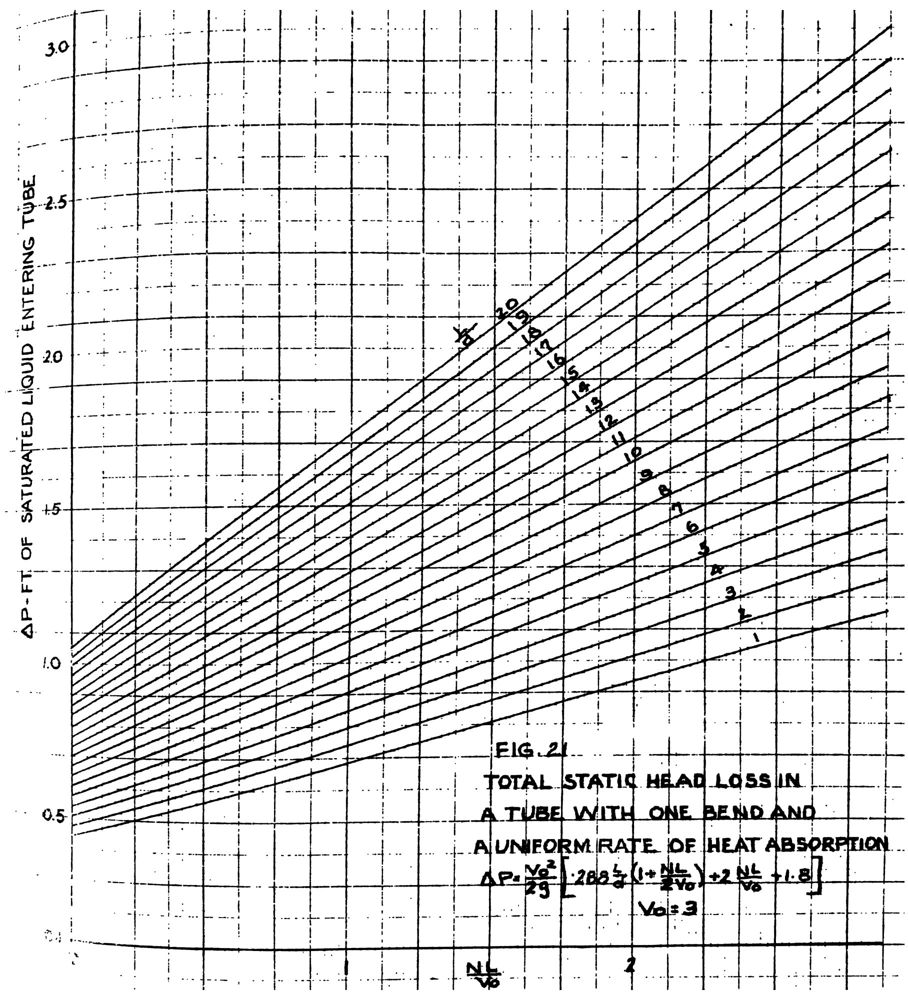



FIG. 21
TOTAL STATIC HEAD LOSS IN
A TUBE WITH ONE BEND AND
A UNIFORM RATE OF HEAT ABSORPTION

$$\Delta P = \frac{V_o^2}{2g}\left[.288\frac{L}{d}\left(1+\frac{NL}{2V_o}\right)+2\frac{NL}{V_o}+1.8\right]$$

$V_o = 3$

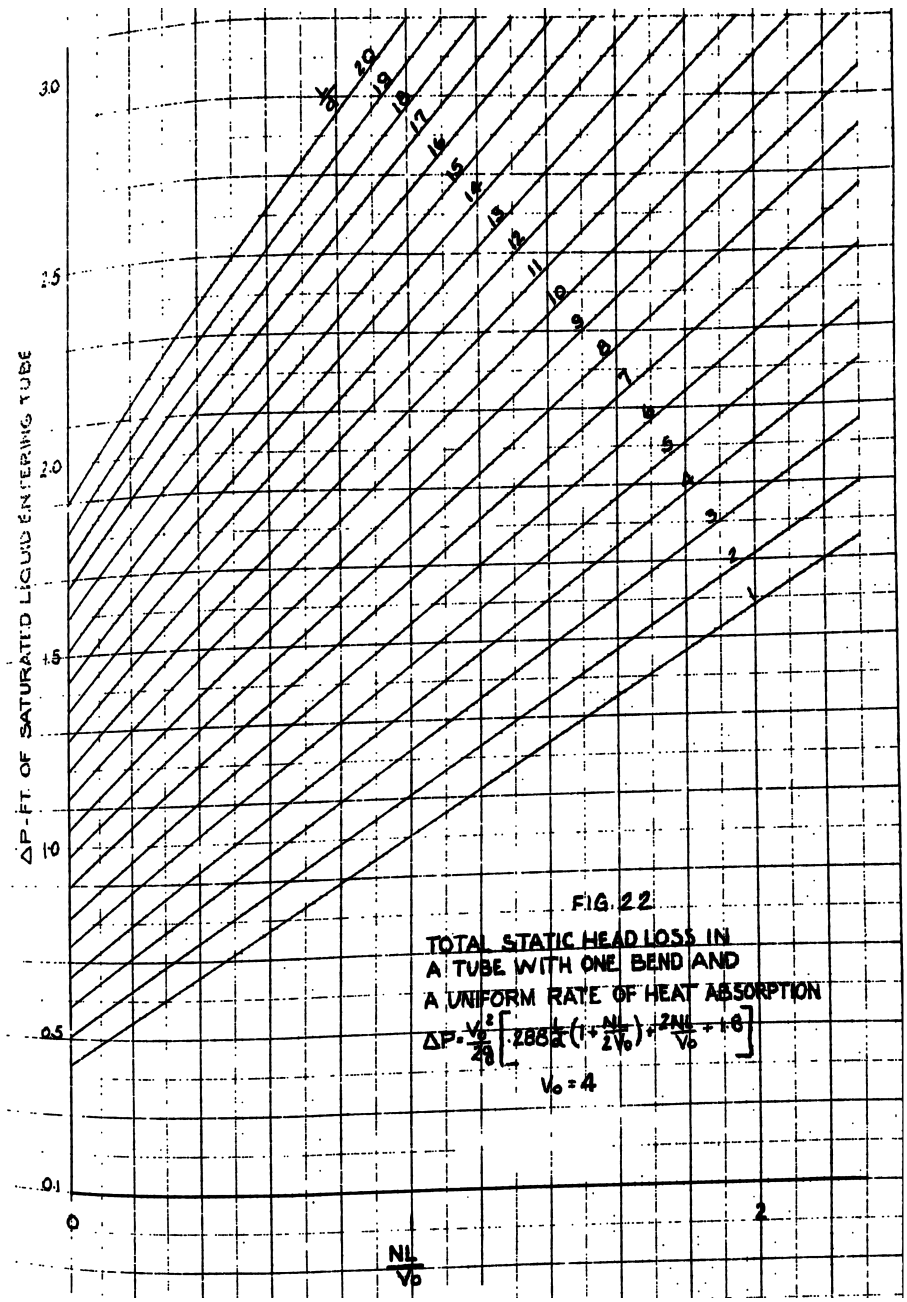



FIG. 22

TOTAL STATIC HEAD LOSS IN A TUBE WITH ONE BEND AND A UNIFORM RATE OF HEAT ABSORPTION

$$\Delta P = \frac{V_o^2}{2g}\left[.288\frac{L}{D}\left(1+\frac{NL}{2V_o}\right)+\frac{2NL}{V_o}+1.8\right]$$

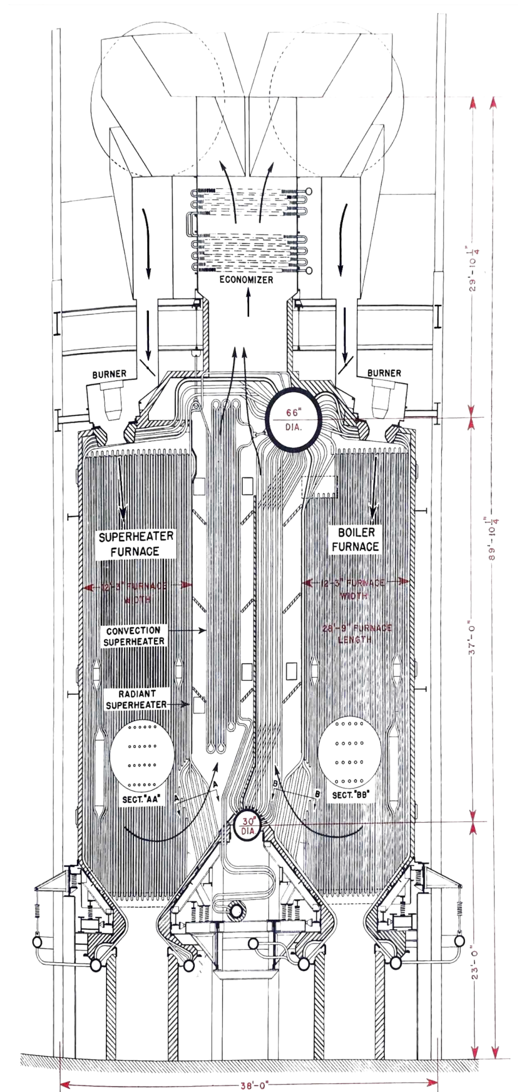
$V_o = 4$

# III *The Prediction of Circulation in a Large Boiler Installation*

The unit chosen for this section is the Twin Furnace Installation of the Duquesne Light Company's Frank R. Phillips station, Pittsburgh, Pa. This unit was chosen for two reasons: the computations involved are such that they bring out all of the problems likely to be encountered in predicting the circulation in any large unit, and secondly, through the courtesy and cooperation of the Foster W'heeler Corporation test data taken on this unit has been made available for comparison with predicted results.

The papers by John Blizard & A. C. Foster,[30] and Martin Frisch[58] give excellent descriptions of this type of steam generator and the reasons for its design. However, a brief description of this unit will be given along with all the pertinent data necessary for the computations.

Fig. 23 shows a vertical section taken looking to the rear of the installation. This illustration is used with the kind permission of the Foster Wheeler Corporation. The unit consists of a separately fired superheater furnace and a separately fired boiler furnace. The superheater furnace has both a radiant and convection superheater so arranged as to give almost constant final steam temperature over the greater part of the load range of the unit. In addition, both furnaces are designed to fire pulverized coal from the top, and the rate of firing is closely equal for both furnaces at full load. Final steam temperature is controlled by varying the rate of firing in both furnaces.




DUQUESNE LIGHT CO., FRANK R. PHILLIPS STATION, PITTSBURGH, PA.

PMD-624-J
SG-900-1
)-668

Table 2 gives the downcomer, feeder, and water wall data for this unit as given in Mr. R. A. Lorenzini's circulation report,[25] while Table 3 shows the performance data for a steam output of 387,500 lbs. *per* hr. Though the design maximum load is 500,000 lbs. *per* hr., the maximum load reported in the tests was 357,500 lbs. per hour, and the circulation characteristics will be predicted for this load. All the data in Table 3 are actual test data except those marked with an asterisk (*), which were calculated by the author from the test data, and the expected performance characteristics of this unit.

Fig. 24 is from Mr. Lorenzini's report and shows a plan view of the water wall feeders and also the location of the pressure taps used in the tests. During these tests, the density of the water flowing in the downcomers was measured utilizing the pressure drop method.[69] As far as could be determined within the limit of the accuracy of the instruments used, the water in the downcomers was essentially saturated and contained no entrained steam. For the calculations of this unit, it will therefore be assumed that the water in the downcomers is at saturation conditions and contains no steam.

This particular unit is fired from the top and the length of the flame is relatively short.[58] Therefore, it would be expected that the rate of heat absorption near the top of the tubes is very much greater than the rate of heat absorption near the bottom of the tubes. From Fig. 23, it is also apparent that the furnace gases can enter the screen bank without passing the tubes that constitute the ash hopper. The ash in this hopper section will also tend to further insulate the tubes in this section of the furnace. For the reasons just cited, the following assumptions regarding what will be considered as a reasonable distribution of heat absorption in this unit will be made:

1. The rate of heat absorption in the upper third of the heated area is twice that of the lower two-thirds area.

2. 50% of the total heat absorption occurs in the upper third of the heated area.

3. The length of the tube in the hopper section is essentially unheated, and heating is assumed to start at the top of the hopper line.

4. In all the heated wall sections, the heat absorption is uniform. As noted above, while the heat absorption is uniform in all the heated sections, the sections do not have equal rates of absorption.

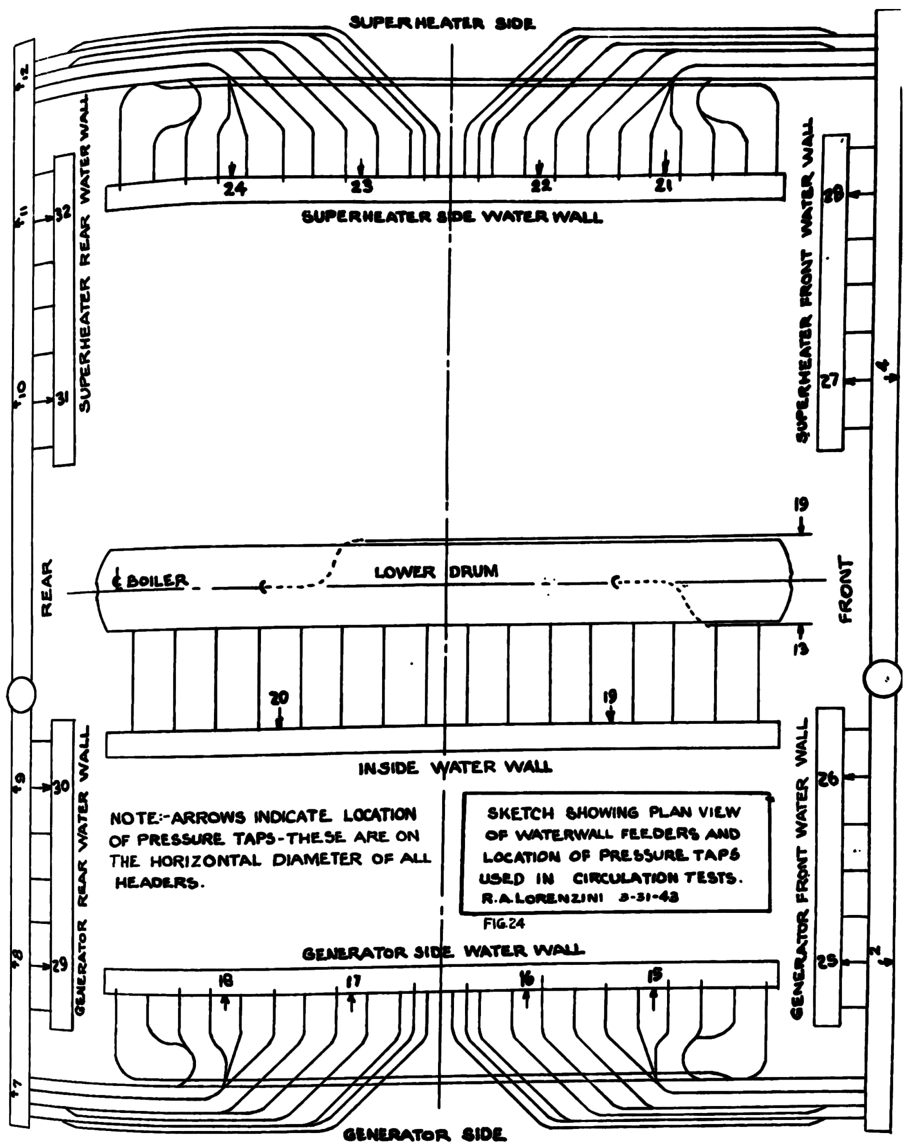

SUPERHEATER SIDE
SUPERHEATER SIDE WATER WALL
24
23
22
21
SUPERHEATER REAR WATER WALL
32
31
SUPERHEATER FRONT WATER WALL
28
27
REAR
FRONT
BOILER
LOWER DRUM
19
13
20
19
INSIDE WATER WALL
GENERATOR REAR WATER WALL
30
29
GENERATOR FRONT WATER WALL
26
25
NOTE:-ARROWS INDICATE LOCATION OF PRESSURE TAPS-THESE ARE ON THE HORIZONTAL DIAMETER OF ALL HEADERS.
SKETCH SHOWING PLAN VIEW OF WATERWALL FEEDERS AND LOCATION OF PRESSURE TAPS USED IN CIRCULATION TESTS.
R.A. LORENZINI 3-31-43
GENERATOR SIDE WATER WALL
18
17
16
15
GENERATOR SIDE

FIG.24

**Table 2: Downcomer, Feeder, and Wall Data**

| **Designation** | **Number** | **O.D. In.** | **I.D. In.** | **Total Area Sq.Ft.** | **Ratio-Feeder To Wall Area** |
|---|---|---|---|---|---|
| GEN. FRONT | 45 | 3 | 2.52 | 1.558 | .1556 |
| SUP. FRONT | 45 | 3 | 2.52 | 1.558 | .1556 |
| GEN REAR | 45 | 3 | 2.52 | 1.558 | .1556 |
| SUP. REAR | 103 | 3 | 2.52 | 1.558 | .1556 |
| GEN. SIDE | 103 | 3 | 2.52 | 3.566 | .2333 |
| INSIDE WALL | 103 | 3 | 2.52 | 3.566 | .1553 |
| BOILER BANK | 402 | 2 | 1.70 | 6.337 | — |
| **Location** | | | | | |
| Front | 1 | 16 | 13.5 | .994 | |
| Rear | 1 | 16 | 13.5 | .994 | |
| Inside | 2 | 10.75 | 8.75 | .835 | |
| FRONT | 14 | 3 | 2.52 | .485 | |
| REAR | 14 | 3 | 2.52 | .485 | |
| RIGHT SIDE | 24 | 3 | .832 | | |
| LEFT SIDE | 24 | 3 | 2.52 | .832 | |
| INSIDE | 16 | 3 | 2.52 | .554 | |

**Table 3: Circulation Test (3-31-43)**

| | |
|---|---|
| Test | 1 |
| DATE | 3-15-43 |
| TIME | 3:00 PM |
| CORRECTED STEAM FLOW LBS./HR. | 387,500 |
| DRUM PRESSURE psig | 928 |
| PRESSURE-SUPERHEATER OUTLET psig | 900 |
| FINAL STEAM TEMPERATURE F. | 894 |
| TEMP. WATER ENT. ECON. F. | 394 |
| TEMP. WATER LVG. ECON. F. | 488 |
| GAS TEMP. LVG. BOILER | 982 |
| *FUEL BURNED LBS/HR | 37,600 |
| *WET GAS AT BOILER EXIT LBS/HR | 462,000 |
| *TEMP. AIR ENT. UNIT F. | 80 |
| *TEMP. AIR GAS LVG. AIR HEATER F. | 505 |
| *TEMP. GAS LVG. FURNACE F. | 1880 |
| *FUEL | Bituminous—12,900 Btu/lb. as fired |
| #TOTAL FURNACE VOLUME CU.FT. | 27,000 |

From the data given in Table 3, the total heat absorbed in both furnaces can readily be determined.

Heat content in gases as fired in furnace: 1093 Btu/lb. of gas

Heat content in gases leaving furnace at 1880°F: 515 Btu/lb. of gas

Heat absorbed in furnace: 578 Btu/lb .of gas

Total furnace heat absorption: $462,000 \times 578 = 267,000,000$Btu/hr.

The projected area of each wall is given below for both furnaces:

| | | |
|---|---|---|
| Generating side wall | $\frac{1}{4} \times 103 \times 37.3$ | $= 961$ |
| Generating front and rear walls | $2 \times \frac{1}{4} \times 4.5 \times 387$ | $= 871$ |
| Superheater side wall | $\frac{1}{4} \times 103 \times 37.3$ | $= 961$ |
| Superheater front and rear walls | $2 \times \frac{1}{4} \times 45 \times 38.7$ | $= 871$ |
| Inside wall | $\frac{1}{4} \times 103 \times 39.25$ | $= 1012$ |
| Radiant wall (assumed same as inside wall) | | $= 5688$sq.ft. |
| absorption *per* sq.ft. of total wall area | $q = \frac{267,000,000}{\pi \times 5688}$ | $= 14,950$ |

$D = 3.00$ $\quad N = \frac{Dq(S-s)}{75d^2h}$

$q = 14,950$ $\quad N = \frac{3.00 \times 14,950 \times (.4543)}{75 \times (2.52)^2 \times 660.5}$

$S = .4757$ $\quad N = 0.646$

$s = .0214$

$h = 660.5$ $d = 2.52$

From Fig.11, at $\frac{D}{d^2}$ .473,and a pressure of 942.7psis,

$$N = 0.43 \text{ for a } q \text{ of } 10{,}000$$

$$N = 1.495 \times 0.43 \approx 0.643$$

It is apparent that using the curves of Fig. 11 eliminates the necessity of looking up the necessary thermodynamic properties and interpolating in the Steam Tables. In this case, the error incurred by using this graph is on the order of 0.46.

For the upper heated sections, on the basis of the heat absorption distribution made:

$$N = 1.5 \times 0.646 \approx 0.97$$

and for the lower heated section:

$$N = \frac{1}{2} \times 0.97 = 0.485$$

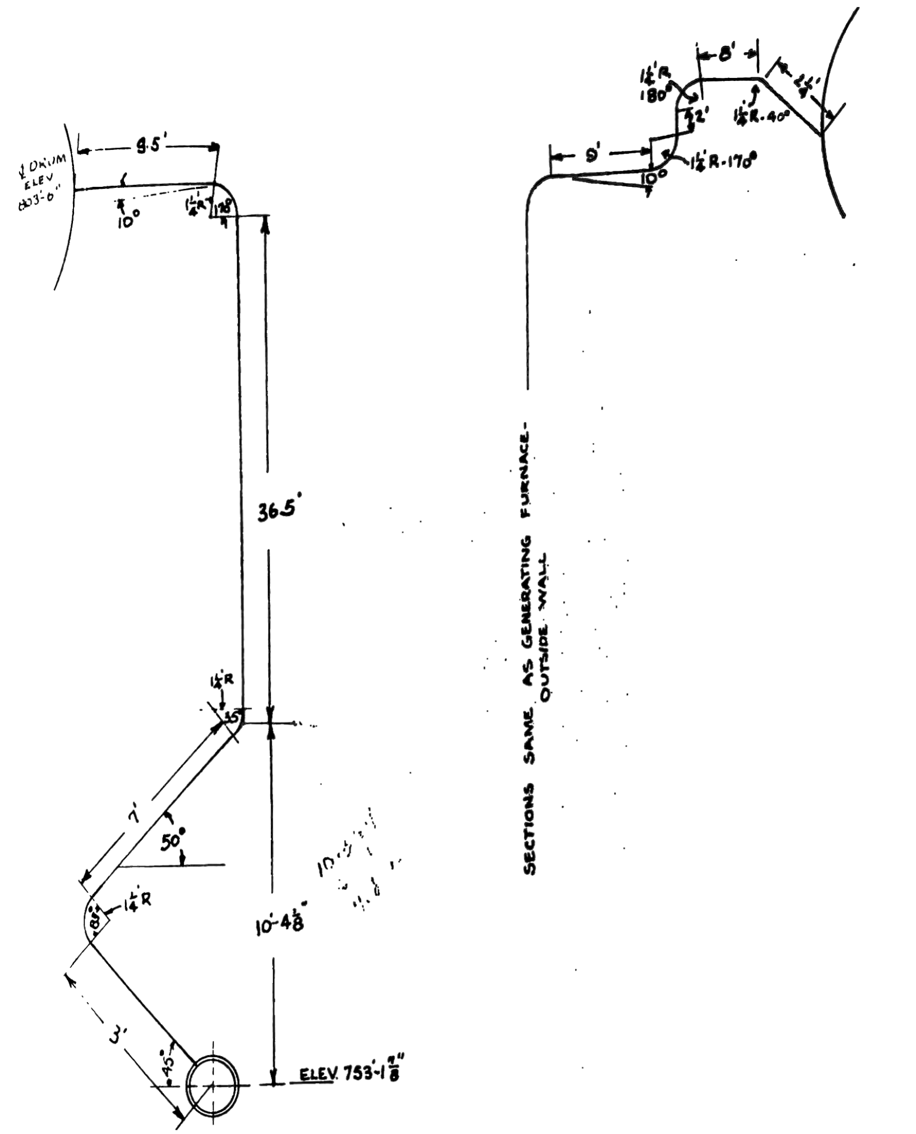
On the following pages, a sketch of each wall being computed is shown and the necessary computations are made using the equations and also the curves developed in Section II of this work. These computations are mostly self-explanatory and are readily followed by referring to the typical wall sketches and to the equations for the effect being computed. It must be fully understood that these computations are subject to all the inherent assumptions made in the derivation of the equations in Section II. Therefore, these computations were done on a slide rule and it is important to understand that further accuracy is not justified.

**Calculation of Entrance & Exit, Bend and Friction Loss Coefficients in Feeders**

| Feeder Number | Bends Deg. | Bend Loss Coeff | Total | Dev length Ft. |
|---|---|---|---|---|
| 450-451 | 45-90-90-12.5-12.5 | .21-.3-.3-.075-.075 | 1.0 | 35.07 |
| 452-453 | 90-180-90 | .3-.4-.3 | 1.0 | 16.02 |
| 454-455 | 45-135-45-45 | .21-.36-.21-.21 | 1.0 | 13.87 |
| 456-457 | 90-12.5-12.5-90-45 | .3-.075-.075-.3-.21 | 1.0 | 15.30 |
| 458-459 | 90-60-30 | .3-.24-.15 | 0.7 | 15.34 |
| 460-461 | 45-45-45-45 | .21-.21-.21-.21 | 0.8 | 15.32 |
| 462-463 | 90-45-45 | .3-.21-.21 | 0.7 | 17.73 |
| 464-465 | 45-45-90-45 | .21-.21-.3-.21 | 0.9 | 19.17 |
| 466-467 | 45-12.5-12.5-45-45 | .21-.075-.075-.21-.21 | 0.8 | 19.37 |
| 468-469 | 90-45-45 | .3-.21-.21 | 0.7 | 22.01 |
| 470-471 | 45-45-90-45 | .21-.21-.3-.21 | 0.9 | 22.64 |
| 472-473 | 12.5-12.5-45-45-45 | .075-.075-.21-.21-.21 | 0.8 | 22.03 |
| 474 | 90 | .4 | 0.4 | 4.48 |
| 475 | 90-70-50 | .3-.26-.23 | 0.8 | 15.5 |

| Feeder Number | Ent. Loss Coeff. | Exit Loss Coeff. | Bend Loss Coeff. | Frict. Loss Coeff. | Total Loss Coeff. |
|---|---|---|---|---|---|
| 450-451 | .5 | 1.0 | 1.0 | 3.34 | 5.84 |
| 452-453 | .5 | 1.0 | 1.0 | 1.53 | 4.03 |
| 454-455 | .5 | 1.0 | 1.0 | 1.32 | 3.82 |
| 456-457 | .5 | 1.0 | 1.0 | 1.46 | 3.96 |
| 458-459 | .5 | 1.0 | 0.7 | 1.46 | 3.66 |
| 460-461 | .5 | 1.0 | 0.8 | 1.46 | 3.76 |
| 462-463 | .5 | 1.0 | 0.7 | 1.69 | 3.89 |
| 464-465 | .5 | 1.0 | 0.9 | 1.83 | 4.23 |
| 466-467 | .5 | 1.0 | 0.8 | 1.85 | 4.15 |
| 468-469 | .5 | 1.0 | 0.7 | 2.10 | 4.30 |
| 470-471 | .5 | 1.0 | 0.9 | 2.26 | 4.66 |
| 472-473 | .5 | 1.0 | 0.8 | 2.10 | 4.40 |
| 450-473 | (Side Wall Feeders) | | Average Total Loss Coeff. | | 4.23 |
| 474 | (Front & Side Wall Feeders) | | Average Total Loss Coeff. | | 2.71 |
| 475 | (Inside Wall Feeders) | | Average Total Loss Coeff. | | 3.78 |

The calculations above refer to the un-numbered feeders shown in Fig. 24.




TYPICAL WALL TUBE
GENERATING FURNACE-OUTSIDE WALL
(NOT TO SCALE)

TYPICAL WALL TUBE
SUPERHEATER FURNACE-OUTSIDE WALL
(NOT TO SCALE)

FIG. 25

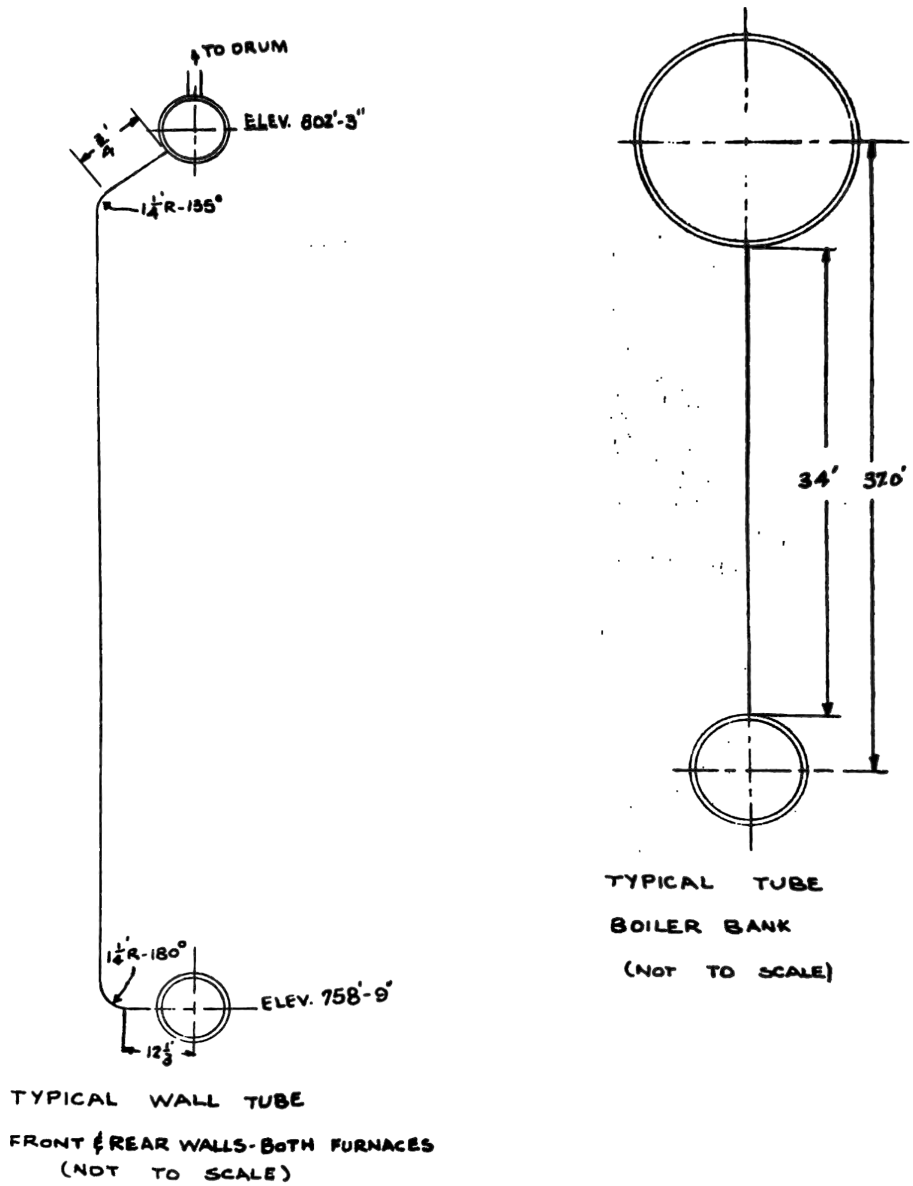
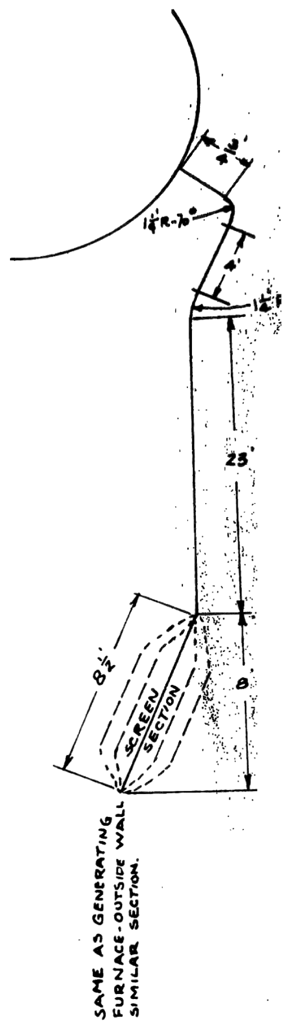

TO DRUM
ELEV. 802'-3"
1¼'R-135°
1¼'R-180°
ELEV. 758'-9"
TYPICAL WALL TUBE
FRONT & REAR WALLS-BOTH FURNACES
(NOT TO SCALE)
34'
TYPICAL TUBE
BOILER BANK
(NOT TO SCALE)

FIG. 26




TYPICAL WALL TUBE
GENERATING FURNACE - INSIDE WALL
(NOT TO SCALE)

FIG. 27

**Generating Furnace: Outside Wall**

*(Refer to Fig. 25)*

**Gravity Head:**

**Circuit 1** 10.34

**Circuit 2** $\frac{L_n}{X_n}\log_e(1+X_n)$

| $V_0$ | $X_n$ | $1+X_n$ | Gravity Head |
|---|---|---|---|
| 0.5 | 2.42 | 3.42 | 12.70 |
| 1.0 | 1.21 | 2.21 | 16.40 |
| 1.5 | 0.806 | 1.806 | 18.30 |
| 2.0 | 0.605 | 1.605 | 19.51 |

**Circuit 3** $\frac{L_n}{X_n}\log_e\left[1+\frac{X_n}{1+\sum_1^n X_{n-1}}\right]$

| $V_0$ | $\sum_1^n X_{n-1}$ | $1+\sum_1^n X_{n-1}$ | $X_n$ | $\frac{X_n}{1+\sum_1^n X_{n-1}}$ | $\frac{X_n}{1+\frac{X_n}{1+\sum_1^n X_{n-1}}}$ | Gravity Head |
|---|---|---|---|---|---|---|
| 0.5 | 2.42 | 3.42 | 2.230 | 0.653 | 1.653 | 2.59 |
| 1.0 | 1.21 | 2.21 | 1.115 | 0.504 | 1.504 | 4.20 |
| 1.5 | 0.806 | 1.806 | 0.743 | 0.412 | 1.412 | 5.34 |
| 2.0 | 0.605 | 1.605 | 0.557 | 0.347 | 1.347 | 6.14 |

**Circuit 4** $\frac{L_n \sin\theta}{1+\sum_1^n X_{n-1}}$

| $V_0$ | $\sum_n^1 X_{n-1}$ | $1+\sum_1^n X_{n-1}$ | Gravity Head |
|---|---|---|---|
| 0.5 | 4.65 | 5.65 | 0.62 |
| 1.0 | 2.325 | 3.325 | 1.05 |
| 1.5 | 1.55 | 2.55 | 1.37 |
| 2.0 | 1.162 | 2.162 | 1.62 |

**Losses: Acceleration**

**Circuit 1** 0

**Circuit 2** $\frac{V_o^2}{2g}\times 2X_n$

**Generating Furnace: Outside Wall (cont.)**

| $V_0$ | $\frac{V_0^2}{2g}$ | $\frac{V_0^2}{2g}$ | $X_n$ | $2X_n$ | Acceleration Loss |
|---|---|---|---|---|---|
| 0.5 | 0.25 | 0.0039 | 2.42 | 4.84 | 0.019 |
| 1.0 | 1.00 | 0.0155 | 1.21 | 2.42 | 0.038 |
| 1.5 | 2.25 | 0.035 | 0.806 | 1.612 | 0.056 |
| 2.0 | 4.00 | 0.062 | 0.605 | 1.21 | 0.075 |

**Circuit 3** $\frac{V_o^2}{2g} \times 2X_n$

| $V_0$ | $\frac{V_0^2}{2g}$ | $\frac{V_0^2}{2g}$ | $X_n$ | $2X_n$ | Acceleration Loss |
|---|---|---|---|---|---|
| 0.5 | 0.25 | 0.0039 | 2.23 | 4.46 | 0.017 |
| 1.0 | 1.00 | 0.0155 | 1.115 | 2.23 | 0.035 |
| 1.5 | 2.25 | 0.035 | 0.743 | 1.486 | 0.052 |
| 2.0 | 4.00 | 0.062 | 0.557 | 1.115 | 0.069 |

**Circuit 4** 0

**Entrance and Exit Loss** $1.5\frac{V_o^2}{2g}$

| $V_0$ | $\frac{V_0^2}{2g}$ | Entrance and Exit Loss |
|---|---|---|
| 0.5 | 0.0039 | 0.006 |
| 1.0 | 0.0155 | 0.023 |
| 1.5 | 0.035 | 0.053 |
| 2.0 | 0.062 | 0.093 |

**Bend Loss**

**Circuit 1-2 Bends**

$$K_{B_1} = 0.3$$

$$K_{B_2} = 0.18$$

$$\sum K_B = 0.48$$

$$\text{Bend Loss} = K_B \frac{V_0^2}{2g}$$

**Generating Furnace: Outside Wall (cont.)**

| $V_o$ | $\frac{V_o^2}{2g}$ | Bend Loss |
|---|---|---|
| 0.5 | 0.0039 | 0.002 |
| 1.0 | 0.0155 | 0.008 |
| 1.5 | 0.035 | 0.017 |
| 2.0 | 0.062 | 0.030 |

| | |
|---|---|
| **Circuit 2** | 0 |
| **Circuit 3** | 0 |
| **Circuit 4** | 1 bend |

$$K_B = 0.39\ ;\ K_B \frac{V_o^2}{2g}\left[1 + \sum_1^n X_{n-1}\right]$$

| $V_o$ | $\frac{V_o^L}{2g}$ | $\sum_1^n X_{n-1}$ | $1 + \sum_1^n X_{n-1}$ | Bend Loss |
|---|---|---|---|---|
| 0.5 | 0.0039 | 4.65 | 5.65 | 0.009 |
| 1.0 | 0.0155 | 2.325 | 3.325 | 0.020 |
| 1.5 | 0.035 | 1.55 | 2.55 | 0.035 |
| 2.0 | 0.062 | 1.162 | 2.162 | 0.052 |

**Friction Loss**

**Circuit 1** $\frac{.288L}{d}\frac{V_o^2}{2g}$

| $V_o$ | $\frac{V_o^2}{2g}$ | $.288\frac{V_o^2}{2g}$ | $\frac{L}{D}$ | Friction Loss |
|---|---|---|---|---|
| 0.5 | 0.0039 | 0.001125 | 5.02 | 0.006 |
| 1.0 | 0.0155 | 0.00447 | 5.02 | 0.022 |
| 1.5 | 0.035 | 0.0101 | 5.02 | 0.050 |
| 2.0 | 0.062 | 0.01785 | 5.02 | 0.090 |

**Circuit 2** $.288\frac{L}{d}\frac{V_o^2}{2g}\left[1 + \frac{X_n}{2}\right]$

**Superheater Furnace: Outside Wall**

*Refer to Fig. 25*

All values of Gravity Head and Losses for this wall are essentially the same as in corresponding sections of the **Generating Furnace: Outside Wall** with the exception of the bend loss and friction loss in the last section.

**Bend Loss** **Circuit 4** 4 bends

$$K_{B_1} = 0.39$$
$$K_{B_2} = 0.39$$
$$K_{B_3} = 0.39$$
$$K_{B_4} = 0.20$$

$$\sum K_B = 1.37$$

$$K_B \frac{V_o^2}{2g}\left[1 + \sum_1^n X_{n-1}\right]$$

| $V_o$ | $\frac{V_o^2}{2g}$ | $\sum_1^n X_{n-1}$ | $1 + \sum_1^n X_{n-1}$ | Bend Loss |
|---|---|---|---|---|
| 0.5 | 0.0039 | 4.65 | 5.65 | 0.030 |
| 1.0 | 0.0155 | 2.325 | 3.325 | 0.071 |
| 1.5 | 0.035 | 1.55 | 2.55 | 0.122 |
| 2.0 | 0.062 | 1.162 | 2.162 | 0.84 |

**Friction Loss**

**Circuit 4** $.288 \frac{V_o^2}{2g} [1 + \sum_1^n X_{n-1}]$

| $V_o$ | $.288 \frac{V_o^2}{2g}$ | $\frac{L}{d}$ | $1 + \sum_1^n X_{n-1}$ | Friction Loss |
|---|---|---|---|---|
| 0.5 | 0.001125 | 11.7 | 5.65 | 0.075 |
| 1.0 | 0.00447 | 11.7 | 3.325 | 0.174 |
| 1.5 | 0.0101 | 11.7 | 2.55 | 0.302 |
| 2.0 | 0.01785 | 11.7 | 2.162 | 0.453 |

**Front and Rear Walls: Both Furnaces**

*Refer to Fig. 26*

**Gravity Head**

| | |
|---|---|
| **Circuit 1** | 4.75 |
| **Circuit 2** | Essentially the same as corresponding section **Generating Furnace: Outside Wall** |
| **Circuit 3** | $\frac{L_n}{X_n} \log_e \left[1 + \frac{X_n}{1+\sum_1^n X_n \ldots}\right]$ |

| $V_o$ | $\sum_1^n X_{n-1}$ | $X_n$ | $1 + \sum_1^n X_{n-1}$ | Gravity head |
|---|---|---|---|---|
| 0.5 | 2.42 | 2.64 | 3.42 | 2.95 |
| 1.0 | 1.21 | 1.32 | 2.21 | 4.82 |
| 1.5 | 0.806 | 0.86 | 1.806 | 6.16 |
| 2.0 | 0.605 | 0.66 | 1.605 | 7.09 |

**Entrance and Exit Loss** $1.5\frac{V_o^2}{2g}$

| $V_o$ | $\frac{V_o^2}{2g}$ | Entrance & Exit Loss |
|---|---|---|
| 0.5 | .0039 | .006 |
| 1.0 | 0.0155 | .023 |
| 1.5 | .035 | .053 |
| 2.0 | .062 | .093 |

**Acceleration Loss**

| | |
|---|---|
| **Circuit 1** | 0 |
| **Circuit 2** | Essentially the same as corresponding section **Generating Furnace: Outside Wall** |
| **Circuit 3** | $\frac{V_0^2}{2g} \times 2X_n$ |

| $V_o$ | $\frac{V_o^2}{2g}$ | $X_n$ | $2X_n$ | Acceleration Loss |
|---|---|---|---|---|
| 0.5 | .0039 | 2.64 | 5.28 | .021 |
| 1.0 | .0155 | 1.32 | 2.64 | .041 |
| 1.5 | .035 | 0.86 | 1.72 | .060 |
| 2.0 | .062 | 0.66 | 1.32 | .082 |

**Front and Rear Walls: Both Furnaces (cont.)**

**Bend Loss**

**Circuit 1** 1 bend

$$K_B = .39$$

$$K_B \frac{V_o^2}{2g}$$

| $V_o$ | $\frac{V_o^2}{2g}$ | $K_B$ | Bend Loss |
|---|---|---|---|
| 0.5 | .0039 | .39 | .002 |
| 1.0 | .0155 | .39 | .006 |
| 1.5 | .035 | .39 | .014 |
| 2.0 | .062 | .39 | .24 |

**Circuit 2** —0 **Circuit 3**

$$K_B = .30$$
$$K_B \frac{V_o^2}{2g} [1 + \textstyle\sum_1^n X_{n-1}]$$

| $V_o$ | $\frac{V_o^2}{2g}$ | $K_B$ | $\sum_1^n$ | $1 + \sum_1^n X_{n-1}$ | Bend Loss |
|---|---|---|---|---|---|
| 0.5 | .0039 | .30 | 5.06 | 6.06 | .006 |
| 1.0 | .0155 | .30 | 2.53 | 3.53 | .016 |
| 1.5 | .035 | .30 | 1.666 | 2.666 | 0.28 |
| 2.0 | .062 | .30 | 1.265 | 2.265 | .042 |

**Friction Loss**

**Circuit 1** $.288 \frac{L}{d} \frac{V_o^2}{2g}$

| $V_o$ | $\frac{V_o^2}{2g}$ | $.288 \frac{V_o^2}{2g}$ | $\frac{L}{d}$ | Friction Loss |
|---|---|---|---|---|
| 0.5 | .0039 | .001125 | 3.30 | .004 |
| 1.0 | .0155 | .00447 | 3.30 | .015 |
| 1.5 | .035 | .0101 | 3.30 . | 033 |
| 2.0 | .035 | .0101 | 3.30 | .033 |
| 2.0 | .062 | .01785 | 3.30 | .059 |

| | |
|---|---|
| **Circuit 2** | Essentially the same as corresponding section **Generating Furnace: Outside Wall** |

**Front and Rear Walls: Both Furnaces (cont.)**

**Circuit 3** $.288 \frac{L}{d} \frac{V_o^2}{2g} \left[1 + \frac{X_n}{2} + \sum_1^n X_{n-1}\right]$

| $V_o$ | $\frac{V_o^2}{2g}$ | $.288 \frac{V_o^2}{2g}$ | $\frac{L}{d}$ | $\frac{X_n}{2}$ | $1+ \sum_1^n X_{n-1}$ | $1 + \frac{X_n}{2} + \sum_1^n X_{n-1}$ | Friction Loss |
|---|---|---|---|---|---|---|---|
| 0.5 | .0039 | .001125 | 4.83 | 1.32 | 3.42 | 4.74 | .026 |
| 1.0 | 0.155 | .00447 | 4.83 | 0.66 | 2.21 | 2.87 | .062 |
| 1.5 | .035 | .0101 | 4.83 | 0.44 | 1.806 | 2.246 | .110 |
| 2.0 | .062 | .01785 | 4.83 | 0.33 | 1.605 | 1.935 | .167 |

**Feeder Loss** $2.71 \frac{V_o^2}{2g} \times \left(\frac{1}{.1556}\right)^2 = 11.25 \frac{V_o^2}{2g}$

| $V_o$ | $\frac{V_o^2}{2g}$ | Feeder Loss |
|---|---|---|
| 0.5 | .0039 | .438 |
| 1.0 | 0.155 | 1.748 |
| 1.5 | .035 | 3.940 |
| 2.0 | 0.62 | 6.980 |

Each of these walls ends in a header and from this header feeders are taken to the drum. These are essentially unheated.

$$\text{Area of wall} = 1.558 \times 144 = 224\text{sq.in.}$$

$$\text{Area of feeders} = \frac{\pi}{4}(2.52)^2 \times 4 + \frac{\pi}{4}(3.44)^2 \times 7 = 85\text{sq.in.}$$

$$K^2 = \left(\frac{224}{85}\right)^2 = 6.95$$

**Superheater Furnace Walls**

**Entrance and Exit Loss** $1.5\frac{V_o^2}{2g} \times 6.95\,[1 + \sum^n +1X_{n-1}]$

| $V_o$ | $10.4\frac{V_o^2}{2g}$ | $1 + \sum_1^n X_{n-1}$ | Entrance and exit loss |
|---|---|---|---|
| 0.5 | .0406 | 6.06 | .246 |
| 1.0 | .1615 | 3.53 | .570 |
| 1.5 | .364 | 2.666 | .969 |
| 2.0 | .645 | 2.265 | 1.460 |

**Front and Rear Walls: Both Furnaces (cont.)**

**Gravity Head** $\frac{4.75}{1+\sum_1^n Xn-1}$

| $V_o$ | $1+\sum_1^n Xn-1$ | Gravity Head |
|---|---|---|
| 0.5 | 6.06 | .784 |
| 1.0 | 3.53 | 1.345 |
| 2.0 | 2.265 | 2.100 |

**Friction Loss** $.288\frac{L}{d} \times 6.95\frac{V_o^2}{2g}\left[1+\sum_1^n Xn-1\right]$

| $V_o$ | $6.95 \times .288\frac{V_o^L}{2g}$ | $\frac{L}{d}$ | $1+\sum_1^n Xn-1$ | Friction Loss |
|---|---|---|---|---|
| 0.5 | .00782 | 1.52 | 6.06 | 0.72 |
| 1.0 | .0311 | 1.52 | 3.53 | .167 |
| 1.5 | .0703 | 1.52 | 2.666 | .284 |
| 2.0 | .124 | 1.52 | 2.265 | .427 |

**Generating Furnace**

**Gravity Head** $\frac{3.00}{1+\sum_1^n Xn-1}$

| $V_o$ | | Gravity Head |
|---|---|---|
| 0.5 | 6.06 | .495 |
| 1.0 | 3.53 | .850 |
| 1.5 | 2.666 | 1.125 |
| 2.0 | 2.265 | 1.322 |

**Friction Loss** $.288\frac{L}{d} \times 6.95\frac{V_o^L}{2g}\left[1+\sum_1^n Xn-1\right]$

| $V_o$ | $6.95\times 2.88\frac{V_o^2}{2g}$ | $\frac{L}{d}$ | $1+\sum_1^n Xn-1$ | Friction Loss | $\sum$ Gravity Head $+\sum$ Losses: Generating Furnace | Superheater Furnace |
|---|---|---|---|---|---|---|
| 0.5 | .00782 | 0.96 | 6.06 | .045 | 21.73 | 22.05 |
| 1.0 | .0311 | 0.96 | 3.53 | .105 | 29.52 | 30.07 |
| 1.5 | .0703 | 0.96 | 2.666 | .180 | 35.92 | 36.68 |
| 2.0 | 124 | 0.96 | 2.265 | .270 | 41.16 | 43.09 |

### Boiler Bank and Inside Wall: Generating Furnace

The **Boiler Bank and Inside Wall: Generating Furnace** of this unit are more involved since both radiation and convection occur in these walls. The methods and experimental data used to compute the heat transfer in design are usually as given in McAdams[33] or other standard sources, but once the conditions of operation are predicted or known, the total heat absorbed in each section can be obtained as follows:

| | |
|---|---|
| **Enthalpy of steam at Superheater outlet** | $1447.7 \frac{\text{Btu}}{\text{lb.}}$ |
| **Enthalpy of saturated steam at drum conditions** | $1193.9 \frac{\text{Btu}}{lb.}$ |
| **Change in enthalpy** | $253.8 \frac{\text{Btu}}{lb.}$ |
| **Heat absorbed in Superheater** | $387,500 \times 253.8 = 98,000,000 \frac{\text{Btu}}{\text{hr.}}$ |
| **Change in Enthalpy in heating water leaving economizer** | $\begin{array}{rr} & 1447.7 \\ - & 473.9 \\ \hline & 973.8 \frac{\text{Btu}}{\text{lb.}} \end{array}$ |
| **Total water absorption** | $973.8 \times 422,200 = 410,000,000 \frac{\text{Btu}}{\text{hr.}}$ |
| **Heat absorbed in Boiler Bank** | $45,000,000 \frac{\text{Btu}}{\text{hr.}}$ |

The heat absorption in the **Inside Wall: Generating Furnace** due to radiation can be taken as essentially the same as in the rest of the furnace. The Boiler Bank is essentially a convection heating surface and as such can properly be treated by the equations developed for this case in Section II. However, since the rates of heat absorption are comparatively low, the equations for the case of uniform heat absorption can be applied with very little error.

### Inside Wall: Generating Furnace

*Refer to Fig. 27*

The values of $N$ are essentially the same as used in previous similar sections except for the screen section. Using the data in the McAdams,[33] the value of the overall coefficient of heat transfer for convection will be found to be close to 5.4, and the temperature drop due to convection in the screen section is 170°F. The logarithmic mean temperature is 1260°F. Therefore, the heat absorption due to convection is $6,600\frac{\text{Btu}}{\text{sq.ft.hr}}$ based on total outside surface, and $N$ for this section is .0779.

**Gravity Head**

**Circuit 1** 10.34

**Circuit 2** $\frac{L}{X_n}\log_e[1+X_n]\frac{8}{8.5}$

| $V_0$ | $X_n$ | $1+X_n$ | Gravity Head |
|---|---|---|---|
| 0.5 | 1.32 | 2.32 | 5.10 |
| 1.0 | 0.66 | 1.66 | 6.13 |
| 1.5 | 0.44 | 1.44 | 6.60 |
| 2.0 | 0.33 | 1.33 | 6.90 |

**Circuit 3** $\frac{L}{X_n}\log_e\left[1+\frac{X_n}{1+\sum_1^n X_{n-1}}\right]$

| $V_0$ | $\sum_1^n X_{n-1}$ | $1+\sum_1^n X_{n-1}$ | $X_n$ | $\frac{X_n}{1+\sum_1^n X_{n-1}}$ | $1+\frac{X_n}{1+\sum_1^n X_{n-1}}$ | Gravity Head |
|---|---|---|---|---|---|---|
| 0.5 | 1.32 | 2.32 | 1.68 | 0.725 | 1.725 | 5.62 |
| 1.0 | 0.66 | 1.66 | 0.84 | 0.506 | 1.506 | 8.45 |
| 1.5 | 0.44 | 1.44 | 0.56 | 0.389 | 1.389 | 10.15 |
| 2.0 | 0.33 | 1.33 | 0.42 | 0.316 | 1.316 | 11.32 |

**Circuit 4** $\frac{L_n}{X_n}\log_e\left[1+\frac{X_n}{1+\sum_1^n X_n}\right]$

| $V_0$ | $\sum_1^n X_{n-1}$ | $1+\sum_1^n X_{n-1}$ | $X_n$ | $\frac{X_n}{1+\sum X_{n-1}}$ | $1+\frac{X_n}{1+\sum X_{n-1}}$ | Gravity Head |
|---|---|---|---|---|---|---|
| 0.5 | 3.00 | 4.00 | 1.104 | 0.276 | 1.276 | 1.26 |
| 1.0 | 1.50 | 2.50 | 0.652 | 0.261 | 1.221 | 2.06 |
| 1.5 | 1.00 | 2.00 | 0.368 | 0.184 | 1.184 | 2.62 |
| 2.0 | 0.75 | 1.75 | 0.276 | 0.158 | 1.158 | 3.03 |

**Circuit 5** $\frac{L_n}{X_n} \sin\theta \log_e \left[1 + \frac{X_n}{1+\sum_1^n X_{n-1}}\right]$

| $V_0$ | $\sum_1^n X_{n-1}$ | $1+\sum_1^n X_{n-1}$ | $X_n$ | $\frac{X_n}{1+\sum_1^n X_{n-1}}$ | $1+\frac{X_n}{\sum_1^n X_{n-1}}$ | Gravity Head |
|---|---|---|---|---|---|---|
| 0.5 | 4.104 | 5.104 | 0.904 | 0.177 | 1.177 | 0.73 |
| 1.0 | 2.052 | 3.052 | 0.452 | 0.148 | 1.148 | 1.23 |
| 1.5 | 1.368 | 2.368 | 0.301 | 0.127 | 1.127 | 1.60 |
| 2.0 | 1.026 | 2.026 | 0.226 | 0.112 | 1.112 | 1.89 |

**Circuit 6** $\frac{L}{X_n} \sin\theta \log_e \left[1 + \frac{X_n}{1+\sum X_{n-1}}\right]$

| $V_0$ | $\sum_1^n X_{n-1}$ | $1+\sum_1^n X_{n-1}$ | $X_n$ | $\frac{X_n}{1+\sum_1^n X_{n-1}}$ | $1+\frac{X_n}{1+\sum_1^n X_{n-1}}$ | Gravity Head |
|---|---|---|---|---|---|---|
| 0.5 | 5.008 | 6.008 | 0.388 | 0.065 | 1.065 | 0.21 |
| 1.0 | 2.504 | 3.504 | 0.194 | 0.0554 | 1.0554 | 0.36 |
| 1.5 | 1.669 | 2.669 | 0.129 | 0.0484 | 1.0484 | 0.47 |
| 2.0 | 1.252 | 2.252 | 0.097 | 0.043 | 1.043 | 0.56 |

**Losses**

**Acceleration Loss**

**Circuit 1** —0

**Circuit 2** $\frac{V_o^2}{2g} \times 2X_n$

| $V_o$ | $\frac{V_o^2}{2g}$ | $X_n$ | Acceleration Loss |
|---|---|---|---|
| 0.5 | .0039 | 1.32 | .0103 |
| 1.0 | .0155 | 0.66 | .0205 |
| 1.5 | .035 | 0.44 | .0309 |
| 2.0 | .062 | 0.33 | .041 |

**Circuit 3** $\frac{V_o^2}{2g} \times 2X_n$

| $V_o$ | $\frac{V_o^2}{2g}$ | $X_n$ | Acceleration Loss |
|---|---|---|---|
| 0.5 | 0.0039 | 1.68 | 0.013 |
| 1.0 | 0.0155 | 0.84 | 0.026 |
| 1.5 | 0.0350 | 0.56 | 0.039 |
| 2.0 | 0.0620 | 0.42 | 0.052 |

**Circuit 4** $\frac{V_o^2}{2g} \times 2X_n$

| $V_o$ | $\frac{V_o^2}{2g} \times 2X_n$ | $X_n$ | Acceleration Loss |
|---|---|---|---|
| 0.5 | 0.0039 | 1.104 | 0.009 |
| 1.0 | 0.0155 | 0.552 | 0.017 |
| 1.5 | 0.0350 | 0.368 | 0.0260 |
| 2.0 | 0.0620 | 0.276 | 0.034 |

**Circuit 5** $\frac{{V_o}^2}{2g} \times 2X_n$

| $V_o$ | $\frac{V_o^2}{2g}$ | $X_n$ | Acceleration Loss |
|---|---|---|---|
| 0.5 | 0.0039 | 0.904 | 0.007 |
| 1.0 | 0.0155 | 0.452 | 0.014 |
| 1.5 | 0.0350 | 0.301 | 0.021 |
| 2.0 | 0.0620 | 0.226 | 0.028 |

**Circuit 6** $\frac{V_o^2}{2g} \times 2X_n$

| $V_o$ | $\frac{V_o^2}{2g}$ | $X_n$ | Acceleration Loss |
|---|---|---|---|
| 0.5 | 0.0039 | 0.388 | 0.003 |
| 1.0 | 0.0155 | 0.194 | 0.006 |
| 1.5 | 0.0350 | 0.129 | 0.009 |
| 2.0 | 0.0620 | 0.097 | 0.012 |

**Inside Wall: Generating Furnace (cont.)**

**Entrance and Exist Loss**

$$1.5\frac{V_o^2}{2g}$$

| $V_o$ | $\frac{V_1^2}{2g}$ | Entrance and Exit Loss |
|---|---|---|
| 0.5 | 0.039 | 0.006 |
| 1.0 | 0.155 | 0.023 |
| 1.5 | 0.350 | 0.053 |
| 2.0 | 0.620 | 0.093 |

**Bend Loss**

**Circuit 1:** Essentially as corresponding section Generating Furnace: Outside Wall.

**Circuit 2:** 1 bend

$$K_B = 0.25; \quad K_B\frac{V_o^2}{2g}\left[1+\textstyle\sum_1^n X_{n-1}\right]$$

| $V_o$ | $\frac{V_1^2}{2g}$ | $1+\sum X_{n-1}$ | Bend Loss |
|---|---|---|---|
| 0.5 | 0.039 | 2.32 | 0.002 |
| 1.0 | 0.155 | 1.66 | 0.006 |
| 1.5 | 0.350 | 1.44 | 0.013 |
| 2.0 | 0.620 | 1.33 | 0.021 |

**Circuit 3:** —0

**Circuit 4:** 1 bend

$$K_B = 0.24; \quad K_B\frac{V_o^2}{2g}\left[1+\textstyle\sum_1^n X_{n-1}\right]$$

| $V_o$ | $\frac{V_1^2}{2g}$ | $1+\sum X_{n-1}$ | Bend Loss |
|---|---|---|---|
| 0.5 | 0.039 | 5.104 | 0.005 |
| 1.0 | 0.155 | 3.052 | 0.011 |
| 1.5 | 0.35 | 2.368 | 0.020 |
| 2.0 | 0.62 | 2.026 | 0.030 |

**Circuit 5:** 1 bend

$$K_B = 0.3; \quad K_B \frac{V_o^2}{2g} \left[1 + \sum_1^n X_{n-1}\right]$$

| $V_o$ | $\frac{V_o^2}{2g}$ | $1 + \sum X_{n-1}$ | Bend Loss |
|---|---|---|---|
| 0.5 | 0.0039 | 6.008 | 0.007 |
| 1.0 | 0.0155 | 3.504 | 0.016 |
| 1.5 | 0.0350 | 2.669 | 0.028 |
| 2.0 | 0.0620 | 2.252 | 0.042 |

**Circuit 6:** $O$

**Friction Loss**

**Circuit 1** Essentially the same as corresponding section Generating Furnace: Outside Wall.

**Circuit 3**

$$.288 \frac{L}{d} \frac{V_o^2}{2g} \left[1 + \frac{X_n}{2} + \sum_1^n X_{n-1}\right]$$

| $V_o$ | $\frac{.288 V_o^2}{2g}$ | $\frac{L}{d}$ | $\frac{X_n}{2}$ | Friction Loss |
|---|---|---|---|---|
| 0.5 | 0.001125 | 3.17 | 1.660 | 0.006 |
| 1.0 | 0.004470 | 3.17 | 1.330 | 0.019 |
| 1.5 | 0.010100 | 3.17 | 1.220 | 0.039 |
| 2.0 | 0.017850 | 3.17 | 1.165 | 0.066 |

**Circuit 4**

$$.288 \frac{L}{d} \frac{V_o^2}{2g} \left[1 + \frac{X_n}{2} + \sum_1^n X_{n-1}\right]$$

| $V_o$ | $\frac{.288 V_o^2}{2g}$ | $\frac{L}{d}$ | $\frac{X_n}{2}$ | $1+ \sum_1^n X_{n-1}$ | Friction Loss |
|---|---|---|---|---|---|
| 0.5 | 0.001125 | 6.87 | 0.84 | 2.32 | 0.024 |
| 1.0 | 0.004470 | 6.87 | 0.42 | 1.66 | 0.064 |
| 1.5 | 0.010100 | 6.87 | 0.28 | 1.44 | 0.120 |
| 2.0 | 0.017850 | 6.87 | 0.21 | 1.33 | 0.189 |

**Circuit 4**

$$.288 \frac{L}{d} \frac{V_o^2}{2g} \left[1 + \frac{X_n}{2} + \sum_1^n X_{n-1}\right]$$

| $V_o$ | $\frac{.288 V_o^2}{2g}$ | $\frac{L}{d}$ | $\frac{X_p}{2}$ | $1 + \sum_1^n X_{n-1}$ | Friction Loss |
|---|---|---|---|---|---|
| 0.5 | 0.001125 | 2.26 | 0.552 | 4.00 | 0.012 |
| 1.0 | 0.004470 | 2.26 | 0.276 | 2.50 | 0.028 |
| 1.5 | 0.010100 | 2.26 | 0.184 | 2.00 | 0.050 |
| 2.0 | 0.017850 | 2.26 | 0.138 | 1.75 | 0.076 |

**Circuit 5**

$$.288 \frac{L}{d} \frac{V_o^2}{2g} \left[1 + \frac{X_n}{2} + \sum_1^n X_{n-1}\right]$$

| $V_o$ | $.288 \frac{V_o^2}{2g}$ | $\frac{L}{d}$ | $\frac{X_n}{2}$ | $1 + \sum_1^n X_{n-1}$ | Friction Loss |
|---|---|---|---|---|---|
| 0.5 | 0.001125 | 1.85 | 0.452 | 5.104 | 0.012 |
| 1.0 | 0.00447 | 1.85 | 0.226 | 3.052 | 0.027 |
| 1.5 | 0.01010 | 1.85 | 0.151 | 2.368 | 0.047 |
| 2.0 | 0.01785 | 1.85 | 0.113 | 2.026 | 0.071 |

**Circuit 6**

$$.288 \frac{L}{d} \frac{V_o^2}{2g} \left[1 + \frac{X_n}{2} + \sum_1^n X_{n-1}\right]$$

| $V_o$ | $.288 \frac{V_o^2}{2g}$ | $\frac{L}{d}$ | $\frac{X_n}{2}$ | $1 + \sum_1^n X_{n-1}$ | Friction Loss |
|---|---|---|---|---|---|
| 0.5 | 0.001125 | 0.80 | 0.194 | 6.008 | 0.006 |
| 1.0 | 0.004470 | 0.80 | 0.097 | 3.504 | 0.013 |
| 1.5 | 0.01010 | 0.80 | 0.065 | 2.669 | 0.022 |
| 2.0 | 0.01785 | 0.80 | 0.048 | 2.252 | 0.033 |

**Feeder Loss**

$$3.78 \left(\frac{1}{0.1553}\right)^2 \times \frac{V_o^2}{2g}$$

| $V_o$ | Feeder Loss | $\sum$ Gravity Head + $\sum$ Losses |
|---|---|---|
| 0.5 | 0.61 | 24.00 |
| 1.0 | 2.43 | 31.32 |
| 1.5 | 5.46 | 37.83 |
| 2.0 | 9.72 | 44.67 |

**Boiler Bank**

**Refer to Fig. 26**

In this section the average heated length of tube is 37 feet and may be taken as a single vertical tube.

$$N = 0.040$$

**Gravity Head**

$$\frac{L_n}{x_n} \log_e [1 + X_n] + 1 + \frac{2}{1 + X_n}$$

| $V_0$ | $x_n$ | $1 + x_n$ | Gravity Head |
|---|---|---|---|
| 0.5 | 2.720 | 3.720 | 16.94 |
| 1.0 | 1.360 | 2.360 | 23.35 |
| 1.5 | 0.907 | 1.907 | 26.20 |
| 2.0 | 0.680 | 1.680 | 28.09 |

**Acceleration Loss**

$$\frac{V_o}{2g} \times 2X_n$$

| $V_o$ | $\frac{V_o^2}{2g}$ | $2X_n$ | Acceleration Loss |
|---|---|---|---|
| 0.5 | 0.039 | 5.440 | 0.021 |
| 1.0 | 0.155 | 2.720 | 0.042 |
| 1.5 | 0.350 | 1.813 | 0.064 |
| 2.0 | 0.620 | 1.360 | 0.084 |

**Bend Loss:** Essentially 0

**Entrance and Exit Loss; Friction Loss**

$$\frac{1.5V_o^2}{2g}; .288\frac{V_o^2}{2g}\frac{L}{d}\left[1+\frac{X_n}{2}\right]$$

| $V_o$ | $\frac{V_o^2}{2g}$ | Entrance & Exit Loss | $\frac{L}{d}$ | $1+\frac{X_n}{2}$ | Friction Loss | $\sum$ Gravity Head+ $\sum$ Losses |
|---|---|---|---|---|---|---|
| 0.5 | 0.039 | 0.06 | 200 | 2.360 | 0.53 | 17.02 |
| 1.0 | 0.155 | 0.23 | 200 | 1.680 | 1.50 | 23.57 |
| 1.5 | 0.350 | 0.53 | 200 | 1.440 | 2.91 | 26.61 |
| 2.0 | 0.620 | 0.93 | 200 | 1.340 | 4.78 | 28.75 |

From Mr. Lorenzini's report the following values were found:

**VELOCITY ENTERING WALL TUBES - Ft./Sec.**

| **WALL** | $V_o$ |
|---|---|
| Generating side wall | 1.687 |
| Superheater side wall | 1.692 |
| Generating front wall | 0.894 |
| Superheater front wall | 1.011 |
| Generating rear wall | 1.262 |
| Superheater rear wall | 1.278 |
| Inside wall | 1.182 |

**B. Calculations By Use of Curves**

All the computations in this section are independent of those made in the previous section of this work. For each wall, refer to the figure of the typical tube in the wall.

**Generating Furnace: Outside Wall**

**Circuit 1**

| $V_o$ | Gravity Head | $\frac{L}{d}$ | Losses |
|---|---|---|---|
| 0.5 | 10.34 | 5.02 | 0.012 |
| 1.0 | 10.34 | 5.02 | 0.055 |
| 1.5 | 10.34 | 5.02 | 0.115 |
| 2.0 | 10.34 | 5.02 | 0.210 |

**Circuit 2**

| $X_n$ | Gravity Head | $\frac{L}{d}$ | Losses |
|---|---|---|---|
| 2.42 | $25 \times 0.51 = 12.75$ | 9.93 | 0.050 |
| 1.21 | $25 \times 0.655 = 16.40$ | 9.93 | 0.138 |
| 0.806 | $25 \times 0.735 = 18.40$ | 9.93 | 0.260 |
| 0.605 | $25 \times 0.783 = 19.55$ | 9.93 | 0.440 |

**Circuit 3**

| $X_n$ | $\sum_1^n X_{n-1}$ | $\frac{L}{d}$ | Gravity Head | Losses |
|---|---|---|---|---|
| 2.23 | 2.42 | 4.58 | 11.5 × 0.225 = 2.59 | 0.035 + 0.029 = 0.64 |
| 1.115 | 1.21 | 4.58 | 11.5 × 0.367 = 4.22 | 0.95 + 0.64 = 1.59 |
| 0.743 | 0.806 | 4.58 | 11.5 × 0.460 = 5.30 | 0.175 + 0.088 = 0.263 |
| 0.557 | 0.605 | 4.58 | 11.5 × 0.535 = 6.15 | 0.280 + 0.121 = 0.401 |

**Circuit 4**

| $V_o$ | $X_n$ | $\sum_1^n X_{n-1}$ | $\frac{L}{d}$ | Gravity Head | Losses | Feeder Loss $78.4\frac{V_o^2}{2g}$ | $\sum$ Gravity Head+ $\sum$ Losses |
|---|---|---|---|---|---|---|---|
| 0.5 | 0 | 4.65 | 5.25 | 0.62 | 0.67 | 0.305 | 26.80 |
| 1.0 | 0 | 2.325 | 5.25 | 1.05 | 0.175 | 1.218 | 33.75 |
| 1.5 | 0 | 1.549 | 5.25 | 1.37 | 0.282 | 2.740 | 39.07 |
| 2.0 | 0 | 1.162 | 5.25 | 1.62 | 0.432 | 4.860 | 44.00 |

### Superheater Furnace: Outside Wall

All values of Gravity Head are essentially the same as corresponding sections of **Generating Furnace: Outside Wall**. All values of Losses are the same as those in corresponding sections of **Generating Furnace: Outside Wall** except for the last section.

**Circuit 4**

| $V_o$ | $\sum_1^n; X_n = 0$ | $\frac{L}{d}$ | Losses | $\sum$ Gravity Head+ $\sum$ Losses |
|---|---|---|---|---|
| 0.5 | 4.650 | 11.7 | 0.118 | 26.85 |
| 1.0 | 2.325 | 11.7 | 0.316 | 33.90 |
| 1.5 | 1.550 | 11.7 | 0.472 | 39.26 |
| 2.0 | 1.165 | 11.7 | 0.692 | 44.26 |

**Front and Rear Walls: Both Furnaces**

**Circuit 1**

| $V_o$ | Gravity Head | $\frac{L}{d}$ | Losses | $X_n$ |
|---|---|---|---|---|
| 0.5 | 4.75 | 3.30 | 0.011 | 2.42 |
| 1.0 | 4.75 | 3.30 | 0.045 | 1.21 |
| 1.5 | 4.75 | 3.30 | 0.100 | 0.806 |
| 2.0 | 4.75 | 3.30 | 0.180 | 0.605 |

**Circuit 2**

| $V_o$ | Gravity Head | $\frac{L}{d}$ | Losses |
|---|---|---|---|
| 0.5 | $25 \times 0.51 = 12.75$ | 9.93 | 0.050 |
| 1.0 | $25 \times 0.655 = 16.40$ | 9.93 | 0.138 |
| 1.5 | $25 \times 0.735 = 18410$ | 9.93 | 0.260 |
| 2.0 | $25 \times 0.783 = 195.75$ | 9.93 | 0.440 |

**Circuit 3**

| $V_o$ | $X_n$ | $\sum_1^n X_{n-1}$ | $\frac{L}{d}$ | Gravity Head | Losses |
|---|---|---|---|---|---|
| 0.5 | 2.64 | 2.42 | 4.83 | $136 \times 217 = 295$ | 0.071 |
| 1.0 | 1.32 | 1.21 | 4.83 | $136 \times 355 = 483$ | 0.169 |
| 1.5 | 0.86 | 0.806 | 4.83 | $136 \times 454 = 616$ | 0.279 |
| 2.0 | 0.66 | 0.605 | 4.83 | $136 \times 523 = 710$ | 0.431 |

**Feeder to Drum**

| | Feeder to Wall | Generating Furnace | | | | Superheater Furnace | | | |
|---|---|---|---|---|---|---|---|---|---|
| $V_o$ | $2.71\frac{V_o^2}{2g}\times\left(\frac{1}{0.1556}\right)^2$ | $\sum_1^n X_{n-1}$ | $\frac{L}{d}$ | Gravity Head | Losses | $\sum_1^n X_{n-1}$ | $\frac{L}{d}$ | Gravity Head | Losses |
| 0.5 | 0.438 | 5.06 | 0.96 | 3 × 1.65 = 4.95 | 0.337 | 5.06 | 1.52 | 4.75 × 0.165 = 0.785 | 0.379 |
| 1.0 | 1.748 | 2.53 | 0.96 | 3 × 0.284 = 0.85 | 0.797 | 2.53 | 1.52 | 4.75 × 0.284 = 1.350 | 0.859 |
| 1.5 | 3.940 | 1.666 | 0.96 | 3 × 0.375 = 1.125 | 1.130 | 1.666 | 1.52 | 4.75 × 0.375 = 1.770 | 1.210 |
| 2.0 | 6.980 | 1.265 | 0.96 | 3 × 0.441 = 1.323 | 1.800 | 1.265 | 1.52 | 4.75 × 0.441 = 2.090 | 1.920 |

**Feeder to Drum (cont.)**

**$\sum$ Gravity Head + $\sum$ Losses**

| Generating Furnace | Superheater Furnace |
|---|---|
| 21.85 | 22.18 |
| 29.73 | 30.29 |
| 36.14 | 36.87 |
| 42.56 | 43.44 |

**Inside Wall: Generating Furnace**

**Circuit 1**

| $V_o$ | Gravity Head | $\frac{L}{d}$ | Losses |
|---|---|---|---|
| 0.5 | 10.34 | 5.02 | 0.012 |
| 1.0 | 10.34 | 5.02 | 0.055 |
| 1.5 | 10.34 | 5.02 | 0.115 |
| 2.0 | 10.34 | 5.02 | 0.210 |

**Circuit 2**

| $V_o$ | $X_n$ | Gravity Head | $\frac{L}{d}$ | Losses |
|---|---|---|---|---|
| 0.5 | 1.32 | $8 \times 0.635 = 5.07$ | 3.17 | 0.023 |
| 1.0 | 0.66 | $8 \times 0.77 = 6.16$ | 3.17 | 0.068 |
| 1.5 | 0.44 | $8 \times 0.828 = 6.63$ | 3.17 | 0.130 |
| 2.0 | 0.33 | $8 \times 0.865 = 6.92$ | 3.17 | 0.210 |

**Circuit 3**

| $V_o$ | $X_n$ | $\sum_1^n X_{n-1}$ | $\frac{L}{d}$ | Gravity Head | Losses |
|---|---|---|---|---|---|
| 0.5 | 1.68 | 1.32 | 6.87 | $17.3 \times 0.325 = 5.63$ | $0.034 + 0.20 = 0.054$ |
| 1.0 | 0.84 | 0.66 | 6.87 | $17.3 \times 0.49 = 8.46$ | $0.098 + 0.042 = 0.140$ |
| 1.5 | 0.56 | 0.44 | 6.87 | $17.3 \times 0.587 = 10.15$ | $0.185 + 0.058 = 0.243$ |
| 2.0 | 0.42 | 0.33 | 6.87 | $17.3 \times 0.655 = 11.35$ | $0.310 + 0.076 = 0.386$ |

**Circuit 4**

| $V_o$ | $X_n$ | $\sum_1^n X_{n-1}$ | $\frac{L}{d}$ | Gravity Head | Losses |
|---|---|---|---|---|---|
| 0.5 | 1.104 | 3.00 | 2.26 | 5.7 × 0.22 = 1.26 | 0.019 + 0.023 = 0.042 |
| 1.0 | 0.552 | 1.500 | 2.26 | 5.7 × 0.36 = 2.06 | 0.057 + 0.060 = 0.117 |
| 1.5 | 0.368 | 1.00 | 2.26 | 5.7 × 0.458 = 2.62 | 0.115 + 0.087 = 0.202 |
| 2.0 | 0.276 | 0.75 | 2.26 | 5.7 × 0.535 = 3.05 | 0.190 + 0.116 = 0.306 |

**Circuit 5**

| $V_o$ | $X_n$ | $\sum_1^n X_{n-1}$ | $\frac{L}{d}$ | Gravity Head | Losses | Feeder Loss |
|---|---|---|---|---|---|---|
| 0.5 | 0.904 | 4.104 | 1.85 | $4.03 \times 0.18 = 0.73$ | $0.017 + 0.041 = 0.058$ | 0.61 |
| 1.0 | 0.904 | 4.104 | 1.85 | $4.03 \times 0.305 = 1.23$ | $0.053 + 0.081 = 1.34$ | 2.43 |
| 1.5 | 0.301 | 1.368 | 1.85 | $4.03 \times 0.40 = 1.61$ | $0.108 + 0.117 = 0.225$ | 5.46 |
| 2.0 | 0.226 | 1.026 | 1.85 | $4.03 \times 0.47 = 1.90$ | $0.180 + 0.156 = 0.336$ | 9.73 |

**Circuit 6**

| $V_o$ | $X_n$ | $\sum_1^n X_{n-1}$ | $\frac{L}{d}$ | Gravity Head | Losses | $\sum$ Gravity Head $+\sum$ Losses |
|---|---|---|---|---|---|---|
| 0.5 | 0.388 | 5.008 | 0.80 | 1.3 × 0.162 = 0.21 | 0.011 + 0.040 = 0.051 | 24.10 |
| 1.0 | 0.194 | 2.504 | 0.80 | 1.3 × 0.278 = 0.36 | 0.035 + 0.075 = 0.110 | 31.66 |
| 1.5 | 0.129 | 1.669 | 0.80 | 1.3 × 0.365 = 0.47 | 0.70 + 0.100 = 0.170 | 38.36 |
| 2.0 | 0.097 | 1.252 | 0.80 | 1.3 × 0.435 = 0.56 | 0.130 + 0.150 = 0.280 | 45.57 |

**Boiler Bank**

| $V_o$ | $\frac{L}{d}$ | $X_n$ | Gravity Head (total) | Losses | $\sum$ Gravity Head $+\sum$ Losses |
|---|---|---|---|---|---|
| 0.5 | 20.0 | 2.720 | 16.95 | 0.081 | 17.03 |
| 1.0 | 20.0 | 1.360 | 23.35 | 0.218 | 23.57 |
| 1.5 | 20.0 | 0.907 | 26.20 | 0.420 | 26.62 |
| 2.0 | 20.0 | 0.680 | 28.10 | 0.650 | 28.75 |

The values of Gravity Head and Losses as tabulated on the preceding pages were obtained as follows:

To obtain the Gravity Head of any particular element, the values of $X_n$ and $X_n^{-1}$ were tabulated as previously. The vetical height of the ciruit was then multiplied by the value of $\bar{P}$ as read from Fig. 14 at these values of $X_n$ and $\sum_1^n Xn - 1$. This gave the Gravity Head of the circuit. The Losses were obtained by entering the proper curve for $V_o$ at the proper values of $\frac{L}{d}$ and $X_n$. As previously explained in Section II, to this value read from the curves, the quantity equal to this value read at $X_n = 0$ multiplied by $\sum_1^n X_n - 1$ was added to obtain the total losses in the circuit in question.

It will be noted from the tabulations of $\sum$ Gravity Head $- \sum$ Losses that the values as computed from the graphical method are numerically greater than the values obtained from the straightforward application of the equations developed earlier in this work. In this particular case, the tube was divided into $n$ series circuits. Therefore, the use of the curves introduced $n - 1$ extraneous entrance and exit losses. This accounts for the greater part of the difference found in the tabulations. The assumption of one bend with a $K_B$ of 0.3 in each of the series circuits does not affect the final answer obtained by use of the curves to any appreciable extent. However, the total deviation in the final values is so small as to be negligible. The inherent assumptions made in the derivations and the assumptions in the rates of heat absorption can lead to errors that may be of the order of hundreds of times greater than any made by using the curves. The use of the curves will, in general, yield values of $V_o$ slightly lower than use of the equations, and consequently, slightly lower circulation. This is conservative for design purposes.

On the following pages Figs. 29 & 30 show the values of $\sum$ Gravity Head $- \sum$ Losses plotted as functions of $V_o$.

The Front and Rear, and Outside Walls of this installation are fed by two downcomers having an inside diameter of 13.5 in. and a length of 49.4 ft.

$$\textbf{Friction Loss} = 4f\frac{L}{d}\frac{V^2}{2g} = 1.06\frac{V^2}{2g}$$

$$\textbf{Entrance and Exit Loss} = 1.5\frac{V^2}{2g}$$

$$\textbf{Bend Loss} \quad \text{(essentially)} = 0.3\frac{V^2}{2g}$$

As can be seen from Fig. 23, these downcomers join a manifold from which the separate walls are fed. The loss at this "T" connection is very closely $2\frac{V^2}{2g}$.[88] In addition, there is the friction loss in the manifold which, when taken as one-third of the normal friction loss,[49] is closely $0.7\frac{V_2^2}{2g}$.

Total loss in these downcomers is equal to $5.56\frac{V^2}{2g}$.

$$\textbf{Head available: Front and Rear Walls} = 44.75 - 5.56\frac{V^2}{2g}$$

$$\textbf{Head available: Outside Walls} = 50.34 - 5.56\frac{V^2}{2g}$$

The method of finding the operating point of the various walls consists of assuming various values of downcomer velocities. From each assumption, the head available to each of the walls is obtained and consequently each $V_o$. If the values of $V_o$ for the head assumed give a total water flow that differs from the initial assumption, a new value of downcomer velocity is assumed until agreement is reached. Doing this, the following values of $V_o$ are found:

| | |
|---|---|
| **Superheater Furnace: Outside Wall** | 1.680 |
| **Superheater Furnace: Front and Rear Walls** | 1.360 |
| **Generating Furnace: Outside Wall** | 1.710 |
| **Generating Furnace: Front and Rear Walls** | 1.400 |

The downcomers for the **Boiler Bank** and **Inside Wall: Generating Furnace** extend from the upper steam drum to the mud drum. From the mud drum, feeders are taken to the lower header of the **Inside Wall: Generating Furnace**.

| | |
|---|---|
| **Head available: Boiler Bank** | $37.0 - 2.93\frac{V^2}{2g}$ |
| **Head available: Inside Wall-Generating Furnace** | $50.34 - 2.93\frac{V^2}{2g}$ |

The values of $V_o$ for these walls are closely aligned:

| | |
|---|---|
| **Inside Wall: Generating Furnace** | 1.50 |
| **Boiler Bank** | 1.25 |

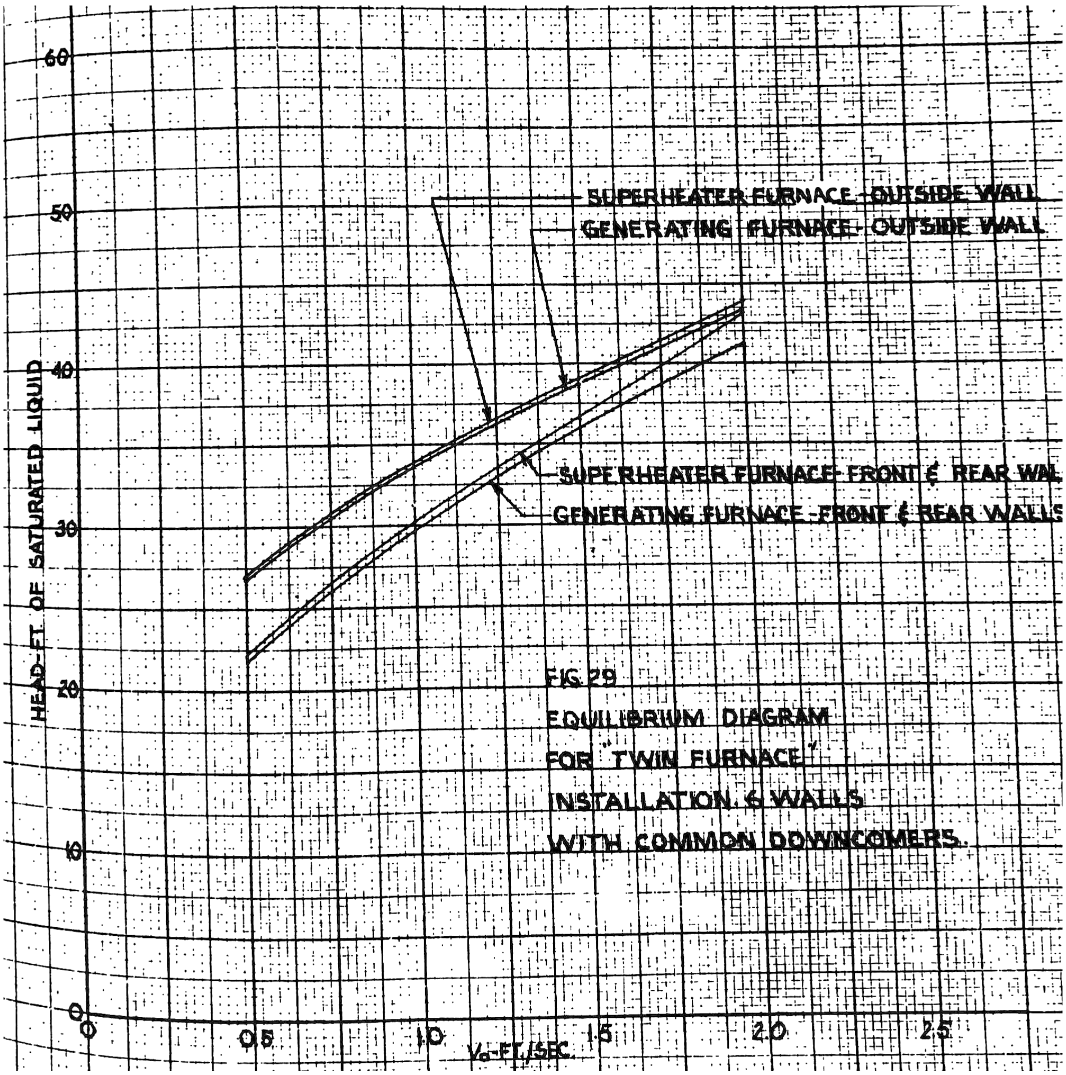



FIG 29
EQUILIBRIUM DIAGRAM
FOR "TWIN FURNACE"
INSTALLATION 6 WALLS
WITH COMMON DOWNCOMERS.

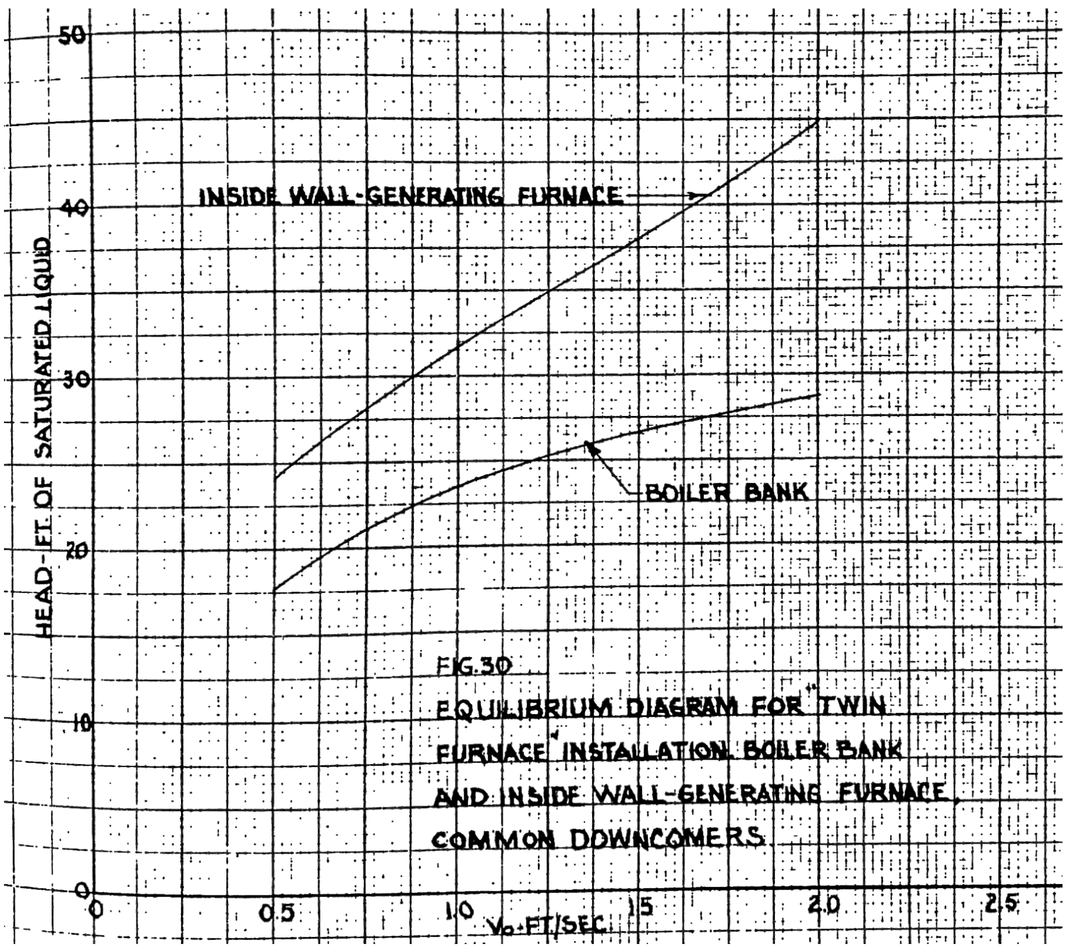



FIG.30
EQUILIBRIUM DIAGRAM FOR "TWIN FURNACE" INSTALLATION BOILER BANK AND INSIDE WALL-GENERATING FURNACE COMMON DOWNCOMERS.

Knowing $V_o$, it is now possible to completely determine the circulation characteristics of each wall. These are tabulated below. because of the close agreement of the values determined for each wall by use of the special curves and by direct computation, only the values found by the direct computation method are given here.

| | $V_o$ | Water Entering Wall (lbs./hr.) | Steam Generated (lbs/hr) | $\beta_n$ | $\lambda_n$ | $\frac{1}{\lambda_n}$ = Circulation Ratio |
|---|---|---|---|---|---|---|
| **Wall Outside Generating** | 1.71 | 1,028,000 | 66,000 | 0.605 | 0.0642 | 15.6 |
| **Outside Superheater Front** | 1.68 | 1,010,000 | 66,000 | 0.609 | 0.0654 | 15.3 |
| **Generating Front** | 1.40 | 366,000 | 31,300 | 0.676 | 0.0855 | 11.7 |
| **Superheater Rear** | 1.36 | 355,000 | 31,200 | 0.682 | 0.0878 | 11.4 |
| **Generating Rear** | 1.40 | 366,000 | 31,300 | 0.676 | 0.0855 | 11.7 |
| **Superheater Inside** | 1.36 | 355,000 | 31,200 | 0.682 | 0.0878 | 11.4 |
| **Generating Inside** | 1.50 | 902,000 | 76,500 | 0.675 | 0.0848 | 11.8 |
| **Boiler Bank** | 1.25 | 1,330,000 | 68,500 | 0.547 | 0.0515 | 19.4 |
| **Total** | | **5,712,000** | **402,000** | **Overall Circulation Ratio** | | **14.2** |

From the tabulation on the preceding page and the values of $V_o$ from Mr. Lorenzini's tests, it can be seen that the computed values are in fair agreement with test values. It will also be noted that the total steam generation computed does not agree with the value of $422,000$ (including steam condensation to heat the feed water to saturation conditions) used. On a previous page it was assumed that the total heat absorption in the **Radiant Wall: Superheater Furnace** could be taken as essentially the same as that in the **Inside Wall: Generating Furnace**, also the heat absorption in various sections of this unit was found by difference and these methods could easily lead to the discrepancy noted. Since this discrepancy is of the order of 2% it will not affect the final tabulation enough to warrant an adjustment of the values.

The author had occasion to speak to Mr. Lorenzini concerning the tests made on this unit, and Mr. Lorenzini noted the following difficulties encountered during this test:

1. One of the meters used on the downcomers of this unit was apparently faulty and regardless of all the attempts made to correct it, it still gave consistently low readings regardless of where it was positioned.

2. Difficulty was had with the burners during the test at this load and Mr. Lorenzini and the author both feel that this may have caused a condition in the furnaces to account for the fact that the test values for the **Front Walls**—both furnaces were consistently lower than the values for the **Rear Walls**.

In view of these difficulties, it is felt that the agreement of the calculated values with test values is very good for the water walls.

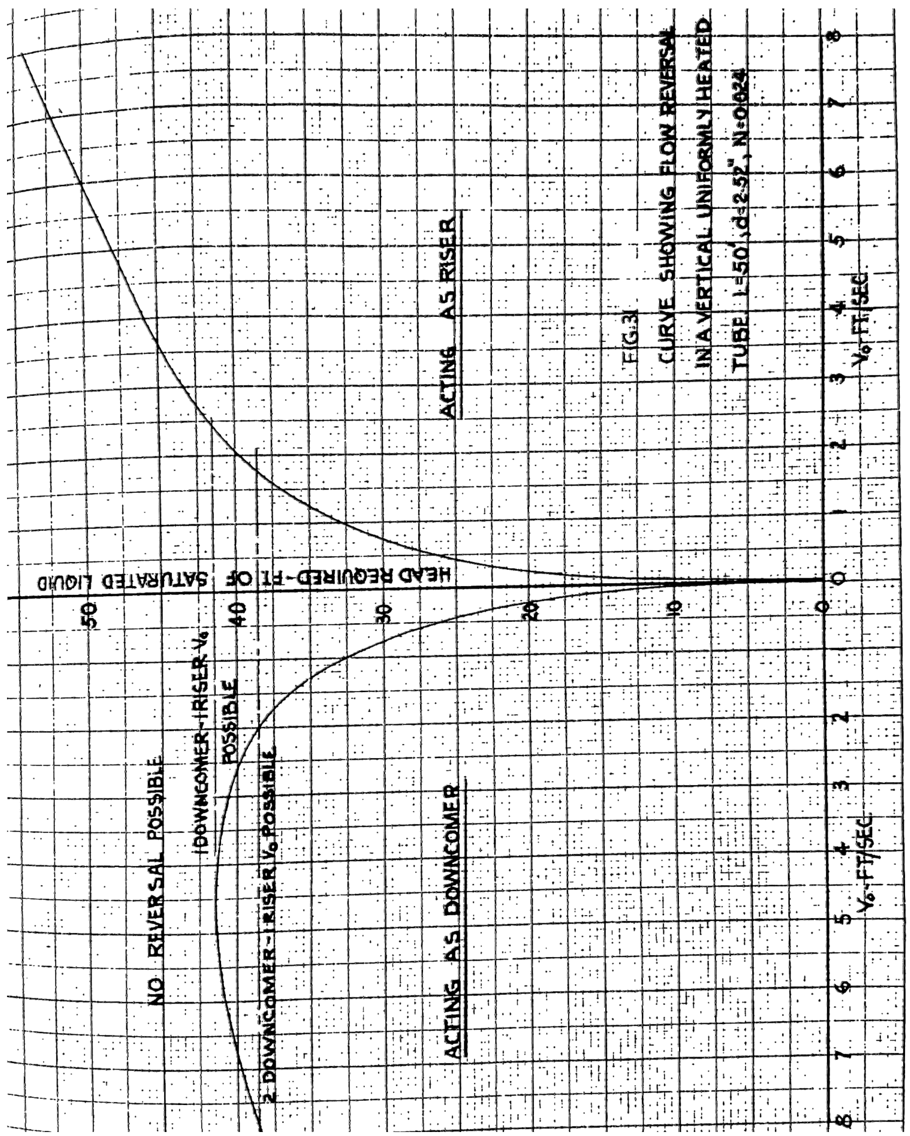
Before leaving this discussion, it must be noted that the entering velocity ($V_o$) of the **Inside Wall: Generating Furnace** as computed is greater than the test values. The downcomers for this wall and the **Boiler Bank** are heated to some extent even though they are insulated. In order to determine the possible effect of steam generation in these downcomers, the author assumed that the steam generated in them was 1% by weight. Using this value, almost exact agreement was found with test values. While in itself, this is not conclusive evidence that this amount of steam is generated in these downcomers, it is an indication that steam is generated in these downcomers. If it had been possible to determine the steam generation in these tubes, it could have been taken into account in the computations. Because of their location, such a determination is practically impossible, and under the circumstances, the indication that steam is generated in these downcomers must be taken as only a surmise.

**C. Flow Reversal**

Dr. B. Leib [64,72] has demonstrated the fact that it is possible for a heated riser tube to reverse its direction of flow and act as a downcomer under certain conditions. If the tube acts as either a downcomer or riser with a positive flow in the direction in which it is acting, the overall effect on the unit may not be too serious. However, if the tube is acting as a riser or downcomer and only a slight change in operating conditions is necessary for it to change its direction of flow, it is possible to have overheating occur in the tube due to steam blanketing or pocketing. The presence of steam in the downcomers will also cause the head available for flow to be materially decreased and consequently, in general, decrease the total water circulated. Special tests conducted by the Foster Wheeler Corporation have indicated that this may have occurred in actual boiler installations.

If a tube acts as a riser, the head necessary to maintain the flow at a given value of $V_o$ is equal to the $\sum$ Gravity Head $+ \sum$ Losses. When the tube acts as a downcomer, this becomes $\sum$ Gravity Head $- sum$ Losses. These statements are based upon the assumptions made in Section II of this work.

As an example, a vertical tube 50 feet long, having an inside diameter of 2.52 in. and a value of $N$ 0.024 will be used. It is further assumed that this tube has no bends. The following tabulation gives the results of all the necessary computations:


NO REVERSAL POSSIBLE
1 DOWNCOMER-1 RISER V₀ POSSIBLE
2 DOWNCOMER-1 RISER V₀ POSSIBLE
ACTING AS DOWNCOMER
ACTING AS RISER
HEAD REQUIRED-FT OF SATURATED LIQUID
50
40
30
20
10
0
1
2
3
4
5
6
7
8
V₀-FT/SEC
FIG.3
CURVE SHOWING FLOW REVERSAL
IN A VERTICAL UNIFORMLY HEATED
TUBE L=50', d=2.52", N=0.024

| $V_0$ | **Gravity Head** | **Acceleration Loss** | **Entrance & Exit Loss** | Friction Loss | **$\sum$ Losses** | **Gravity Head + $\Sigma$ Losses** | Gravity Head $-\Sigma$ Losses |
|---|---|---|---|---|---|---|---|
| 1.0 | 32.8 | 0.037 | 0.023 | 0.142 | 0.202 | 33.0 | 32.6 |
| 2.0 | 39.2 | 0.075 | 0.093 | 0.462 | 0.630 | 39.8 | 38.6 |
| 3.0 | 42.0 | 0.112 | 0.209 | 0.960 | 1.281 | 43.3 | 40.7 |
| 4.0 | 43.6 | 0.149 | 0.373 | 1.640 | 2.162 | 45.8 | 41.4 |
| 5.0 | 44.8 | 0.186 | 0.582 | 2.490 | 3.258 | 48.1 | 41.5 |
| 6.0 | 45.5 | 0.224 | 0.840 | 3.520 | 4.584 | 50.1 | 40.9 |
| 7.0 | 46.0 | 0.270 | 1.140 | 4.720 | 6.220 | 52.2 | 39.8 |
| 8.0 | 46.4 | 0.298 | 1.490 | 6.100 | 7.888 | 54.3 | 38.5 |

A curve of the head required for flow for the case of the tube acting as a downcomer or riser is given in Fig. 31. It will be noted from this set of curves that if the tube can act as a riser with velocities ($V_o$) up to approximately 2.5 ft./sec., it is also possible for this same tube to act as a downcomer at two distinct velocities and still be in equilibrium. At approximately equal to 2.5 ft./sec. (riser) the tube can act as a downcomer at only one velocity and still be in equilibrium. Riser velocities ($V_o$) greater than 2.5 ft./sec. show that there is no velocity at which this tube can act as a downcomer and still be in equilibrium.

From the previous tabulation and the curves in Fig. 31, it is apparent that the introduction of added resistance in the riser circuit will tend to ensure the tube acting as a riser. However, as Nothman has shown,[(1)] care must be taken so as to avoid the restriction of flow to any great extent. Also, the introduction of added resistance in the riser or out may cause the deposition of impurities on the tube walls. Such an action will tend to further restrict the flow and also lower the overall coefficient of heat transfer to a point where tube failure may occur.

In connection with this subject of flow reversal, the question of heated downcomers must be mentioned. It has been demonstrated that if heat is applied to the downcomers so as to bring the fluid to its saturation state, the effect is beneficial as far as circulation is concerned.[(1, 62)] Figs. 34 and 35 show the effect of steam entrainment on circulation. Additional heating beyond the saturation point is usually detrimental to the circulation characteristics of the unit as shown in the figures mentioned above. If sufficient heat is supplied to the downcomers, they will eventually, at some rate of heating, act as risers. However, it must be noted that once a tube starts to circulate in a given direction, reversal of flow will not occur if heat is applied to the downcomers at a later time, up to a limiting condition. Up to this condition, heat supplied to the downcomers is stored in the circulating fluid and released in the risers in the form of steam generation.[(1, 15, 62)]

**D. The Effect of Pressure on Circulation**

Nothman[(1)] and Lewis & Robertson[(62)] have made very comprehensive correlations of the various factors that affect circulation. One of the most important of these is pressure, and while both of the references mentioned above treat this question, it is worthwhile to treat it briefly here because of its importance.

As an example, a fictitious air circuit consisting of a riser circuit of 2.00 in. O.D. and 1.73 in. I.D. and a length of 50 ft. will be used. It will also be assumed that the downcomer circuit is also 50 ft. long and has a flow area equal to that of the riser circuit.

The following set of calculations refers to the circuit mentioned above for a $q$ of 10,000:

| Pressure (psia) | $N$ | $NL$ |
|---|---|---|
| 600 | 0.093 | 4.65 |
| 1000 | 0.058 | 2.90 |
| 1500 | 0.0415 | 2.07 |
| 2000 | 0.032 | 1.60 |
| 3000 | 0.0235 | 1.17 |

$$\text{Gravity Head} = \frac{V_o}{N} \log_e \left[1 + \frac{NL}{V_o}\right]$$

**Pressure psia**

| $V_o$ | 600 | 1000 | 1500 | 2000 | 3000 |
|---|---|---|---|---|---|
| 1 | 18.6 | 23.5 | 27.0 | 29.9 | 33.0 |
| 2 | 25.9 | 30.9 | 34.2 | 36.7 | 39.2 |
| 3 | 30.1 | 34.9 | 37.9 | 40.0 | 42.1 |
| 4 | 33.2 | 37.6 | 40.2 | 42.1 | 43.7 |
| 5 | 35.3 | 39.4 | 41.8 | 43.4 | 44.9 |
| 6 | 37.0 | 40.7 | 42.8 | 44.3 | 45.5 |
| 7 | 38.4 | 41.8 | 43.8 | 45.1 | 46.2 |
| 8 | 39.2 | 42.6 | 44.4 | 45.6 | 47.0 |

$$\text{Losses} = \frac{V_o^2}{2g}\left[2.88\left(1 + \frac{NL}{2V_o}\right) + 2\frac{NL}{V_o} + 1.8\right] \qquad (f = 0.06,\ K_B = 0.3)$$

$$= \frac{V_o^2}{2g}\left[10.12 + 6.16\frac{NL}{V_o}\right]$$

**Pressure (psia) Data**

| $V_0$ | 600 | 1000 | 1500 | 2000 | 3000 |
|---|---|---|---|---|---|
| 0 | 0.62 | 0.42 | 0.37 | 0.32 | 0.28 |
| 1 | 1.52 | 1.18 | 1.03 | 0.995 | 0.85 |
| 2 | 2.75 | 2.25 | 2.01 | 1.88 | 1.76 |
| 3 | 4.32 | 3.63 | 3.32 | 3.13 | 2.97 |
| 4 | 6.16 | 5.32 | 4.93 | 4.70 | 4.50 |
| 5 | 8.40 | 7.32 | 6.86 | 6.59 | 6.35 |
| 6 | 10.85 | 9.65 | 9.12 | 8.81 | 8.50 |
| 7 | 13.68 | 12.30 | 11.68 | 11.30 | 10.97 |

**Totals**

**Totals for Pressure (psia)**

| $V_0$ | 600 | 1000 | 1500 | 2000 | 3000 |
|---|---|---|---|---|---|
| 0 | 19.22 | 23.92 | 27.37 | 30.22 | 33.28 |
| 1 | 27.42 | 32.08 | 35.23 | 37.70 | 40.05 |
| 2 | 32.85 | 37.15 | 39.91 | 41.88 | 43.86 |
| 3 | 37.52 | 41.23 | 43.52 | 45.23 | 46.67 |
| 4 | 41.46 | 44.72 | 46.73 | 48.10 | 49.40 |
| 5 | 45.36 | 48.02 | 49.66 | 50.89 | 51.85 |
| 6 | 49.25 | 51.45 | 52.92 | 53.91 | 54.70 |
| 7 | 52.88 | 54.90 | 56.08 | 56.90 | 57.97 |

**Downcomer Losses**$= 10.12\frac{V_o^2}{2g}$

| $V_0$ | |
|---|---|
| 0 | 0.162 |
| 1 | 0.628 |
| 2 | 1.42 |
| 3 | 2.52 |
| 4 | 3.94 |
| 5 | 5.66 |
| 6 | 6.78 |
| 7 | 7.71 |
| 8 | 10.10 |

On the following page is a curve showing the values of head available and head required as functions of $V_o$. From this curve, the values of $V_o$ are obtained and the circulation computed. By a similar process to that given above, the circulation characteristics of the hypothetical circuit for 1 of 10,000; 20,000; 30,000 & 40,000 have been computed and the results of these computations are shown graphically in Fig. 33.

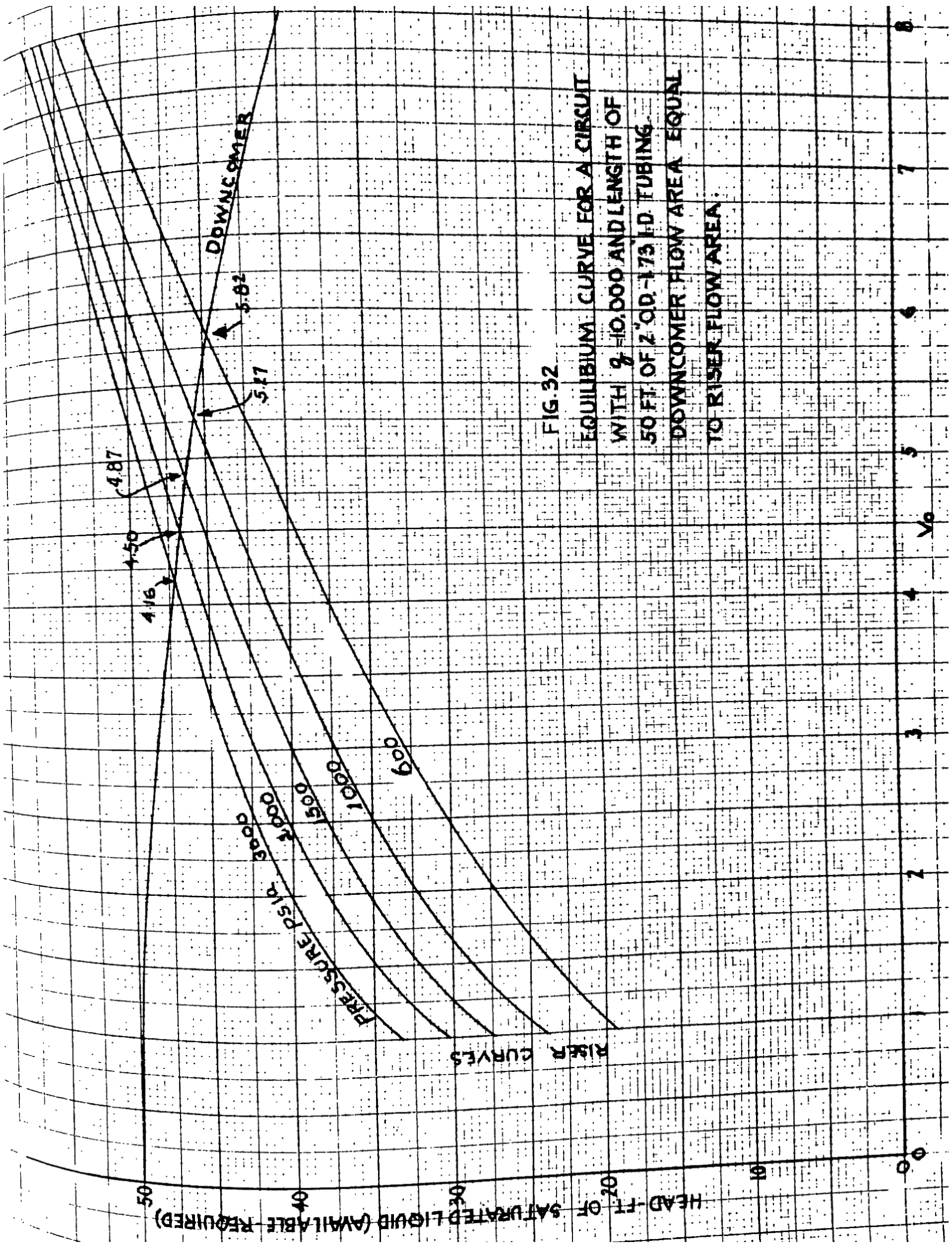



FIG. 32
EQUILIBRIUM CURVE FOR A CIRCUIT WITH $q/s$ = 10,000 AND LENGTH OF 50 FT. OF 2" O.D.–.173 I.D. TUBING. DOWNCOMER FLOW AREA EQUAL TO RISER FLOW AREA.

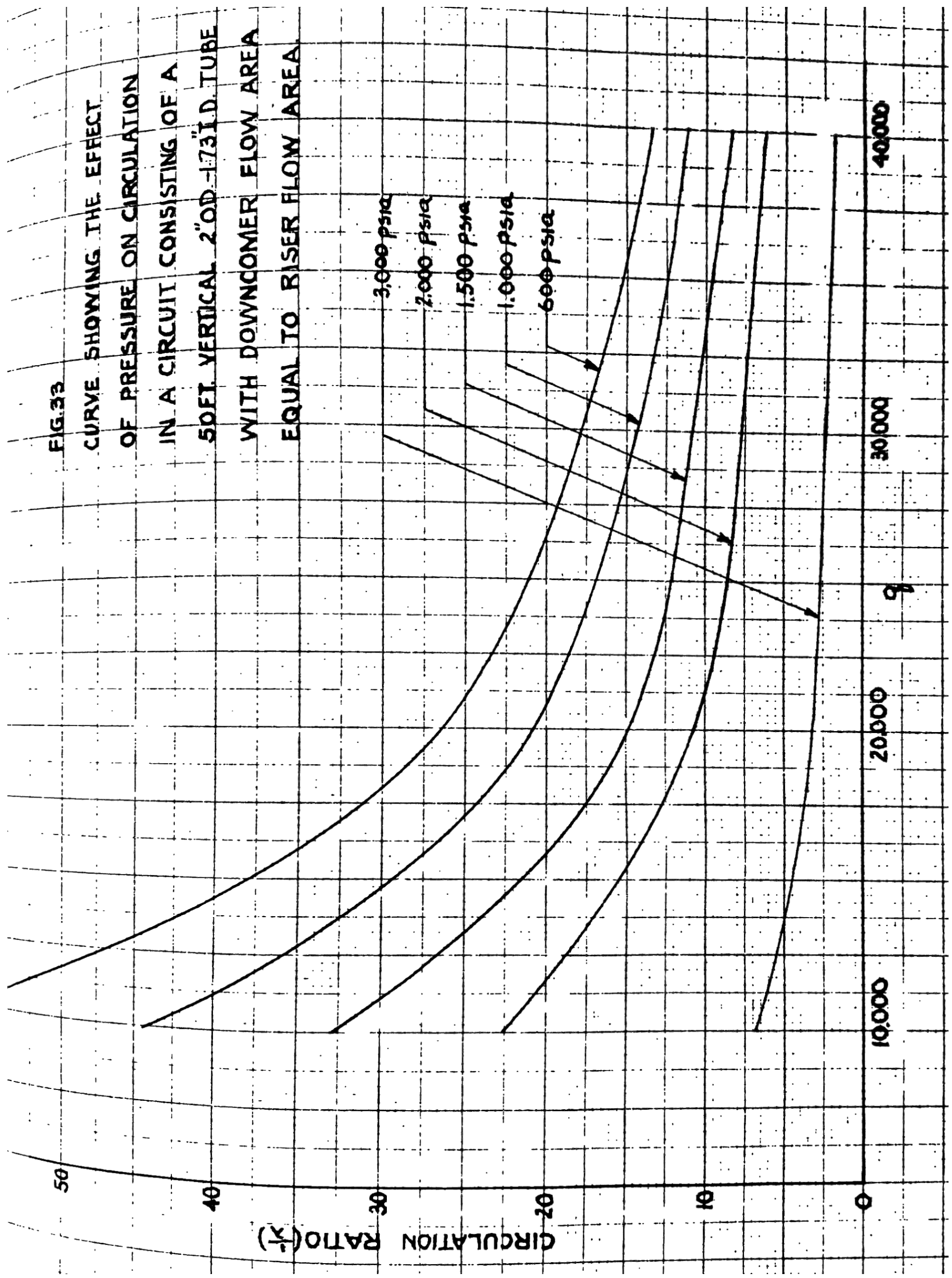


FIG.33
CURVE SHOWING THE EFFECT OF PRESSURE ON CIRCULATION IN A CIRCUIT CONSISTING OF A 50FT. VERTICAL 2" OD -.173" I.D. TUBE WITH DOWNCOMER FLOW AREA EQUAL TO RISER FLOW AREA
3,000 psia
2,000 psia
1,500 psia
1,000 psia
600 psia
CIRCULATION RATIO (1/x)
0
10
20
30
40
50
10,000
20,000
q
30,000
40,000

Fig. 33 clearly indicates the marked effect that an increase in pressure has on the circulation of the circuit chosen for this example. In conjunction with this curve, two things must be understood: for a given value of $q$, a different amount of steam is generated in the tube depending upon the pressure, and secondly, the amount of steam generated at a given pressure is directly proportional to the value of $q$.

The entire analysis and derivations made in this work were based on there being steam and water in the tube at all times. Therefore, they would not be directly applicable if superheated steam is formed in any section of the tube. If saturated steam at the critical pressure enters the circuit, it will, upon being heated, immediately become superheated steam. This follows from the fact that the latent heat of vaporization at the critical pressure is zero. Also, the density of saturated steam and saturated liquid at the critical pressure are equal to each other. Thus, at the critical pressure, the equations and analysis given here are not valid. However, if the curves of Fig. 33 are extrapolated, it would appear that a positive circulation can be maintained in the circuit. Lewis & Robertson,[(62)] in their analysis, also reached the conclusion that a positive and, in their case, an adequate circulation could be maintained at the critical pressure.

In their work, these authors, while discussing the effect of pressure on circulation, make the following statement: "… at 3,400 psi the density of water at 720 deg. F. is 17.04 lb. per cu. ft., and the total heat per pound (evidently Enthalpy) is 1,001.2 Btu." Using the conditions of pressure and temperature that these authors did in the above statement and the data in the Keenan & Keyes Tables, three discrepancies are to be found in this statement. The first discrepancy is that at this pressure and temperature, the steam is superheated and does not exist as water. Secondly, the density of the superheated steam is 12.95, and thirdly, the Enthalpy is 1024.1. It is also stated in this paper that the critical pressure is 3226 psi, while the Keenan & Keyes value is given as 3206.2 psi. The values of the physical properties of steam and water given in Appendix I of this paper also do not agree with the Keenan & Keyes data.

While the use of the incorrect values of the physical properties of steam and water makes the Lewis & Robertson calculations incorrect, their conclusions as to the effect of pressure on circulation appear to be valid on the basis of the example used in this work.

Recently, the paper by Martin Frisch and Robert A. Lorenzini[(98)] has been brought to the author's attention. In this paper, two of the topics discussed are the effect of pressure and steam entrainment in the downcomer on circulation. Figs. 34 & 36 have been reproduced from this work and are extremely interesting. The following is quoted directly from their work:

> The effect on the circulation of even small amounts of steam in the circulating water is appreciable at pressure below 1500 psi. This may be seen in figure, which shows the calculated circulation at various pressures from 500 to 3000 psia in the simple circuit illustrated. This is for a uniform heat absorption rate of 50,000 btu per square foot of projected surface per hour,

> and at three values of steam entrainment in the down-take water, namely the ideal zero, the maximum tolerable one percent, and an extreme value of two percent by weight.
>
> Fig. 34 shows that the total circulation in the circuit falls as the pressure increases and at each pressure as the amount of steam entrainment increases. At a pressure of 500 psia, 1 percent entrainment by weight causes the circulation to fall off by 32 percent and 2 percent entrainment by 48 percent. At a pressure of 1500 psia, the corresponding values are 11 and 19 percent. The effect of steam entrainment on the circulation decreases as the pressure increases and is of less importance than the pressure itself. Thus, the circulation at a pressure of 2500 psia is about one-half of that at 1300 psia and about one-third of the circulation at a pressure of 500 psia when steam in the down-take supply is zero.

Fig. 35, also from this work of Frisch & Lorenzini, shows the effect of steam entrainment on circulation in a Twin Furnace unit similar to the one shown in Fig. 23.

If the circulation ratios are taken from Fig. 33 based on the example chosen for this work at a value of $q$ and pressures equal to those in the Frisch & Lorenzini paper, it will be found that the conclusions that these authors reached as to the effect of pressure on circulation are essentially substantiated. While the author has not completely checked Fig. 34, calculations at several of the pressures and steam entrainment have been made using the equations developed in this work, and these values were found to agree with those shown in Fig. 34. The method used was to assume that a fictitious heated circuit preceded the riser circuit and had a value of $\frac{NL}{V_o}$ necessary to give hte amount of stress by weight assumed to enter the riser circuit. The rest of the computations were made by the direct application of the equations that were developed for the case of circuits in series.

Fig. 36 shows the results of special tests made to determine the effect of pressure on circulation .The trends shown by these curves verify the conclusions above.

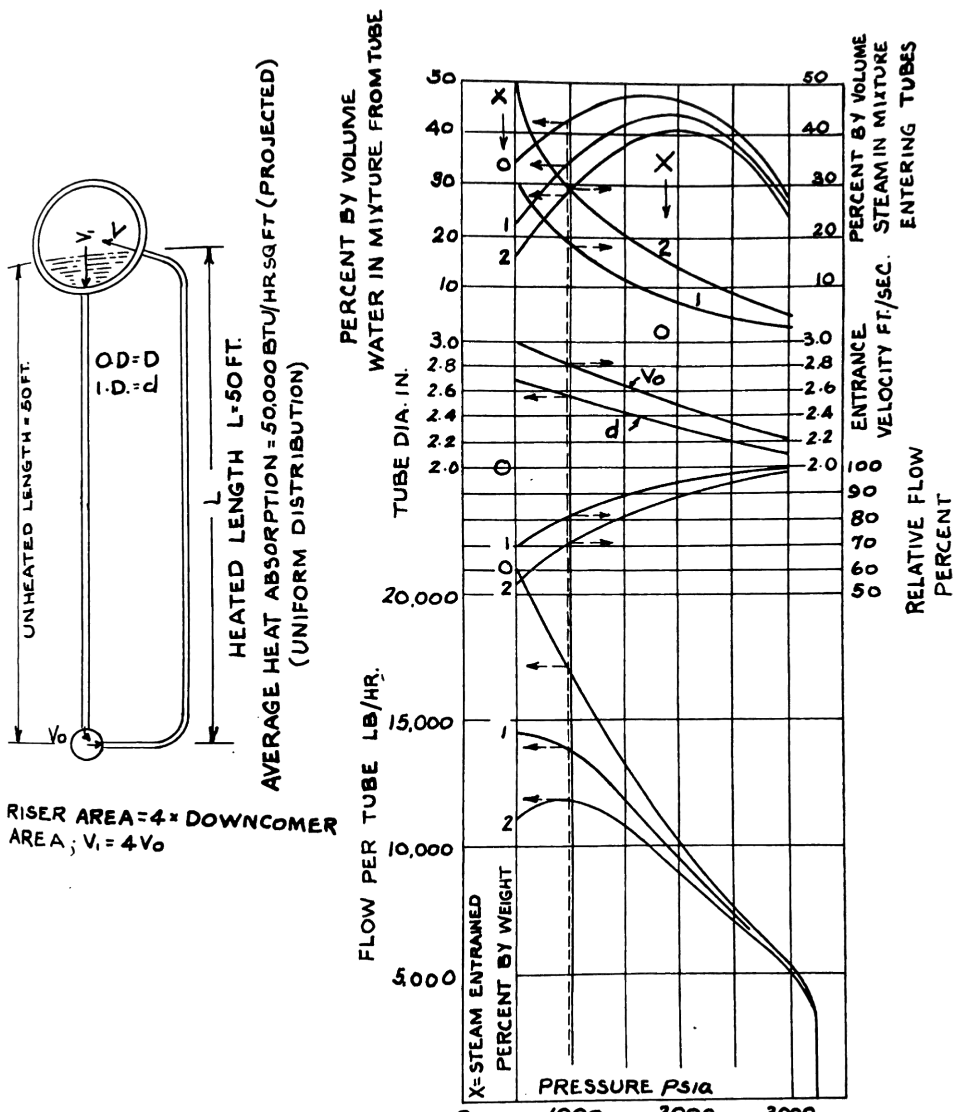



FIG. 34

EFFECT OF PRESSURE AND STEAM ENTRAINMENT IN DOWNCOMER ON CIRCULATION IN A SIMPLE CIRCUIT

-109-

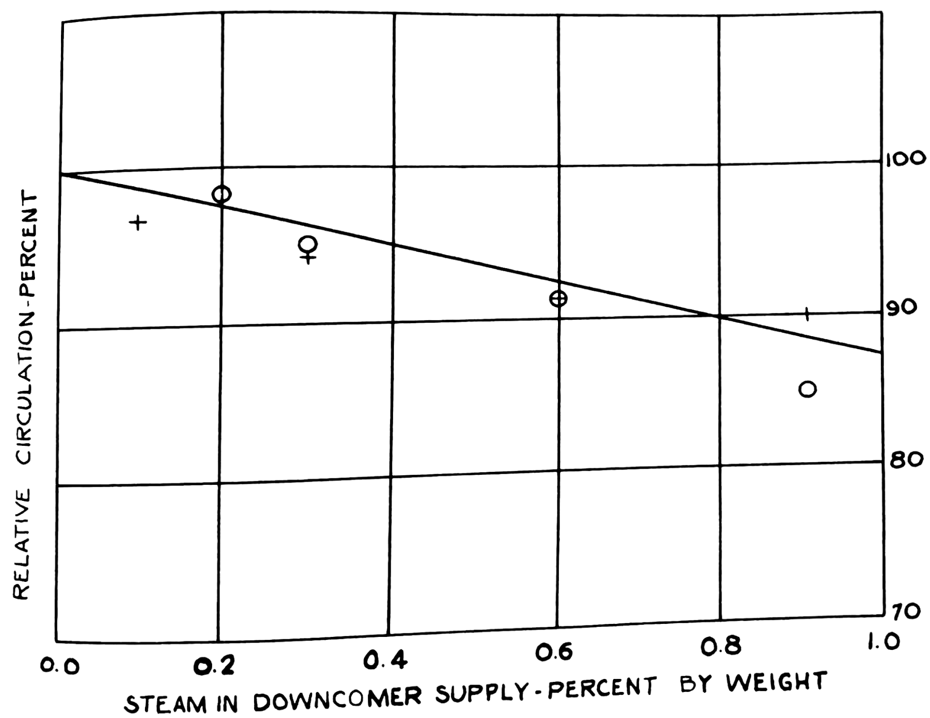



FIG. 35 EFFECT OF STEAM ENTRAINMENT ON CIRCULATION

O @ 400,000 LBS/HR. LOAD

+ @ 445,000 LBS/HR LOAD

PRESSURE - 1400 psia (APPROXIMATELY)

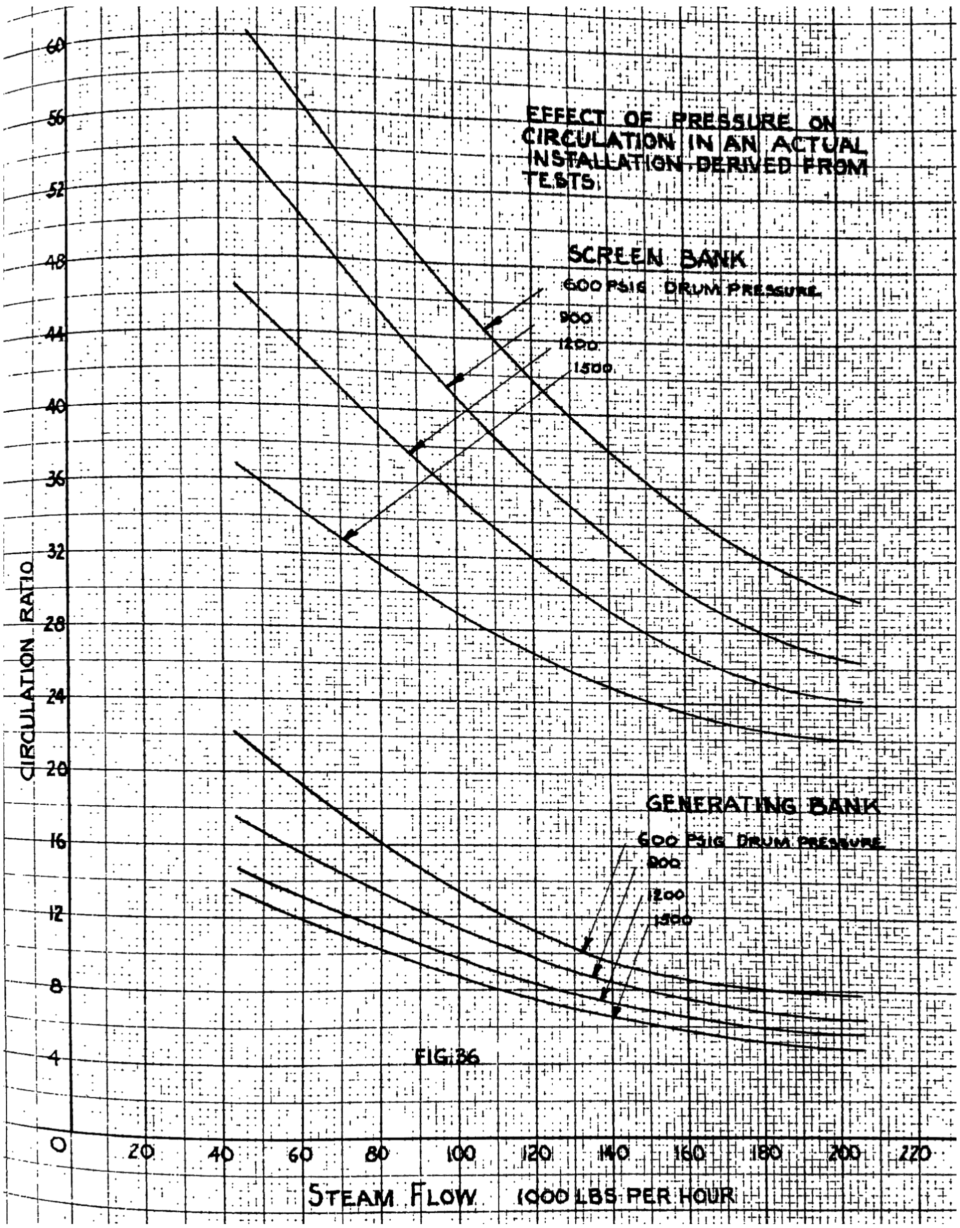

EFFECT OF PRESSURE ON CIRCULATION IN AN ACTUAL INSTALLATION DERIVED FROM TESTS.
SCREEN BANK
600 PSIG DRUM PRESSURE
900
1200
1500
GENERATING BANK
600 PSIG DRUM PRESSURE
900
1200
1500
CIRCULATION RATIO
STEAM FLOW 1000 LBS PER HOUR
0
4
8
12
16
20
24
28
32
36
40
44
48
52
56
60
20
40
60
80
100
120
140
160
180
200
220

FIG.36

## *Summary & Conclusions*

Equations and curves have been developed from fundamental hydraulic and thermodynamic considerations to enable the complete circulation characteristics of a steam generating unit to be predicted. In these derivations, various assumptions about the distribution of heat absorption, friction factor, the effect of bubble slip, and the homogeneity of the mixture were made according to the best available data at the time. While the curves developed are strictly true only to the extent that these assumptions are true, the equations can be readily modified to account for any change in these assumptions when new data becomes available.

With these assumptions, the circulation characteristics of an actual unit were computed by use of both the curves and equations, and good agreement was found with test values. The use of the curves materially decreases the time needed for such computations and yields results that are essentially the same as those obtained by direct application of the equations. The effect of pressure on circulation and the possibility of flow reversal in a heated tube was also briefly discussed because of their importance in the design of natural circulation boilers.

The title of this work was purposely called "The Computation of Natural Circulation in Large Boilers" for very good reason. In the course of the author's work with the Foster Wheeler Corporation, he has had occasion to try to correlate the circulation characteristics of small Marine units on which test data was available by use of the methods of this work. In such units it is usual to find that the greater part of the steam generation occurs in the Boiler Bank, and a comparatively small amount of generation occurs in the Screen Bank and Water Walls. It has been found that the proper choice of friction factor and distribution of heat absorption is much more critical in these units than in a large boiler. An error of one foot of saturated liquid in a unit fifty feet in height causes a much smaller error in the final values than an equal error in a unit only ten feet high.

In almost every boiler built some form of internals or baffling is placed in the steam drum to insure steam free of impurities and decrease carry-over. These internals cause an added pressure drop in addition to those already discussed and, to be correct, this pressure drop should be taken into account in the circulation computations. However, as has been pointed out above, the percentage error incurred in a large boiler is much less than that incurred in a small unit if this pressure drop is neglected as was done in Section III of this work. The determination of the pressure drop in internals by tests is extremely difficult and practically no data is available in the literature to enable one to compute it.

For the reasons just stated, this work has confined itself to large boilers. This does not mean that the derivations of Section II are not valid for small units; it means that further refinement of the assumptions must be made and that further tests are necessary before the same degree of correlation that has been found possible in large boilers is possible in smaller units.

Throughout this work, each assumption has purposely been pointed out to show where further tests are necessary. In the past, it has been found most expedient to run many of the tests with mixtures of air and water at or near atmospheric pressure. While much valuable information has been obtained by this method, there is a definite lack of knowledge concerning the mechanics of certain phases of the processes that occur during the transformation of water to steam at elevated pressures and temperatures. A test program to make such determinations as necessary to fill in the gaps in our present knowledge would be invaluable. Freon 12 has the physical properties at 200 psi that very nearly approach those of steam and water at 1400 psi. Because of the nearness of these properties and the relative ease in handling pressures of 200 psi compared to 1400 psi, it is suggested that a test program utilizing Freon 12 could be carried out to provide the necessary information on pressure drops.

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*Note: A large part of this bibliography is taken from the excellent one given in the work by Nothman.*


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