A MATHEMATICAL INVESTIGATION OF THE EFFECT OF TUBE SPACING, EXCESS AIR, AND BRIDGEWALL AND STACK TEMPERATURES UPON PIPE STILL DESIGN

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ABSTRACT

A MATHEMATICAL INVESTIGATION OF THE EFFECT OF TUBE SPACING, EXCESS AIR, AND BRIDGEWALL AND STACK TEMPERATURES UPON PIPE STILL DESIGN

by William Vere D'A. Saunders

A mathematical model is presented for use in the design of petroleum furnaces. This model includes the requirements of heat load and pressure drop. A program for the solution of the design equations has been prepared, and solutions were obtained for specific furnace requirements by varying tube spacings, per cent excess air, and bridgewall and exit stack temperatures.

In order to predict pressure drop in the presence of two-phase flow an equation similar to the Fanning equation was developed. It is difficult to judge the reliability of this equation because data was not available.

Furnace designs obtained were correlated in terms of the distribution of heat loads to radiant and convection sections, and were plotted as functions of the bridgewall and exit stack temperatures. Radiant heat transfer rates were related to the fraction of the total heat input absorbed in the radiant section. This correlation included the effect of tube spacing.

The principal significance of excess air is its effect on the fraction of total heat input absorbed in the radiant section. At a fixed bridgewall temperature and normal conditions of operation, this fraction can be increased by a factor of approximately 1.5 with a 50 per cent decrease in the excess air.

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INTRODUCTION

The design of modern petroleum furnaces is tending toward larger and more efficient units, sufficiently flexible to adapt to wide variations in physical characteristics of the charging stock, and more responsive to accurate control of the finished product.

It is the current trend in design practice to increase the ratio of heat receiving surfaces to refractory surfaces and to increase the radiant heat transmission rates. These increases affect the size and duty of the convection section and, unless a suitable distribution of heat is obtained throughout the furnace, uneven heating and inefficient units result. There is also a tendency to increase the fraction of vapor discharged from the still to the fractionating tower because increases in efficiency are obtained under these conditions.

It is the purpose of this investigation to study the effect of the tube spacing, percent excess air, and bridgewall and exit stack temperatures and to propose a suitable means of allowing for their effects in the design of tube still heaters.

The successful design of a tube still heater must include the requirements of heat load to the furnace and pressure drop of oil through the tubes. This involves the calculation of (1) flash vaporization data to determine the sensible and latent heat required as well as the fractions of liquid and vapor present in the tubes; (2) suitable heat transfer rates in the radiant and convection sections; (3) the tube length and diameter best suited for the type of still and discharge conditions required.

The mechanisms of heat transfer in combustion chambers are complicated by the phenomenon of radiant heat transmission; as a result, the design of tube still heaters has developed on an empirical basis. Based on the fundamental principle of radiation postulated in the Stefan-Boltzmann equation, $E_b = \delta T^4$, and with the aid of certain generalizations, an equation was developed by Lobo and Evans (1),

$$q = \delta \, [T_1^{\,4} - T_2^{\,4}] \, \alpha A_{cp} \psi$$

which is extensively used in the design of heaters.

Calculations for pressure drop were made using the Fanning equation, with a correction for added frictional loss caused by continuous vaporization of crude throughout the still tubes.

By arbitrarily dividing the tubes into a number of zones, the problem of obtaining heat balances and pressure drops over these sections can be solved but necessitates an elaborate trial-and-error calculation. These calculations not only involve the simultaneous solution of heat transfer equations, but also the evaluation of sixty third degree and seven fourth degree polynomials. This has been accomplished with the aid of a digital computer. A program for the computer was prepared whereby changes in tube spacing, per cent excess air, and bridgewall and exit stack temperatures could be made in a design.

The results of these calculations have shown that the distribution of total heat loads to both sections of petroleum furnaces is an important factor in determining their design. Various distribution ratios can be obtained by changing the bridgewall and exit stack temperatures and the per cent excess air. However, the choice of a specific ratio can only be made after the economic permissible heat transfer rates have been determined.

TUBE STILL HEATERS

One of the most important commercial applications of radiant energy transmission is encountered in petroleum refinery furnaces. These furnaces, or tube still heaters, are extensively used in atmospheric or vacuum crude distillation, high temperature gas processing, and thermal cracking, as well as in various heating, treating, and vaporization services.

History and Development

"The early stills used by the oil-refining industry were of the simplest kind. Holding but a few barrels, they were set directly over a coal or tar-fired furnace. The ascending vapors were condensed in a coil submerged in water with no attempt at fractionating further than the gravity indication of the overhead condensate." (2) The stills were generally potshaped and, owing to their construction, were often called shell stills.

In the early years of the petroleum industry, progress was slow. The only attempts made to improve the design of the shell still were increases in its size, leading later to the "cheese-box" still. "The cheese-box still with its eventual capacities up to 1,000 barrels replaced the shell stills at a number of refineries, starting in the late sixties. These cylindrical stills usually had what was termed a 'vapor chest' connected to the still by vertical pipes. The still had a dome-shaped top and a double curved steel plate bottom. The still was supported by a series of arches." (3)

The increased heat efficiency and capacity of these stills reduced the costs per barrel of throughput in comparison with the earlier shell stills. However, the small effective heating surface and the large volume of charged stock caused them to be very inefficient and to have low rates of heat input.

With the advent of the fractionation art and the introduction of cracking operations, it became necessary to construct heaters that could withstand the high temperatures and pressures of the cracking process. These requirements resulted in a continuous operation permitting the greater use of heat exchange and the steady improvement of design and operating efficiency.

An early attempt at continuous operation involved construction of a battery of shell stills connected in series, with the first still emptying into the second and then in series on to the number of shell units in operation.

"Comparatively successful application of tubular heaters on a small scale for dehydration and refining of emulsified oils led to the gradual adoption of tubular heaters for general refining purposes and eventual substitution for shell heater for large-scale refining operations." (4) The earlier tubular heaters were similar in design to the shell stills, but the stills were displaced by a bank of tubes. This improvement, in some instances, doubled the heat transfer rates from 3000 Btu/ft^2-hr with the shell type batch operation to 5000 or 6000 Btu/ft^2-hr with the tubular heater.

Increases in heat load to the furnace led to localized heating of the tubes and created zones with excessively high transfer rates of 15,000 to 20,000 Btu/ft^2-hr, called "hot spots." Coke formation and tube failures occurred at these points. As these furnaces were designed to obtain the major portion of heat transfer by convection, hot spots were attributed to radiation from flames in the fire box. Furnaces were then built with tube banks shielded from the flame by perforated or solid walls to protect the tubes from its radiation. This resulted in a more uniform heat distribution within the furnace, with higher overall heat transfer rates.

It was then discovered that, even with the shield, the tubes first exposed to the combustion products became easily overheated. Unless the gas temperature was reduced below a certain minimum, hot spots would still occur. This reduction of gas temperature was accomplished either by diluting the fuel with excess air and then with recirculated flue gas, or by installing tubes in the combustion chamber to cool the hot gases by absorbing radiant energy from the flame, the gases, and the refractory surfaces of the chamber.

The basic construction of most tube-still heaters is similar. Such units almost always consist of a radiant section, where the major portion of the heat supplied to the process stream is by radiation, and a convection section, where the major portion of the heat is supplied by convection from the gaseous combustion products.

Heaters vary widely in shape and size, and are designed to meet various requirements for such variables as charging stock, heat distribution, thermal efficiency and time-temperature effect. Because thermal decomposition is a rate process, the degree of decomposition is a function of both temperature and time and is described as the time-temperature effect.

Petroleum heaters have been divided (5) into three main groups according to the amount of decomposition obtained.

(1) Heaters used only for heating, with little or no decomposition;

(2) Heaters where, in addition to heating, substantially all of the decomposition desired for the refining process is obtained;

(3) Heaters where only partial decomposition is obtained in the heater, the remainder in the reaction chambers or soaking drums which are usually not heated externally.

Heaters of the first group are employed in operations where no chemical change is desired in the charged stock, as for nondestructive distillation processes. These heater are designed to obtain a minimum time-temperature effect with a maximum temperature.

Heaters of the second group are used primarily in cracking operations where the decomposition of the stock takes place within the heating coils. These heaters are designed to give maximum time-temperature effect at the highest operating temperature allowable.

Heaters of the third group, used for thermally sensitive residual cracking stocks, must be designed for a time-temperature effect that will permit the highest outlet temperature without excessive decomposition from soaking within the heating coils.

A number of typical furnace arrangements are shown in Figures (1) and (2) to illustrate diagrammatically the arrangement of tubes and the direction of fluid flow in the basic types of tube still heaters.

Figure 1 shows a typical box type furnace fired from the end walls. Radiant tubes cover the side walls, roof, and bridgewall (partition between radiant and convection section) surfaces. The tendency in modern furnace design is to fill the radiant section with cold tube surfaces; to accomplish this tubes may also be placed on the floor surfaces.

In cracking operations, oil is preheated in the upper and lower rows of the convection bank then passed through the radiant tubes. After reaching an elevated temperature conducive to the cracking process, the oil is passed through a large number of convection section tubes wherein it is maintained at a high temperature for a sufficient time to accomplish the desired degree of cracking.

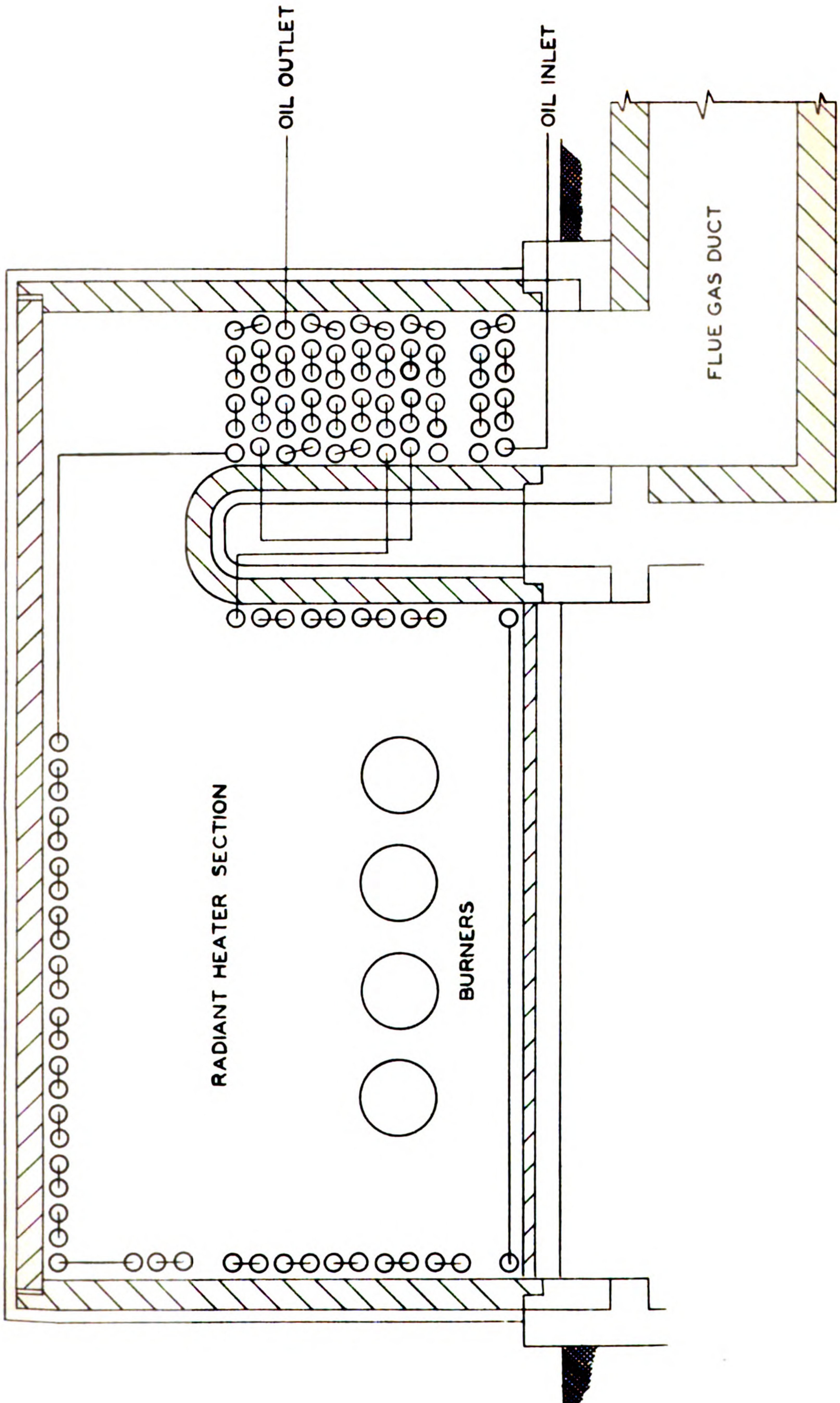
OIL OUTLET
OIL INLET
FLUE GAS DUCT
RADIANT HEATER SECTION
BURNERS

FIGURE 1

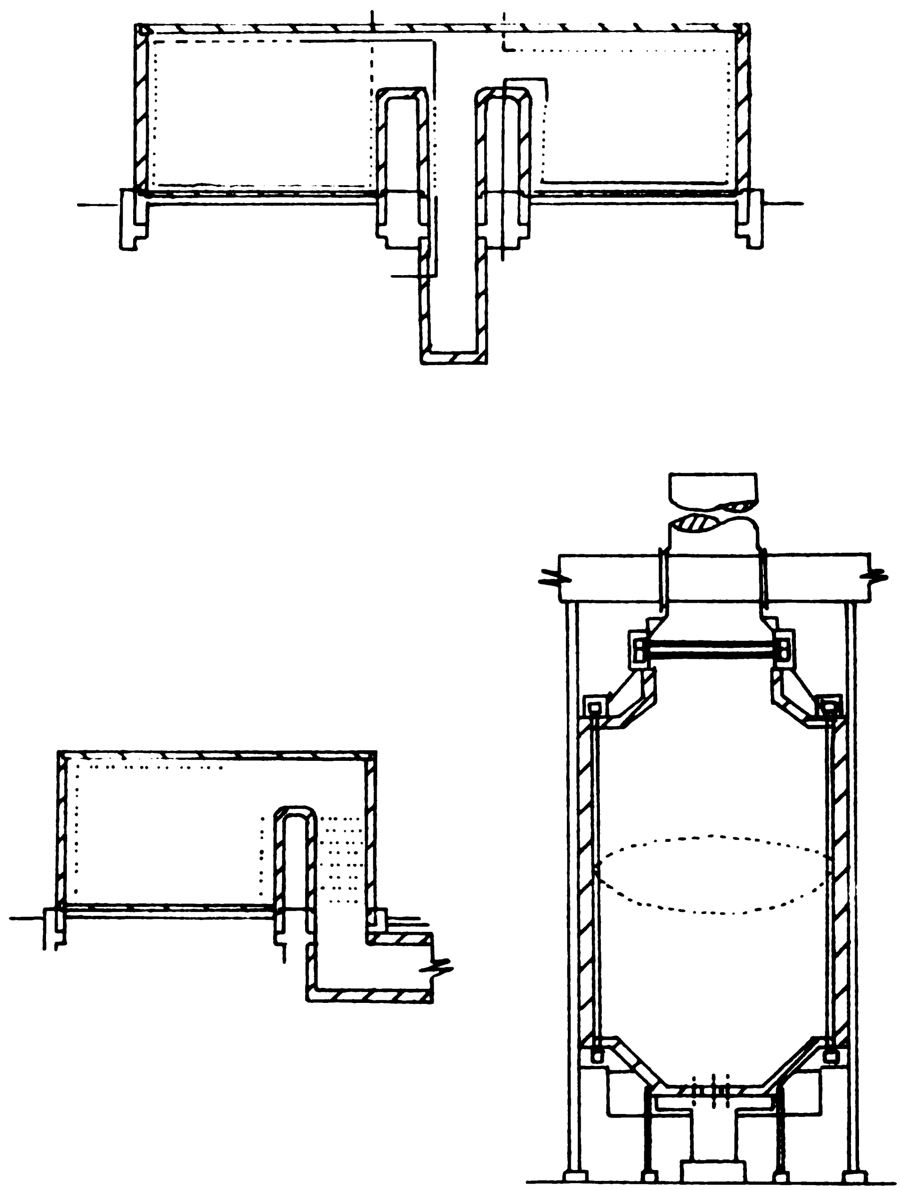

FIGURE 2. Types of Pipestill Heaters.

DEVELOPMENT AND SELECTION OF EQUATIONS

Evaluation of the rate of heat transmission to the cold surfaces in tube still heaters is accomplished by considering the extent to which each mechanism of heat transfer influences the overall rate.

Transfer of heat energy liberated by the chemical union of molecules in the flame takes place first in the radiant section to the surrounding tubes primarily by radiation though some convection occurs. As the gaseous products of combustion progress through the convection section, heat transmission is principally caused by the mechanism of forced convection accompanied by small amounts of gas radiation.

Thermal energy transferred within the furnace enclosure must be equated to the change in enthalpy of the entering and exit streams. To accomplish this, the following equations must be obtained: heat transfer by radiation, heat transfer by convection, and heat balances about the oil.

Radiation

Radiation in a combustion chamber originates from three distinct sources (6):

(1) the chemical union of molecules in the flame,

(2) the hot products of combustion, and

(3) the luminosity or soot content of the flame.

The magnitude of radiation emitted from the first source is dependent on the composition of the fuel, the maximum temperature attained, and the absorbing characteristics of the flame for its own radiation. However, in muffle furnaces where the flame is shielded from the surfaces of the combustion chamber, heat from this source is transferred to the combustion products by conduction and convection.

Radiation of greatest magnitude originates from the combustion products and is dependent on composition, temperature, and shape and size of the gas mass. Of the gases comprising the combustion products, carbon dioxide, carbon monoxide, the hydrocarbons, and water vapor are the only ones with emission bands of sufficient energy to merit consideration. Gases with simple symmetrical molecules, such as N_2, H_2, and O_2, which also comprise the total gas mass, show no absorption bands in the region of importance in radiant heat transmission. Moreover, carbon monoxide and hydrocarbons are present in such small amounts as to be negligible compared with water vapor and carbon dioxide. Finally, the third source of radiation from the flame, its soot content, is dependent on the degree of combustion and the design of the combustion chamber.

Using data obtained from investigations on the infrared spectra of carbon dioxide and water vapor, Hottel (7) has presented charts for use in calculating the quantity of heat transmitted from these gases. He has also shown that the energy emitted from a gas mass to a unit area of bounding surface is a function of the gas and surface temperatures, the absorptivity of the surface, and the product PL, where P is the partial pressure in atmospheres of the radiating constituents, and L is the average length of a blanket of flue gas in all directions for each of the points of the bouding surface of the furnace. Values of L for furnaces of various shapes were determined by Hottel and Table I presents a digest of these values for furnace calculations.

Incident radiation is not completely absorbed by its ultimate heat receiving surfaces immediately but is reflected and absorbed in an infinite series of interchanges between source and surface. Consequently, radiant interchange between the surfaces of an enclosure must involve consideration of the view the surfaces have of each other as well as their emitting and absorbing characteristics.

The absorptivities of bodies are generally dependent on the wavelength of incident radiation and also on the factors affecting their emissivities. Absolute values of the emissive power of bodies are not readily obtainable, however, the ratio of the actual emissive power to the black body emissive power,

TABLE I

Mean Length of Radiant Beams in Various Gas Shapes

Dimensional Ratio (length, width, height in any order)	L_B
Rectangular Furnaces	
1. 1-1-1 to 1-1-3	$2/3 \sqrt[3]{\text{Furnace Volume}}$
1-2-1 to 1-2-4	
2. 1-1-4 to 1-1-∞	1 x smallest dimension
3. 1-2-5 to 1-2-8	1.3 x smallest dimension
4. 1-3-3 to 1-∞-∞	1.8 x smallest dimension

defined as the emissivity, has been determined for many materials and data are presented in most textbooks of heat transfer. In systems such as furnaces composed of walls and pipes, it becomes difficult to evaluate the manner in which radiant energy falls on these surfaces. The next flux between source and surface occurs by a complex process involving multiple reflection from all surfaces forming the enclosure. The new concept necessary here is F, and has been defined by Hottel (8) as the direct interchange factor, dependent on the angle factors between the refractory surfaces and the surfaces surrounding it, together with the emissivities of the source and sink surfaces.

Since heat receiving surfaces or heat sinks in most industrial furnaces are composed of a multiplicity of tubes disposed over walls, roof, and floor of the combustion chamber, it is necessary to evaluate the effective heat transfer area. The development of Hottel, almost exclusively used in design work, assumes that the heat source is a radiating plane parallel to the tube row. The effectiveness factor, α, is the factor by which the surface of a plane replacing the tube row with assumed emissivity of 1.0, must be multiplied to obtain the equivalent cold plane surface. For a detailed development of α , reference should be made to Hottel (9). Figure 3 presents values of α for radiation to single rows of tubes with refractory behind them.

In view of the complexity of the problem, numerous investigators have correlated furnace performance by means of empirical and semitheoretical equations. The most acceptable of these, is the semitheoretical equation proposed by Lobo and Evans (1). Using an equation of the Stefan-Boltzman type in correlating data from 85 tests on 19 different petroleum furances, they developed the following equation:

$$q = \delta\ (T_g^{\ 4} - T_s^{\ 4})\ \alpha\ A_{cp}\psi + h_c A'_r (T_g - T_r) + h_c A_c (T_g - T_s) \qquad (1)$$

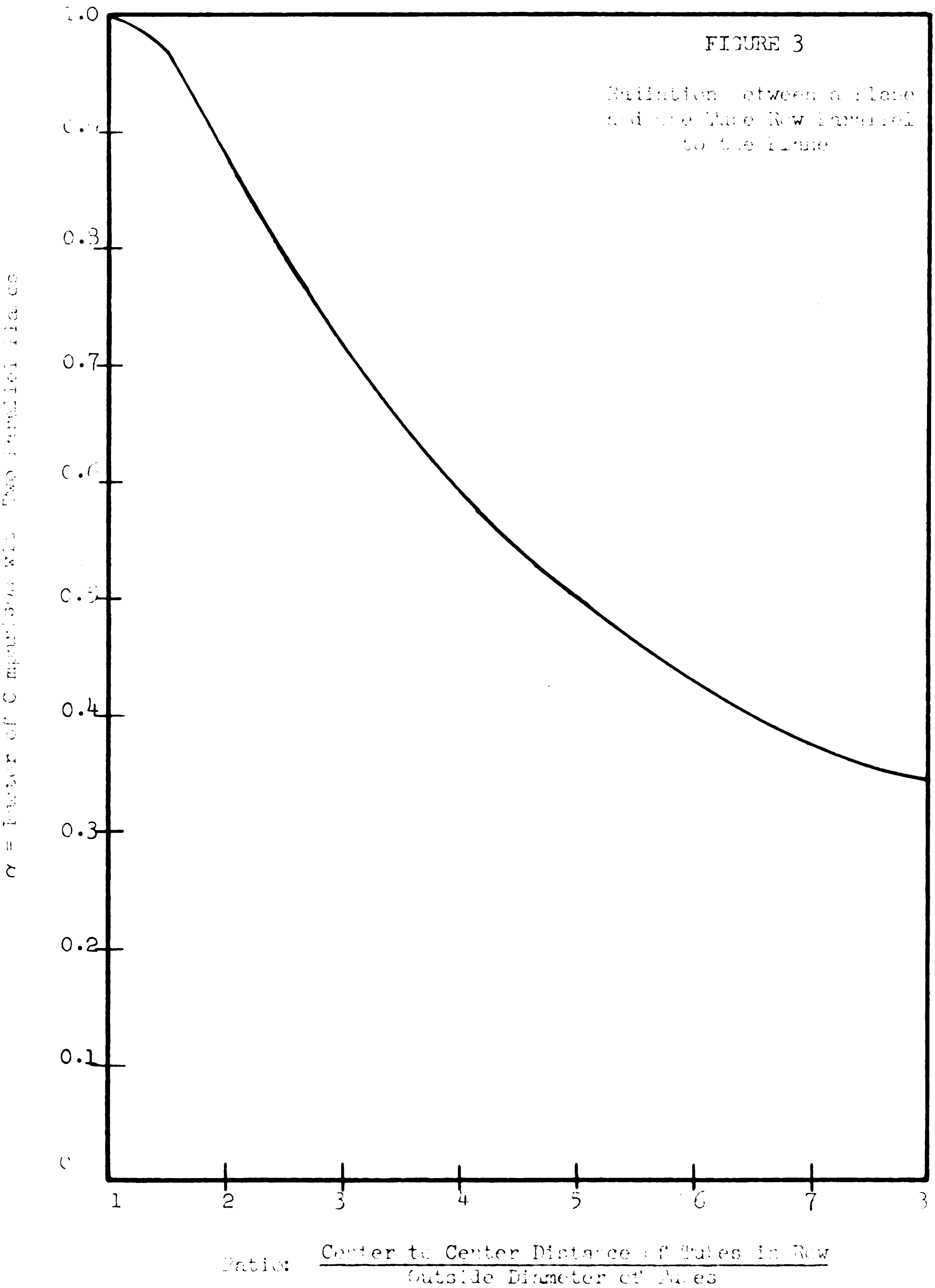

FIGURE 3

Radiation between a Flame and the Tube Row Parallel to the Flame

In this equation the direct interchange factor, F, has been replaced by an overall exchange factor, ψ. ψ includes, in addition to direct interchange, the contributions due to multiple reflection at all surfaces as well as such contributions by reradiation from zones at which the net radiant-heat transfer at the wall surface is zero.

Since both the external losses from the furnace and the net heat transferred to the refractory by convection, given by the term $h_c A'_r (T_g - T_r)$, are usually small, the two may be assumed equal without appreciably affecting the results. Equation (1) may be rewritten

$$q = \delta \, [T_g^{\,4} - T_s^{\,4}] \alpha A_{cp} \psi + h_c A_c (T_g - T_s) \tag{2}$$

By making the following assumptions, Lobo and Evans further simplified their equation:

1. The convection coefficient lies normally between 2 and 3 Btu/hr sq ft - °F;
2. In most furnaces A_c equals $(2\alpha A_{cp})$ approximately;
3. The overall exchange factor ψ has a value of about 0.57.

Therefore, the terms h_c and A_c in Equation (2) can be expressed in terms of and ψ, thus:

$$\frac{h_c A_c}{\alpha A_{cp} \psi} = \frac{(2)(2)}{(0.57)} = 7.0$$

or

$$h_c A_c \; (T_g - T_s) = 7(\alpha A_{cp} \psi)[T_g - T_s]$$

Making the substitution in Equation (2),

$$\frac{q}{\alpha A_{cp} \psi} = \delta [T_g^{\,4} - T_s^{\,4}] + 7 \; (T_g - T_s) \tag{3}$$

In the combustion chamber T_g, the mean temperature of the hot gases in the furnace and the temperature of the exit gases will undoubtedly differ. However, the assumption of complete mixing in the furnace and that T_g could be replaced by the exit gas temperature was justified by satisfactory results. The exact evaluation of ψ is tedious and complicated, however, their development included a plot of ψ versus the ratio $\frac{A_r}{\alpha A_{cp}}$ with the flame emissivity as a parameter. Results indicate that the

ψ plot represents an accurate and simple method of simultaneously allowing for the effect of flame emissivity and the amount of refractory surface present.

Hottels' charts giving the values of the radiant heat transfer flux due to CO_2 and water are most conveniently used in calculating the emissivity of the flame. The radiant flux of H_2O and CO_2 are additive, although a small correction must be included to allow for the influence of one type of molecule with radiation from the other. The flame emissivity is given by the equation:

$$\epsilon_g = \frac{(q_c + q_w)_{T_g} - (q_c + q_w)_{T_s}}{(q_b)_{T_g} - (q_b)_{T_s}} \tag{4}$$

Radiant heat transmission in the convection section is particularly significant to the uppermost tubes or "shield tubes" in the convection bank where the temperature of the gases is still high. In spite of the small beam length, Monrad (10) has shown that radiation may account for 5 to 30 percent of the total heat transfer in the convection section.

Evaluation of the radiant heat coefficient was accomplished by adapting the method of Lobo and Evans and the simplified charts provided by Hottel. The mean length of the radiant beam for exchange between tubes, given in Table I, is based on the center to center spacing ($\not{c}$-$\not{c}$), and the outside diameter of the tubes (D_o).

$$L_B = 0.4\ [(\not{c}-\not{c}) - 0.567]\ D_o \tag{5}$$

$$q_{rc} = \epsilon_s\ [(q_c + q_w)_{T_g} - (q_c + q_w)_{T_s}]\ A\ \frac{(100 - \%)}{100} \tag{6}$$

since

$$h_r = \frac{q_{rc}}{A\,\Delta T}$$

$$h_r = \epsilon_s\ \frac{[(q_c + q_w)_{T_g} - (q_c + q_w)_{T_s}]}{T_g - T_s}\ \frac{(100 - \%)}{100} \tag{7}$$

where the emissivity of the surface ϵ_s is assumed to have a value of 0.95. (11)

Convection

Heat transfer by convection varies widely with gas velocity and size of gas passage, somewhat with temperature of the gas, and very little with gas composition. Although there are numerous relationships available for obtaining convection coefficients, little work has been done to develop a satisfactory relationship for furnace design work. The empirical equation by Monrad (10) is the only comprehensive formulation of convection transfer rates. For direct convection from the gases he proposes the relationship:

$$h_c = \frac{1.6\ G^{2/3}\ T^{0.3}}{{D'_o}^{1/3}} \tag{8}$$

The equation applies to any conventional arrangement of the tubes in the convection section. However, the coefficient h_c is the pure convection coefficient and it does not include radiation from the hot gases or from the walls. Monrad has made a study of these factors. The first of these is designated as h_{rg} or the coefficient of heat transfer from the gas by radiation. A formula for the evaluation of this coefficient was presented in the previous section on radiation.

In his calculations, Monrad also included a correction for the increased thickness of the gas layer at the top of the tube bank. His assumption was that this radiation could be approximated by that of a plane equal in area to the top tube bank at PwL = PcL = 1.0, between the temperature of the gas above the bank and the temperature of the tube. He reasoned that since radiation had already been calculated for PwL and PcL based on the center to center spacing and the tube diameter (h_{rg}), the added gas radiation would be equivalent to that at the top gas temperature between PcL = PwL = 1.0 and PcL and PwL based on the center to center spacing and the diameter; consequently, the correction:

$$h'_{rg} = h(\text{at PL} = 1) - h_{rg}$$

This however, appears to be too severe a correction as PL values for the gas layer above the bank could hardly approach a value of unity. Water vapor, the radiating constituent of highest concentration in the gas mass rarely exceeds a partial pressure of 0.20 atms. Consequently, the average beam length would necessarily have to be greater than 50 feet (for lower concentrations of water vapor) to cause values of PL = 1.0. For this reason, it was concluded that sufficiently accurate results would be obtained without including this correction.

The area of the walls surrounding the tubes comprise a fairly large fraction of the tube area. These walls pick up heat from the gases by convection and radiation, and reradiate to the tubes by black body radiation according to Stefans' Law. With the assumption that factors such as reabsorption of heat and the differences in heat transfer coefficients to the wall and tubes are negligible, the following equations were presented:

$[h_c + h_{rg}][T_g - T_w]\ A_w$ = heat to walls

$= [h_{rb}][T_w - T_t]\ A_t$ = heat from the walls = heat to tubes from walls

Therefore the per cent increase in heat absorption by tubes above that received directly:

$$\frac{h_{rb}\ [T_w - T_t]\ A_w \times 100}{[h_c + h_{rg}][T_g - T_t]\ A_t} = \frac{h_{rb}\ [T_w - T_t]\ A_w \times 100}{A_t \left\{ [h_c + h_{rg}][T_g - T_w] + [h_c + h_{rg}][T_w - T_t] \right\}} \qquad (9)$$

h_{rb} may be approximated by:

$$h_{rb} = 0.00688\ \epsilon_s \left[\frac{T}{100}\right]^3 \qquad (10)$$

$\epsilon_s = 0.95$

T = Temperature of the tube surfaces

The complete coefficient of heat transfer in the convection section was computed from the preceding items as follows:

$$h_u = \frac{(100 + \%\ \text{wall effect})}{100}\ (h_c + h_{rg}) \qquad (11)$$

In considering radiation from the walls to the individual rows of tubes in the bank, the correction for the wall effect becomes less pronounced as the gases cool on their way to the stack. The wall effect then, will be of most significance to the tubes in the shield section. If a correction is made for the shield section (suppermost rows surrounded by gas at a temperature of 1300°F or above) only, it will be necessary to evaluate the fraction of the wall surface that will "see" the tubes in the section. An approximation was made by assuming that the area of the walls surrounding the shield tubes would be the only surfaces that could see that section.

Corrections for the wall effect and gas radiation were made only for tubes in the shield section, whereas the effect of radiation was neglected in calculations for the film coefficient for the rest of the tubes in the convection bank.

In general, the coefficient of heat transfer on the gas side is controlling; although the resistance of the liquid film may be assumed negligible, approximate values were estimated using a modified Dittus-Boelter (11) equation.

$$\frac{h_i D_i}{K} = 0.027 \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{C\mu}{K}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad (12)$$

Based on the outside pipe diameter, the overall coefficient of heat transfer (U) is given by:

$$U = \cfrac{1}{\cfrac{1}{h} + \cfrac{D_o\,(D_o - D_i)}{K_m\,(D_o + D_i)} + \cfrac{D_o}{D_i h_i}} \qquad (13)$$

and the heat flux to the oil evaluated using the following equation:

$$q_c = U\,(\pi D_o\,\Delta L)(\Delta t) \qquad (14)$$

Heat Balances

The energy balances made throughout these calculations involving the physical properties of the crude oil, can only be fair approximations due to the complex nature of the hydrocarbon mixture. Calculations based on the heat content of the charged stock will involve some error since the difficulty of obtaining the accurate specific heat of the oil, its vapor and its latent heat of vaporization is quite large. The difficulty in estimating external heat losses from the furnace is another source of error.

The liquid heat content of Mid-Continent source crude oils was calculated using the equation obtained by Weir and Eaton (13).

$$(H - Ho)_L = (15d - 27) + (0.811 - 0.465d)t + 0.000290t^2 \qquad (15)$$

H = total heat continent above 32°F.

Ho = heat continent at 32°F.

t = temperature °F.

d = sp. gr. of material at 60°F.

The heat content of liquids do not vary appreciably with pressure, therefore, Equation (15) was used to calculate the liquid heat content over the entire range of pressures encountered in the pipe still. Like the liquid data, Weir and Eaton found it possible to incorporate the vapor heat content versus temperature relationship in a single recommended equation.

$$(H - Ho)_V = (215 - 87d) + (0.415 - 0.104d)t + (0.000310 - 0.000078d)t^2 \qquad (16)$$

It is necessary to include a correction for the variations with pressure. Using the equation of state proposed by Linde to predict PV data, and the constants as obtained by Baklke and Kay (14) an equation for the total heat content of the vapor was developed.

$$PV = AT - P\frac{(C + EP)}{T^3} + P(D + FP) \qquad (17)$$

$$V = \frac{AT}{P} - \frac{C + EP}{T^3} + D + FP \qquad (17b)$$

T = °R

V = ft^3/lb

P = $\#/in^2$

A, C, D, E, and F = constants

$$d_H = C_p dT - [T(\frac{\partial V}{\partial T})_P - V]\ dP \qquad (a)$$

Differentiating Equation (17)

$$P(\frac{\partial V}{\partial T})_P = A + \frac{3\ CP}{T^4} + \frac{3\ EP^2}{T^4} \qquad \text{or}$$

$$T(\frac{\partial V}{\partial T})_P = \frac{AT}{P} + \frac{3C}{T^3} + \frac{3EP}{T^3} \qquad (b)$$

Substituting (b) and (17b) into (a) and simplifying

$$d_H = C_p dT - [\frac{4C}{T^3} + \frac{4EP}{T^3} - D - FP]\, dP$$

integrating

$$H - Ho = \int_{32}^{T} C_p dT + \frac{1}{9331.7} [DP + \frac{FP^2}{2} - \frac{4CP}{T^3} - \frac{2EP^2}{T^3}] \qquad (18)$$

and

$$\int_{32}^{T} C_p dT = (215 - 8Td) + (0.415 - 0.104d)t$$
$$+ (0.000031 - 0.000078d)t^2$$

The constants, as obtained by Bahlke and Kay are:

$A = 157$

$C = 7234 \times 10^7$

$D = 20$

$E = 102 \times 10^7$

$F = 0.52$

Energy balances on the gas side were obtained from heat capacity equations for constituent in the combustion product. These equations were summed according to the total moles of each constituent and evaluated as a single equation. These equations are presented in the appendix.

Continuous Equilibrium Vaporization - Per cent Vaporized

In most cases of distillation of such complex mixtures as crude oil, continuous equilibrium vaporization is used. It is then necessary to know the relationship between equilibrium vaporization temperature and per cent vaporized for any given pressure if intelligent design calculations are to be made.

Piromoor and Beiswenger (15) have established a widely used correlation which enables the flash curve of the crude (flash zone temperature versus per cent oil vaporized) to be estimated from the true boiling point curve of the crude. These correlations were later modified by Maxwell (16) and are based upon the empirical facts that:

1. The True Boiling Point curves (TBP) of many commonly encountered crudes and fractions are nearly straight lines between their 70% and 10% vaporized points.

2. There exists a fairly close relationship between the slope of the TBP curve (straight position) and the flash vaporization curve.
3. There exists a relationship between the 50% distillation, TBP temperature and the 50% distillation point of the flash curve.

An example of the use of these relationships is outlined in reference 17. To correct flash curves to other pressure, the flash curve is displaced parallel to itself at a higher or lower temperature (depending on whether the pressure is higher or lower than atmospheric) as determined by the vapor pressure versus temperature chart (Cox Chart). The vapor pressure versus temperature relationship with an atmospheric temperature corresponding to that at the 50% distilled point on the flash curve is chosen.

Physical Properties of the Crude

The physical properties of an oil are found to vary gradually throughout the range of compounds that constitute the oil. The properties such as specific gravity and viscosity are found to be different for each drop or fraction of the material distilled. The rate at which these properties change may be plotted as mid per cent curves; i.e., a plot of the desirable property versus percentage distilled. A mid per cent yeild curve was used to determine the specific gravity at 60°F of the crude at its different stages of vaporization. The viscosity was obtained using the relationship obtained by Nelson (18) for the high temperature viscosities of hydrocarbons.

Calculation of the Density of the Crude at any Temperature (19)

It is assumed that the thermal expansion of any sample may be represented by an equation of the form:

$$D_t = D_T + A(t-T) + B(t-T)^2$$

D_t = density at any temperature t.

D_T = density at a standard temperature

and A and B are based on the change at 25°C.

Since the specific gravities of most materials are given at 15.56°C (60°F), this temperature is chosen as standard. Then,

$$SG_{60} = SG + [\alpha_T + 2\beta(t-25)][t-15.56] + \beta[t-15.56]^2 \qquad (19)$$

converted to degrees Fahrenheit

$$SG_{60} = SG + [\frac{t-60}{1.8}] \quad \alpha_T + \frac{\beta_T}{1.8} \; [3t-217] \qquad (19a)$$

Values of α and β were obtained from charts as functions of the specific gravity of the material.

Thermal Conductivity of Petroleum Liquids (20)

The thermal conductivity of the crude is given by the following equation:

$$K = \frac{0.813}{12(S.G)} [1 - 0.0003 \; (t-32)] \qquad (20)$$

Pressure Drop Calculations

The problem of calculating pressure drop in tube still heaters cannot be solved by the conventional Fanning equation; since vaporization of the crude with increasing temperatures, results in the presence of two phases, making the equation inapplicable.

Pressure drops encountered in two-phase systems are higher than those resulting from single phase flow for a number of reasons. The energy change of the phase transition, the frictional energy remaining in the system due to the internal shear at the boundaries of each phase, and the reduced cross-sectional area of flow for one fluid produced by the presence of a second fluid all affect the pressure drop. Consideration must also be given to the Hydraulic Energy (PV) of the fluid mass which not only changes with temperature and pressure, but also with the composition of the liquid and vapor phases.

The complex conditions of multiphase flow and the number of variables involved has been the subject to intensive survey, and relationships (21, 22, 23) have been proposed which correlate two-phase flow data. These investigations have been restricted to isothermal conditions and no attempts have been made to propose a correlation for non-isothermal systems in which there is a continuous change of phase. Therefore it was assumed that each section of pipe would behave isothermally at the average volume fractions and physical properties of the crude. The data of Reid, Reynolds, et al (22) was chosen because their investigations were conducted on pipes of similar diameter to those encountered in tube stills.

By a proper definition of terms, pressure drop calculations for two phase flow could be predicted by a Fanning type equation:.

$$\Delta P_{TP} = \frac{2\ \phi L G^2}{\lambda\ g_c D_i} \tag{21}$$

λ = pseudo density of the two phase mixture.

ϕ = pressure coefficient analogous to the friction factor of the Fanning equation.

In order to calculate λ, the following assumptions were made:

1. The volume occupied by the liquid plus the volume occupied by the vapor, at any instant, must equal the total volume of the pipe.
2. The linear velocities of vapor and liquid phases are equal.
3. Flow is sufficiently turbulent to cause complete mixing of both phases.

From (2) $\dfrac{W}{\lambda D_i^2} = \dfrac{W_L}{\rho_L D_L^2} = \dfrac{W_v}{\rho_v D_v^2}$

From (1) $D_L^2 + D_v^2 = D_i^2$

$$\frac{W_L}{\rho_L D_L^2} = \frac{W_v}{\rho_v D_v^2}$$ Substituting for D_v

$$\frac{W_L \rho_v D_i^2}{\rho_L W_v + \rho_v W_L} = D_L^2$$

$$\frac{W}{\lambda D_i^2} = \frac{W_L}{\rho_L\left[\frac{W_L \rho_v D_i^2}{\rho_v W_L + \rho_L W_v}\right]}$$

$$\lambda = \frac{W_L \rho_L \rho_v}{\rho_L W_v + \rho_v W_L} + \frac{W_v \rho_L \rho_v}{\rho_L W_v + \rho_v W_L}$$

The liquid volume fraction (LVF) = $\dfrac{W_L \rho_v}{W_L \rho_v + W_v \rho_L}$

The vapor volume fraction (VVF) $= \frac{W_V \rho_L}{W_L \rho_V + W_V \rho_L}$

$$\lambda = \rho_L \text{ (LVF)} + \rho_V \text{ (VVF)} = \text{LVF } (\rho_L - \rho_V) + \rho_V \qquad (22)$$

Since ϕ is an empirical constant it must be obtained from a correlation of two phase data.

Reid, Reynolds, et. al. (23) have shown that for liquid volume fractions above 10 per cent, one follows the relationship:

$$\Delta P_{TP} = \Delta P_L \text{ (LVF)}^{-1} \qquad (23)$$

In this equation the following assumptions are made: (1) single phase friction factor correlating charts are applicable to two phase flow problems; (2) the superficial average densities and velocities can be employed.

ΔP_L is the liquid phase pressure drop as calculated from single phase correlations if the liquid were flowing alone in the pipe at the same rate as the two phase flow.

If the assumption is made that $\lambda = \text{LVF } (\rho_L)$, a comparison of the two Equations (21 and 23) indicates that ϕ may be approximated by the friction factor of the liquid phase. For Reynolds numbers, Re>2.5 x 10^6 Perry (24) recommends the following equation for friction factors:

$$f = 0.0014 + 0.09 \left(\frac{\mu}{DG}\right)^{0.27}$$

Hence

$$\phi = 0.0014 + 0.09 \left(\frac{\mu}{DG}\right)^{0.27} \qquad \text{Where } 0.1 \leq \text{LVF} \leq 1.0 \qquad (24a)$$

At liquid volume fractions below 0.1 the vapor can be assumed to behave independently of the liquid. Therefore values of ϕ were approximated from the single phase friction factor correlating charts at vapor Reynolds numbers greater than 2 x 10^5. These values varied from 0.0014 to 0.005. It was also observed from two phase data that ϕ decreases as the LVF decreases, and increases as the Reynolds' number of the liquid phase decreases.

In view of this, it was concluded that the following equation could predict sufficiently accurate values of ϕ at low liquid volume fractions

$$\phi = 0.0014 + \text{LVF} \left(\frac{\mu}{DG}\right)^{0.27} \qquad \text{Where LVF} < 0.1 \qquad (24b)$$

STATEMENT OF THE PROBLEM

Restrictions and Limitations

This investigation was initiated to study the effect of the variables of per cent excess air, center to center spacing, bridgewall and exit stack temperatures on the design of Petroleum Furnaces.

Based on a semi-theoretical equation proposed by Lobo and Evans for the calculation of heat transfer rates in the radiant section of tube still furnaces, and on an empirical equation developed by Monrad, a series of design equations were written. These equations were solved for different values of the variables and their effect on heat transfer rates and furnace dimensions studied.

The Equation of Lobo and Evans was developed for applications only to already designed or completed furnaces. However, as it was used for the actual design in the investigation, certain assumptions were necessary. These are:

1. View factors to individual tubes can be calculated using the empirical relationship of Lobo and Evans.

2. The mean beam path to the individual tubes can be approximated by the average beam length of the furnace.

The furnaces considered in this investigation were limited to the conventional box-type with the width equal to the height, single rows of tubes in the radiant section and with equal center to center spacing in the convection and radiant sections. Four tubes per row were placed in the convection bank and the length of the furnace arbitrarily fixed and equal to the length of tubes.

Calculations for the inside film coefficient of the tubes were based on the liquid phase only. The errors caused by this approximation are very slight as in most cases the outside film coefficient controls. Errors in the tube temperature calculated from this coefficient do not appreciably affect the heat transfer rates as the difference between the fourth power of the gas and the surface temperatures are extremely large.

The reliability of the final design is limited by the accuracy of the physical properties data, and the validity of the assumptions used in obtaining the design equations. Perhaps the most severe of these restrictions is that encountered in the approximation of friction factors used in pressure drop calculations.

RESULTS OF CALCULATIONS

The results of the calculations are summarized in Tables III and IV. Table II gives the service requirements used in the investigation, as well as the furnace characteristics. The results presented in Table III were obtained without considering pressure drop requirements. Table IV shows the results obtained when the diameters are calculated to satisfy the requirements.

TABLE II

Service Requirements of Furnaces

Furnace Group Symbol Unit	Flow Rate lb/hr	Flow Rate Bbls/hr	Inlet Temp. °F	Exit Temp. °F	Inlet Press. Psia	Exit Press. Psia	Tube Spacing Tube Diam.	Tube Length Ft.	Tube Diam. Ft.	Per Cent Excess Air
1	39,262.4	133	410	670	117.7	-	1.5	20	0.41	50
2	39,262.4	133	410	670	117.7	-	2.0	20	0.41	50
3	80,000.0	-	400	750	240.0	-	1.5	25	0.45	50
4	80,000.0	-	400	750	240.0	-	2.0	25	0.45	50
5	80,000.0	-	400	800	240.0	-	1.5	25	0.45	50
6	80,000.0	-	400	800	240.0	50	1.5	25	0.45	25
7	80,000.0	-	400	800	240.0	50	1.5	-	-	-

TABLE III

Stack Temp.	Bridge Wall Temp.	Lb. Flue Gas Per Hour	Q_R x10^{-6}	Q_C x10^{-6}	Q_L x10^{-6}	Exit Conditions Temp.	Exit Conditions Press.	Exit Conditions Size	Number of Tubes N_R	Number of Tubes N_C	R_A	$\frac{Q_R}{Q_C}$	$\frac{Q}{\alpha A_{cp}}$ x10^{-4}
Furnace 1													
830	1416.04	13870.7	4.6606	2.4702	4.7980	671.00	63.59	8.123	26	44	0.3907	1.887	3.85
830	1494.67	13870.7	4.4495	2.8138	4.7980	675.81	64.29	7.446	22	48	0.3689	1.581	4.6
830	1670.77	13870.7	3.4409	3.5929	4.7980	667.82	66.78	5.416	14	56	0.2908	0.958	7.09
930	1395.58	14914.5	5.0685	2.1199	5.5997	671.89	72.85	9.477	28	32	0.3963	2.391	4.08
930	1554.33	14859.7	4.5095	2.8566	5.5791	678.34	74.43	7.466	20	40	0.3484	1.579	5.43
930	1646.57	14717.3	3.7871	3.2630	5.5256	667.22	75.83	5.416	16	44	0.3011	1.161	6.6
1030	1417.78	15861.6	5.3092	1.8905	6.4296	671.53	79.88	9.477	28	24	0.3895	2.808	4.08
1030	1495.73	15789.5	4.8882	2.2697	6.4003	670.12	80.78	7.446	24	28	0.3605	2.154	4.74
1030	1671.58	15596.7	3.9474	3.1170	6.3222	666.95	82.49	5.416	16	36	0.2949	1.266	7.01
Furnace 2													
830	1463.61	13870.7	4.4744	2.6777	4.7980	674.52	49.68	8.123	18	68	0.3744	1.671	4.6
830	1575.14	13370.7	3.9513	3.1682	4.7980	674.26	44.59	6.318	14	76	0.3316	1.247	5.9
930	1426.84	14781.4	4.8026	2.2458	5.5497	668.14	70.19	9.026	20	68	0.3812	2.139	4.37
930	1586.82	14781.4	4.0437	3.0129	5.5837	669.32	65.52	6.318	14	74	0.3199	1.342	6.27
1030	1454.11	15780.3	5.0714	2.0611	6.3966	669.84	80.17	9.026	20	36	0.3749	2.461	4.7
1030	1558.48	15906.5	4.5847	2.6032	6.4478	672.28	77.74	7.221	16	44	0.3362	1.761	5.93

TABLE III (continued)

Stack Temp.	Bridge Wall Temp.	Lb. Flue Gas Per Hour	Q_R x 10^{-6}	Q_C x10^{-6}	Q_L x10^{-6}	Exit Conditions Temp	Press.	Size	Number of Tubes N_R	N_C	R_A	$\frac{Q_R}{Q_C}$	$\frac{Q}{\alpha A_{cp}}$ x10^{-4}
Furnace 3													
850	1496.59	42648.8	1.2135	8.4268	1.5003	753.20	128.56	14.038	36	64	0.3412	1.44	6.02
950	1494.34	42648.8	1.3495	7.1333	1.6266	753.26	143.96	14.038	40	48	0.3662	1.892	5.63
1050	1506.17	42648.8	1.4798	6.0147	1.7545	755.72	155.85	15.516	42	36	0.3858	2.460	5.56
Furnace 4													
850	1375.82	42648.8	1.3765	6.8074	1.5003	755.93	90.43	19.703	40	84	0.3369	2.022	4.49
850	1481.73	42648.8	1.2404	8.2264	1.5003	757.31	92.77	14.77	30	96	0.3481	1.508	6.0
950	1381.23	42648.8	1.4806	5.6165	1.6266	751.21	128.62	20.668	42	60	0.4036	2.636	4.4
950	1490.65	42648.8	1.3640	7.0836	1.6266	756.43	129.52	15.76	32	72	0.3688	1.926	5.83
1050	1379.5	42648.8	1.6409	4.3152	1.7545	754.88	147.57	22.66	46	40	0.4288	3.803	4.2
1050	1491.65	42648.8	1.4718	5.8139	1.7545	752.0	152.47	16.75	34	52	0.3865	2.529	5.66
1150	1480.67	46931.4	1.6329	4.8174	2.0729	761.05	163.35	18.72	38	36	0.3899	3.39	5.56

TABLE III (continued)

Stack Temp.	Bridge Wall Temp.	Lb. Flue Gas Per Hour	Q_R x10^{-6}	Q_C x10^{-6}	Q_L x10^{-6}	Exit Conditions Temp.	Press.	Size	Number of Tubes N_R	N_C	R_A	$\frac{Q_R}{Q_C}$	$\frac{Q}{\alpha A_{cp}}$ x10^{-4}
Furnace 5													
850	1386.76	42648.8	1.6868	6.9533	1.5003	808.07	55.81	22.166	60	56	0.4345	2.43	3.94
850	1618.03	42648.8	1.3737	1.0075	1.5003	806.64	108.68	13.3	34	72	0.3539	1.36	6.98
950	1357.10	47655.9	1.7662	5.9169	1.8176	802.60	83.25	23.643	66	40	0.4230	2.99	3.86
950	1570.55	48196.6	1.4614	9.2273	1.8382	805.80	125.59	14.78	38	56	0.3461	1.58	6.8
1050	1372.12	47655.9	1.9138	4.7119	1.9605	805.89	104.89	23.643	68	28	0.4404	4.06	3.9
1050	1595.98	48196.6	1.5399	8.1738	1.9827	800.50	144.71	14.78	33	44	0.3548	1.88	6.96
Furnace 6													
850	1404.46	36123.8	1.7477	5.9672	1.2397	800.99	86.82	22.166	58	52	0.488	2.93	3.68
850	1465.63	36123.8	1.7233	6.6423	1.2397	808.61	94.34	19.21	52	56	0.475	2.6	4.15
950	1378.28	36123.8	1.9168	4.6252	1.3443	805.65	96.63	24.38	66	36	0.5148	4.14	3.36
950	1502.57	36123.8	1.7981	6.0090	1.3443	803.16	122.32	19.21	50	44	0.48	2.99	4.45
1050	1355.50	39731.8	2.0178	3.645	1.5951	805.45	104.30	25.86	72	24	0.5073	--	3.28
1050	1472.61	40069.2	1.3609	5.1182	1.6087	803.17	132.95	20.69	54	32	0.4674	--	4.39

TABLE IV

Per Cent Excess Air	Exit Stack Temperature = 850°F: Bridge Wall Temp.	Lb. Flue Gas Per Hour	Q_R x10^{-7}	Q_C x10^{-7}	Q_L x10^{-7}	Exit Conditions: Temp.	Press.	Size	Number of Tubes: N_R	N_C	Tube Dia. Ft.	R_A
Furnace 7												
25	1392.7	36123.8	1.7304	5.8314	1.2397	801.71	51.61	22.166	64	52	0.4128	0.487
25	1462.66	36123.8	1.68906	6.60916	1.2397	805.93	60.86	18.403	54	56	0.4115	0.47
25	1593.9	36123.8	1.56244	8.0838	1.2397	809.07	74.9	14.22	42	64	0.4100	0.433
50	1376.4	42648.7	1.6351	6.8156	1.500	802.38	48.32	22.166	64	56	0.4157	0.428
50	1614.8	42648.7	1.31014	1.00313	1.500	800.15	69.75	12.864	36	72	0.4100	0.344
50	1286.5	42648.7	1.78375	5.6239	1.500	804.35	55.0	22.166	84	48	0.4281	0.464
100	1386.4	64669.5	1.28784	9.87721	2.1508	800.23	61.98	18.003	54	64	0.4184	0.291
100	1266.7	64635.6	1.5427	7.6223	2.1497	798.14	40.0	27.794	76	52	0.4219	0.345
100	1593.4	64635.6	9.3180	1.38224	2.1497	801.44	37.13	10.35	32	80	0.4071	0.209
125	1391.8	83354.9	1.06017	1.2835	2.75988	803.8	54.0	15.89	48	72	0.4191	0.208

DISCUSSION OF RESULTS

The design of tube still heaters should be made by considering distributions of the heat load between both sections of the furnace. When a distribution ratio has been chosen, the design of a petroleum furnace can then be established on a basis of the permissible average radiant heat transfer rate.

The results illustrated in Figures IV and V show the heat distribution ratio to be a function of bridgewall temperature; exit stack temperature; and furnace capacity. These curves include the results obtained using two tube spacings, indicating that the heat distribution ratio is independent of this variable.

Figure VI is presented in order to permit visualizing the effect of excess air on the distribution ratio. At a specific bridgewall temperature a greater percentage of heat can be distributed to the radiant section by decreasing the percentage excess air. Using the bridgewall temperature as a parameter, this effect was correlated in terms of the fraction of the total heat input absorbed in the radiant section, and plotted versus the per cent excess air. These results are shown in Figure VII.

In the preceeding discussion, emphasis is placed upon available distributions of the total heat load between both sections of the furnace without considering the transfer rates per unit area of cold surfaces. It is quite difficult to generalize regarding allowable rates of heat transfer as this would naturally depend upon the rate at which the oil removes heat from the tubes and the maximum temperature to which the tube may be heated without causing corrosion, distortion of the tube, thermal cracking of the crude and coke deposition inside the tubes. It should not be concluded however, that a choice of the maximum allowable rate results in the best furnace design, as this maximum may only be attainable in one section of the furnace. The choice should be based upon rates in both sections which will yield the lowest total tube surface area.

The results of these calculations indicate that this situation is accompanied by high bridgewall temperatures and small quantities of excess air.

In the interests of economy, it is the current trend to fill the radiant section with cold tube surfaces. Figure VIII gives an indication of the relationship between the cold plane area in the radiant section and the fraction of the total heat input absorbed in this section. These curves show that this fraction, although independent of the tube spacings used, varies with excess air and furnace capacity. These curves are also significant in that they give an indication of the optimum size of the radiant section. For example, if the maximum allowable rate of heat transfer was found to be 20,000 Btu/ft^2 for Furnace 1 (illustrated by curve 1 in Figure VIII), then a design could not be made such that the fraction of the total heat absorbed in the radiant section is less than 0.32. Also, if the permissible rate were close to the maximum, for example 18,000 Btu/hr ft^2 then the resulting small radiant section would be obtained at a loss of economy. This, however, can be avoided by increasing the per cent excess air and thus increasing the fraction absorbed for a specific radiant rate.

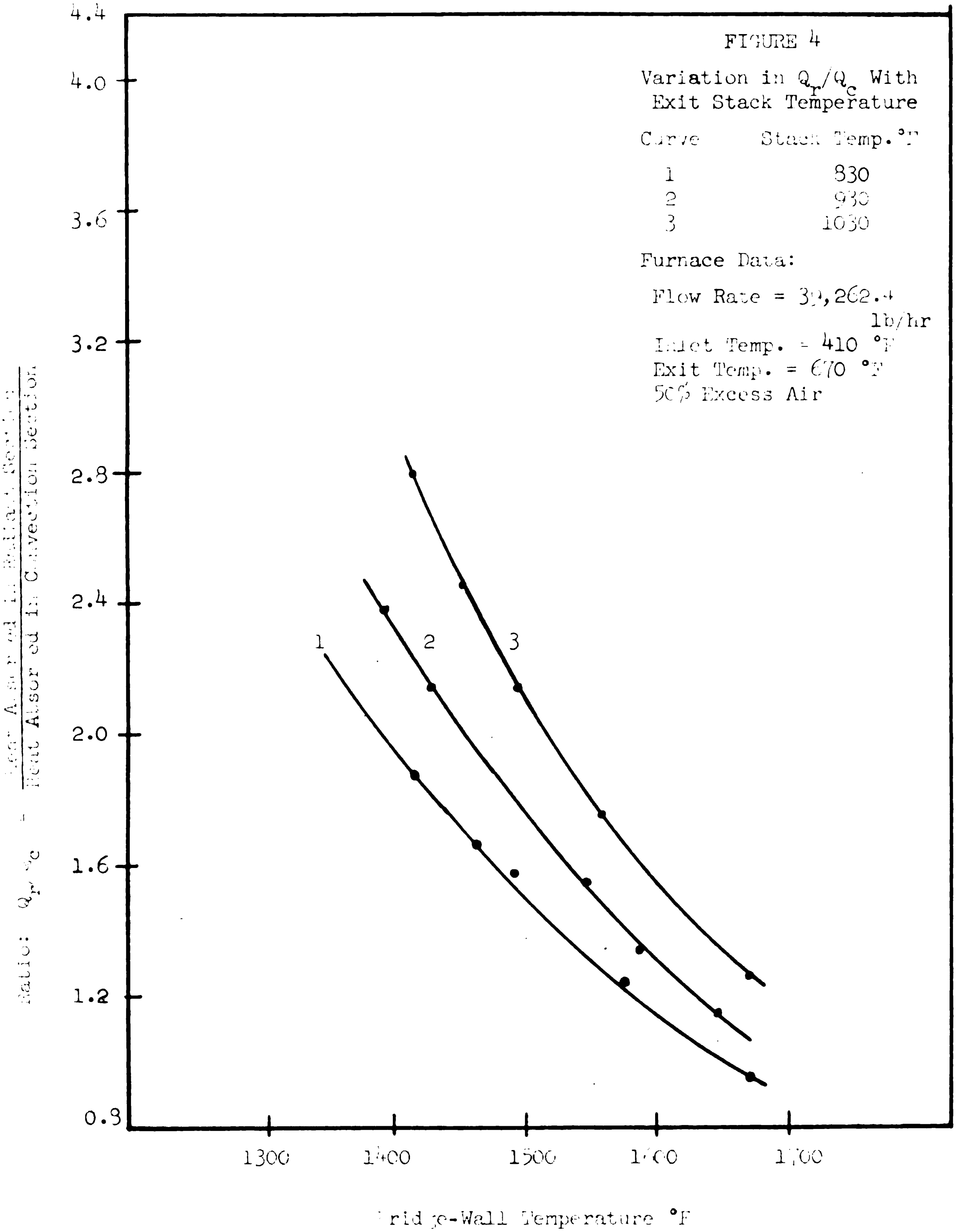

FIGURE 4
Variation in Q_r/Q_c With Exit Stack Temperature
Curve Stack Temp. °F
1 830
2 930
3 1030
Furnace Data:
Flow Rate = 39,262.4 lb/hr
Inlet Temp. = 410 °F
Exit Temp. = 670 °F
50% Excess Air
1
2
3
4.4
4.0
3.6
3.2
2.8
2.4
2.0
1.6
1.2
0.8
1300
1400
1500
1600
1700
Ratio: Q_r/Q_c = Heat Absorbed in Radiant Section / Heat Absorbed in Convection Section
Bridge-Wall Temperature °F

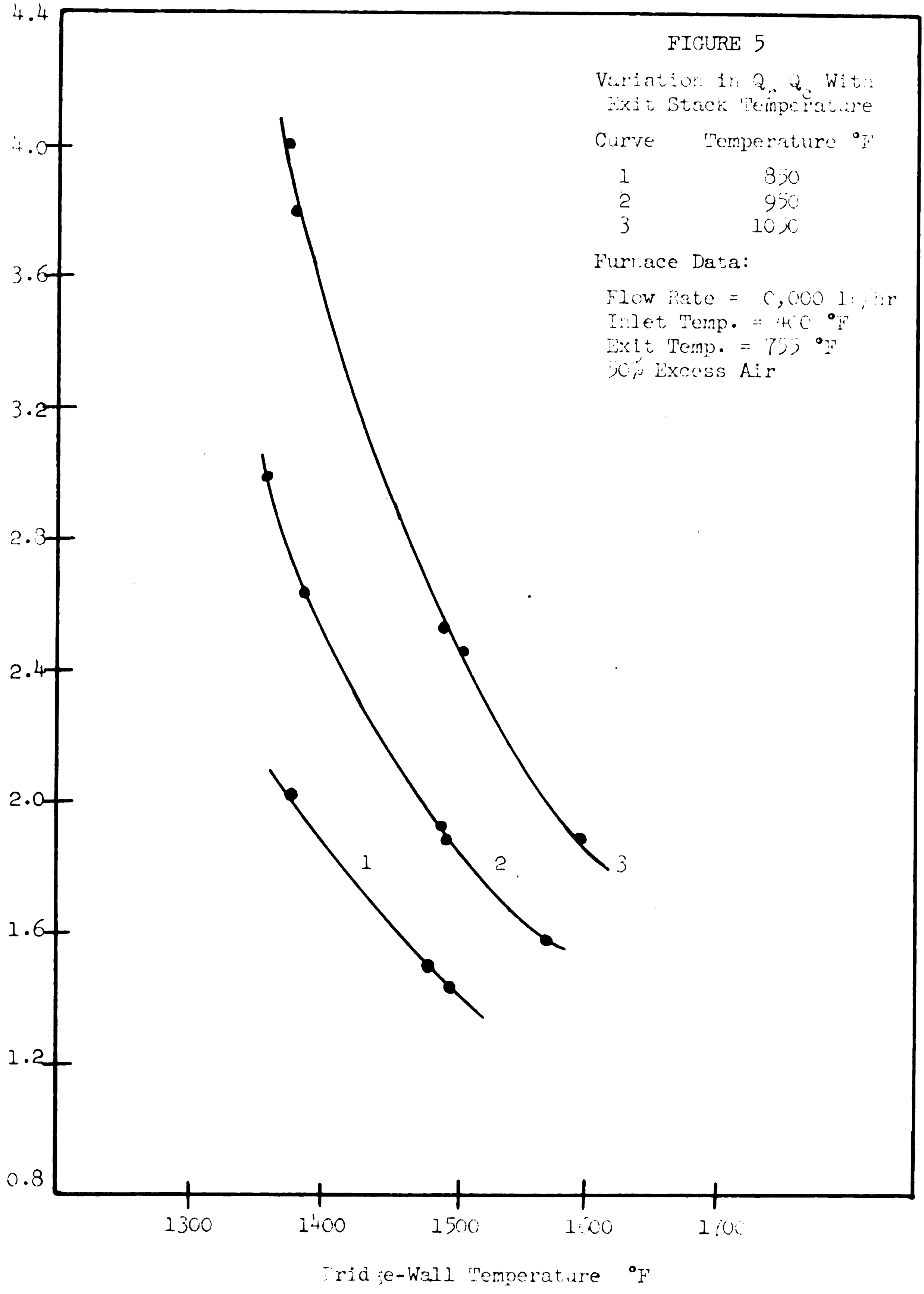

FIGURE 5
Variation in Q_r/Q_c With Exit Stack Temperature
Curve Temperature °F
1 850
2 950
3 1050
Furnace Data:
Flow Rate = ,000 lb/hr
Inlet Temp. = °F
Exit Temp. = 755 °F
Excess Air
1
2
3
4.4
4.0
3.6
3.2
2.8
2.4
2.0
1.6
1.2
0.8
1300
1400
1500
1600
1700
Ratio: Q_r/Q_c = Heat Absorbed in Radiant Section / Heat Absorbed in Convection Section
Bridge-Wall Temperature °F

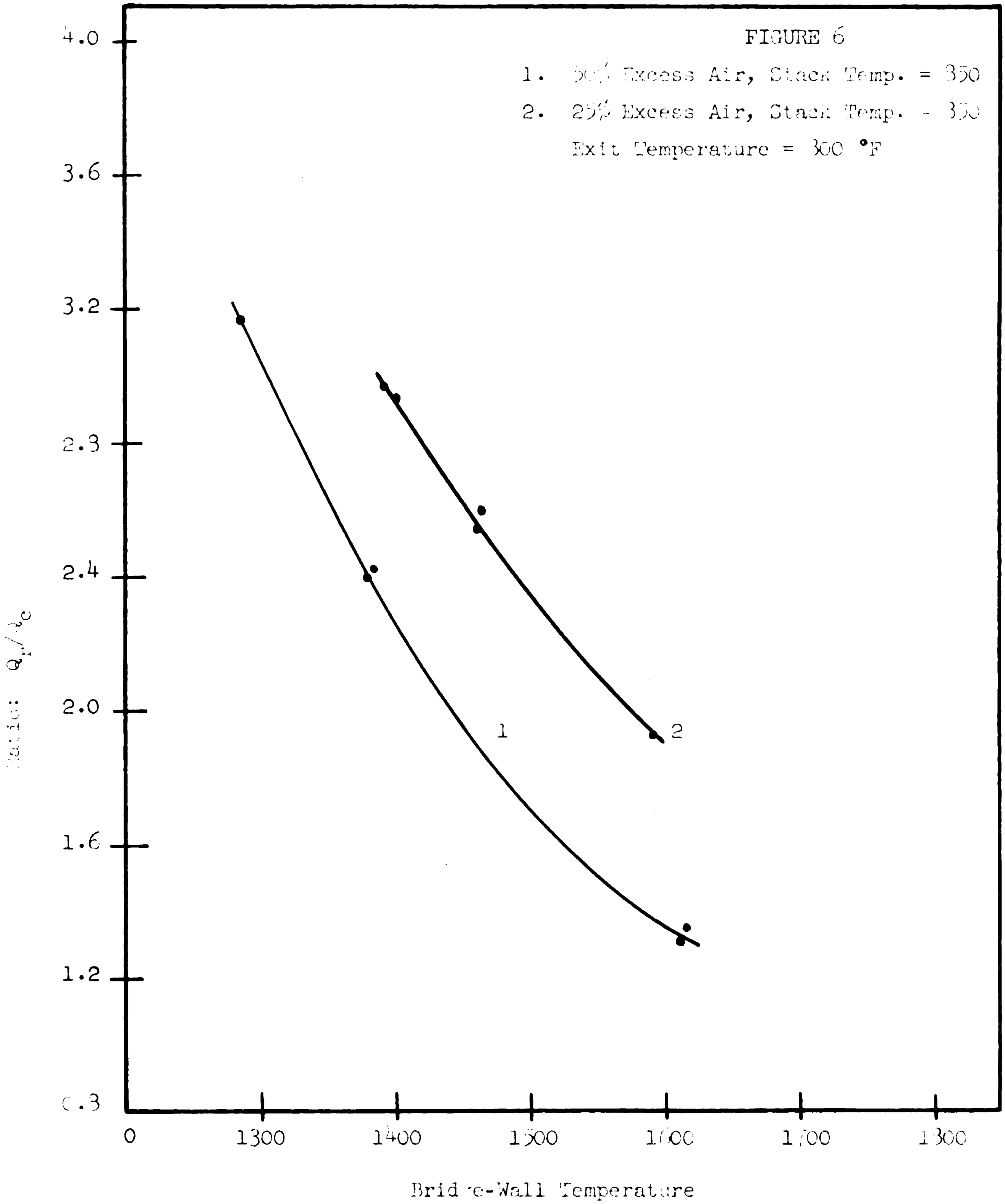

FIGURE 6
1. 50% Excess Air, Stack Temp. = 350
2. 25% Excess Air, Stack Temp. = 350
Exit Temperature = 300 °F
4.0
3.6
3.2
2.8
2.4
2.0
1.6
1.2
0.8
0
1300
1400
1500
1600
1700
1800
1
2
Ratio: Q_r/Q_c
Bridge-Wall Temperature

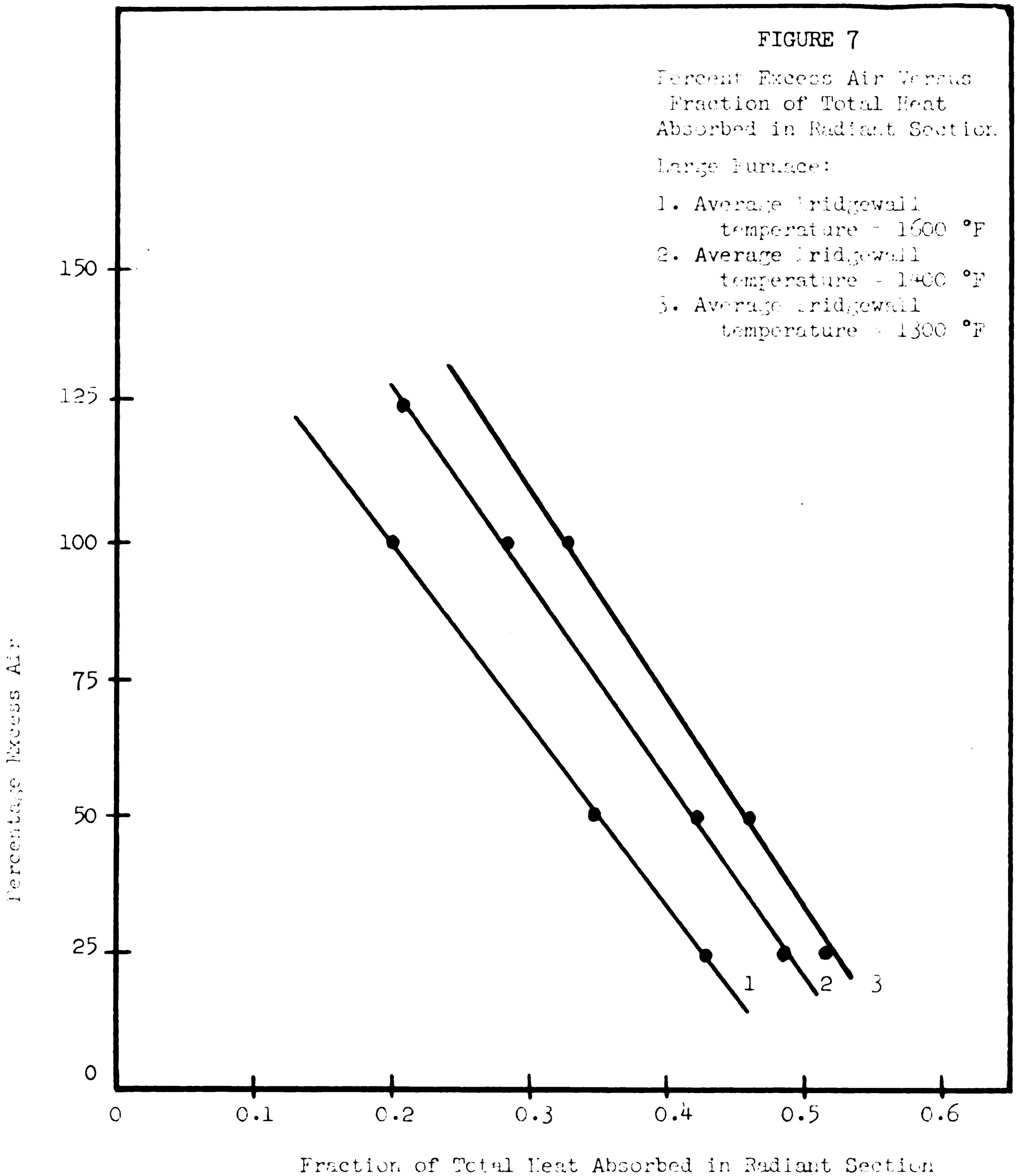
FIGURE 7
Percent Excess Air Versus Fraction of Total Heat Absorbed in Radiant Section
Large Furnace:
1. Average bridgewall temperature - 1600 °F
2. Average bridgewall temperature - 1400 °F
3. Average bridgewall temperature - 1300 °F
150
125
100
75
50
25
0
0.1
0.2
0.3
0.4
0.5
0.6
1
2
3
Percentage Excess Air
Fraction of Total Heat Absorbed in Radiant Section

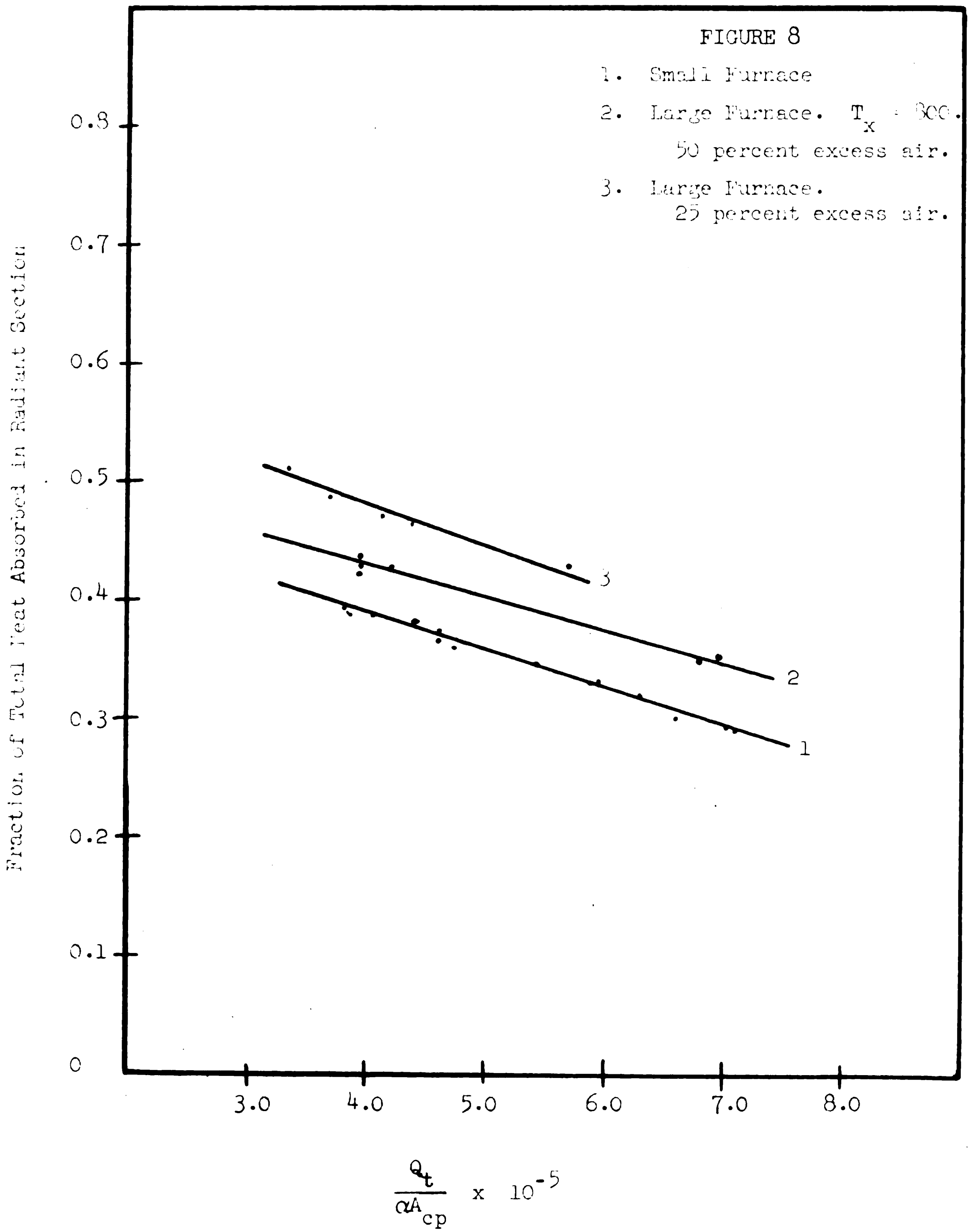
1. Small Furnace
2. Large Furnace. Tx = 800.
50 percent excess air.
3. Large Furnace.
25 percent excess air.
Fraction of Total Heat Absorbed in Radiant Section
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
3.0
4.0
5.0
6.0
7.0
8.0
Qt / αAcp x 10-5
1
2
3

FIGURE 8

CONCLUSIONS AND RECOMMENDATIONS

Accurate furnace designs can be obtained using mathematical models similar to that employed in these calculations. Although these models are useful in determining the effect of various phenomenon in furnace characteristics, they cannot be used to directly determine optimum designs. In order to obtain the best design for a furnace, models must include economic considerations as well as those factors included in this model.

The factors most significant in determining the ultimate design of furnaces are the distribution of total heat load to the radiant and convection section, and the average heat transfer rates.

The heat distribution ratio of a furnace is dependent upon the per cent excess air and bridgewall and exit stack temperatures. Increases in any of these characteristics will result in a larger distribution ratio.

Average heat transfer rates in the radiant section are almost entirely dependent upon the fraction of the total heat input absorbed in this section and increases as the fraction decreases. This fraction is dependent on the per cent excess air and bridgewall temperatures and will increase as either variable decreases.

The total tube area required for a furnace with a specific duty is usually lowest when the bridgewall temperature is high and the per cent excess air is low.

The number of calculations made throughout this investigation was severely restricted by the size of the computer available and the type of programming employed. Should a similar investigation be attempted, it should be conducted on a larger computer using a faster method of interpretive programming. It is also recommended that a more accurate method of predicting pressure drop in the presence of two phase flow be obtained.

APPENDIXES

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NOMENCLATURE

A_c = total outside tube area, ft^2.
A_R = effective refractory area, ft^2.
A'_R = actual refractory area, ft^2.
A_T = total wall area, ft^2.
A_{cp} = area of plane replacing tubes, ft^2.
C_p = heat capacity, Btu/lb °F.
D = pipe diameter, ft.
D' = pipe diameter, ins.
D_L = equivalent diameter of liquid phase.
D_V = equivalent diameter of vapor phase.
d = fraction distilled.
E_b = black body emissive power, Btu/hr ft^2.
F = geometric factor.
ψ = overall exchange factor.
f = Fanning friction factor.
G = superficial mass velocity, lb/hr ft^2.
H = enthalpy, Btu/lb.
ΔH = heat absorbed by crude per node, Btu/lb.
h_c = pure convection coefficient, Btu/hr ft^2 °F.
h_r = radiant coefficient, Btu/hr ft^2 °F.
K = thermal conductivity of crude, Btu/hr ft^2 °F/ft.
K_m = thermal conductivity of tube, Btu/hr ft^2 °F/ft.
L_B = mean length of radiant beam, ft.
L = length of radiant section, ft.
ΔL = length of tube node, ft.
P = pressure, lb/in^2
P_{CO_2} = partial pressure of CO_2, atms.
P_{H_2O} = partial pressure of H_2O, atms.
q = heat transferred to the oil, Btu/hr.
S.G = specific gravity.
T = temperature, °R.
t = temperature, °F.

u = overall heat transfer coefficient, Btu/hr ft^2 °F.

W = mass flow rate, lb/hr.

Z = height of furnace, ft.

α = factor by which A_{cp} must be reduced to obtain effective cold surface, αA_{cp} (effective tube area).

ϵ_G = gas emissivity.

ϕ = pressure drop coefficient.

μ = viscosity, lb/ft hr.

λ = pseudo density of two phase mixture.

¢-¢ = center to center spacing of the tubes.

δ = Stefan-Boetzmann constant, 0.173×10^{-8} Btu/hr ft^2°R^4

ρ = density, lb/ft^3.

Subscripts

b = black body.

c = convection.

g = gas.

i = inside.

m = arithmetic average.

o = outside.

r = radiation.

s = surface.

w = wall.

1 = inlet.

2 = exit.

PROGRAM ABSTRACT

TITLE: Pipe Still Heater Design.

AUTHOR: William V. Saunders.

DESCRIPTION

The program calculates the dimensions, number of tubes and their diameter in the radiant and convection sections of the furnace. The method used includes the correlations of Lobo and Evans for the evaluation of radiant heat transfer rates and those of Monrad for calculating convection coefficients.

COMPUTER

MISTIC, 1024 cathode-ray tube memory locations, perforated tape input and output.

PROGRAM LANGUAGE

Fixed point and floating point coding.

RUNNING TIME

Six to ten hours depending on the accuracy of initial estimates of the guessed quantities.

COMMENTS

The engineer can have the calculation stop after any of the two sections: radiant section, convection section. Program has been successfuly used over forty times in designing furnaces.

AVAILABILITY

A manual for the description of the codes used in this program is available in the Computer Laboratory library at Michigan State University. This program is available from the Computer Library in the Chemical Engineering Department.

DESCRIPTION OF PROGRAM

Description

To handle the lengthy calculations involved in designing the tube-still furnace, a procedure was developed for use with a small sized digital computer. The machine routine is such that a choice of tube length, flow rate of crude stock, center-to-center spacing of the tubes, percentage excess air, and bridgewall and exit stack temperatures may be varied in considering the different designs.

The furnace may be calculated in any increment of tube length desired; however, as the computer routine used was an extremely slow one, it was necessary to shorten the calculations as much as possible in order to have expediency of calculation time. Such being the case, an increment of four tube lengths was chosen in the convection section and two tube lengths in the radiant section.

The flow of calculations around a tube increment is shown in Figure 9. At the inlet of the tube, the temperature, pressure and liquid volume fraction, (t, T, P, LVF) is known from the previous tube, or if the first calculation from the inlet conditions to the furnace. The inlet conditions (t_2 and P_2) are guessed and their arithmetic averages computed. Based on these average values, new values of t_2 and P_2 are calculated.

The calculated values of the outlet temperature and pressure are compared with the assumed values. If the difference between calculated and assumed values are not within tolerance, new values for the exit conditions are chosen and the procedure repeated until the outlet values are within tolerance. These values are then used for the next tube.

When the gas temperature above the tube reaches a certain maximum, which is set as the highest allowable bridgewall temperature, calculations for the convection section are stopped and calculations for the radiant section commences. When the exit temperature and pressure from a tube increment compares

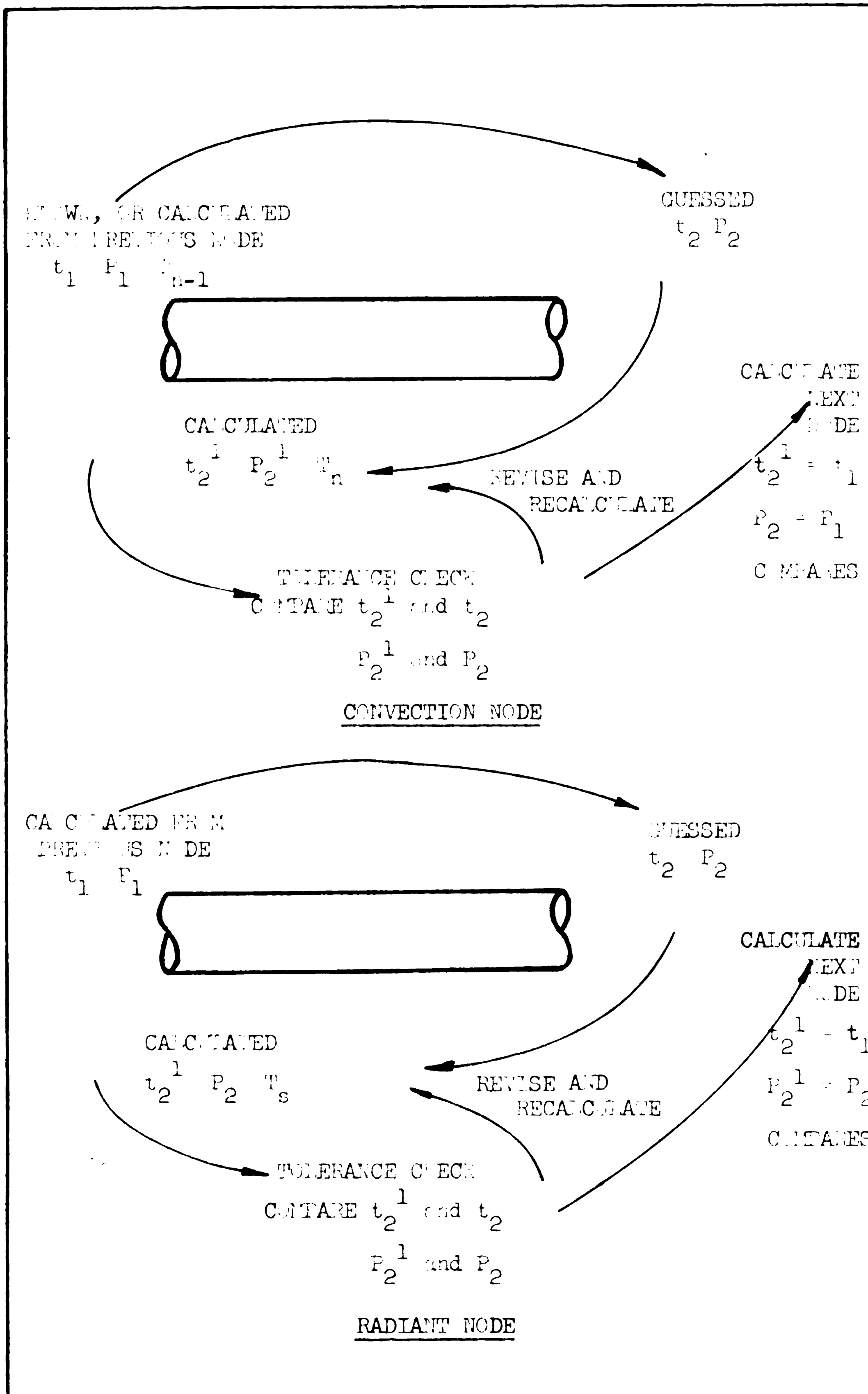

FIGURE 9. Calculations Flow Around One Tube Increment

with the desired discharge conditions, the dimensions of the furnace are calculated from the number of tubes in this section and compared with the assumed dimensions. If the discharge conditions do not check within tolerance a new diameter is assumed and all calculations repeated. Calculations within the radiant section are repeated when the assumed dimensions do not compare with those calculated.

Finally a heat balance is made about the furnace and the fuel rate modified until the heat liberated equals the heat absorbed plus the heat lost from the furnace.

Method of Fitting Equations to Data

The numerous charts necessary to the solution of this problem had to be programmed so that they could be interpreted by the computer. The most convenient and accurate method of doing this is by fitting the curve to polynomials.

Polynomials were obtained by reading a set of points from each curve and by fitting these points to a polynomial of the form

$$F_{(X)} = 1/2 \sum_{s=o}^{n-1} A_s X^s$$

The criterion of excellence for each polynomial was that the sum of the squares of the deviations

$$M = \sum_{i=o}^{n-1} [F_{(X_i)} \div f_{(X_i)}]^2 A_w(X_i)$$

$$f_{(X_i)} = \text{points from the curve}$$

Should be a minimum with respect to arbitrary variations of the coefficients A_s.

Two library routines, K_3 and L7S, were available for the evaluation of these polynomials. The constants, as obtained for the curves used in these calculations are presented in Table 5.

MACHINE REQUIREMENTS

The automatic computer used for the furance calculations was MISTIC, a binary, fractional, single address computer with a word length of 40 bits, a memory of 1024 words, and which puts two instructions in a word.

MISTIC like most digital computers is composed of five units: input, memory or storage, control, arithmetic, and output.

Input

This unit includes as an input medium a 5-level perforated Teletype paper tape by which the problem is communicated to the computer, and as an input device a photo-electric reader which is able to pull the tape past a light. The light shines through the holes activating a photo-sensitive surface which converts the spots to electrical impulses, equivalent to the number represented by the coded character on tape. These impulses are sent directly to an assembly register.

Memory

The numbers received by the assembly register are sent under the control of the computer to memory. The memory is an electronic device composed of 40 vacuum tubes. On the face, or gird, of each tube 1024 spots can be stored. Each spot is assigned a certain address or location, and corresponds to a binary digit from every word. Each tube represents a different binary digit, that is, a power of two (referred to as a bit) to comprise a total of 40 bits, or one word.

Information stored in memory usually consists of a series of instructions (a program), directing the machine to execute certain operations and a set of data on which these operations are performed.

Arithmetic

The instructions and data held by memory are used both by the arithmetic and control sections. The arithmetic section is that section of the computer in which mathematical and logical operations are performed. These operations include: addition, subtraction, multiplication, division, and the comparison of two quantities and are associated with three 40 bit registers and an adder. These are the A register (accumulator), the Q register (quotient), the R register and the adder. In an addition, A holds one of the terms and the result, Q holds the quotient of a division and has no additive properties. The adder is a register on which the number in A is added to the number in R.

Control

The control section of the computer directs every operation executed by the computer. MISTIC stores the program provided by the operator in memory, and before any operation can be executed it must be summoned from memory by the control section. This section is comprised of an instruction register (IR) which holds the instruction currently being executed by control and a control counter which holds the address of the next instruction pair to be sent to IR.

Output

When the result of a calculation has been obtained which must be transferred to the operator, it is sent to the output section. The output section is similar to input in that electrical impuses are converted to a suitable code and translated by an electro-magnetic devise and presented to the reader on tape. Teletypewriters are available and are used for printing on paper the symbols represented by the punched tape.

Subroutines

MISTIC operates in binary, and in view of this, special attention must be given to the location of the decimal or binary points of numbers before every arithmetic operation. Its arithmetic unit necessitates the binary point to be fixed so that any number X used in computation must be in the range $-1 \leq X < 1$. It is necessary then, that each number at every stage of a calculation be scaled within the capacity of the Machine. Many problems encountered in engineering calculations involve numbers of various magnitudes making scaling an additional inconvenient complexity. For these calculations floating point routines may be used. These routines represent numbers as $x = a \times 10^b$ and store a and b. Thus they can represent numbers in the range of $10^{-63} \leq x < 10^{63}$. The floating point routine used in the preparation of this program was a standard library routine designated as A1.

Running Time and Accuracy

The running time for a complete computation cannot be predicted exactly as this would depend on the accuracy of the initial estimates of the guessed quantities and the number of increments chosen. The running time necessary for an increment of tube length to converge was approximately three minutes and that for a complete computation averaged to about five hours. A conservative estimate of the time required to perform a complete computation by hand, would be 110 hours. The accuracy for each arithmetic operation is set by that of the Al Routine which provides an answer to at most nine decimal digits. However, the elaborate trial and error computation involved necessitated a choice of limits-of-convergence which resulted in an accuracy within 0.8 per cent of the correct answer for the exit temperature and 1.5 per cent for the exit pressure for each tube increment.

Error Stops

During the necessarily lengthy calculation periods, the operator was kept informed of progress and possible errors by special features included in the program. At the end of each calculation cycle, print outs showed; (1) dimensions of the furnace, (2) difference between the desired and calculated pressure drops and also tube diameter, and (3) number of tubes in the radiant section and the fuel rate. Obvious errors in any of these values could easily be detected however, the less obvious mistakes could only be detected in the final answers.

Diagrammatic Flow Chart

The backbone of the automatic computation system consists of two separate but complementary computer programs for the solution of the design problem. The first program treats the convection section as an independent unit and makes calculations around tube increments until the gas temperature above an increment reaches a certain maximum. This maximum is set by the bridgewall temperature desired in the radiant section. The second program utilizes this temperature and the exit conditions from the convection section to solve the problem of furnace dimensions, number of tubes required, tube diameter and also the heat balance about the furnace.

The calculation procedure is diagrammed in Figure 10. With the exception of the preliminary calculations (box 1), each step in the diagram corresponds to a series of calculations listed in the smaple problem given on the following pages. In order to shorten machine running time as much as possible certain quantities were evaluated at the beginning of the program to avoid repetition during any cycle for which these values do not change. These are the preliminary calculations and include the evaluation of $1.6G^{1/3}/D'_o{}^{(2/3)}$ from the convection coefficient, Equation (8); the mean beam path of the gases between banks of tubes, Equation (5) and the ratio A_w/A_t.

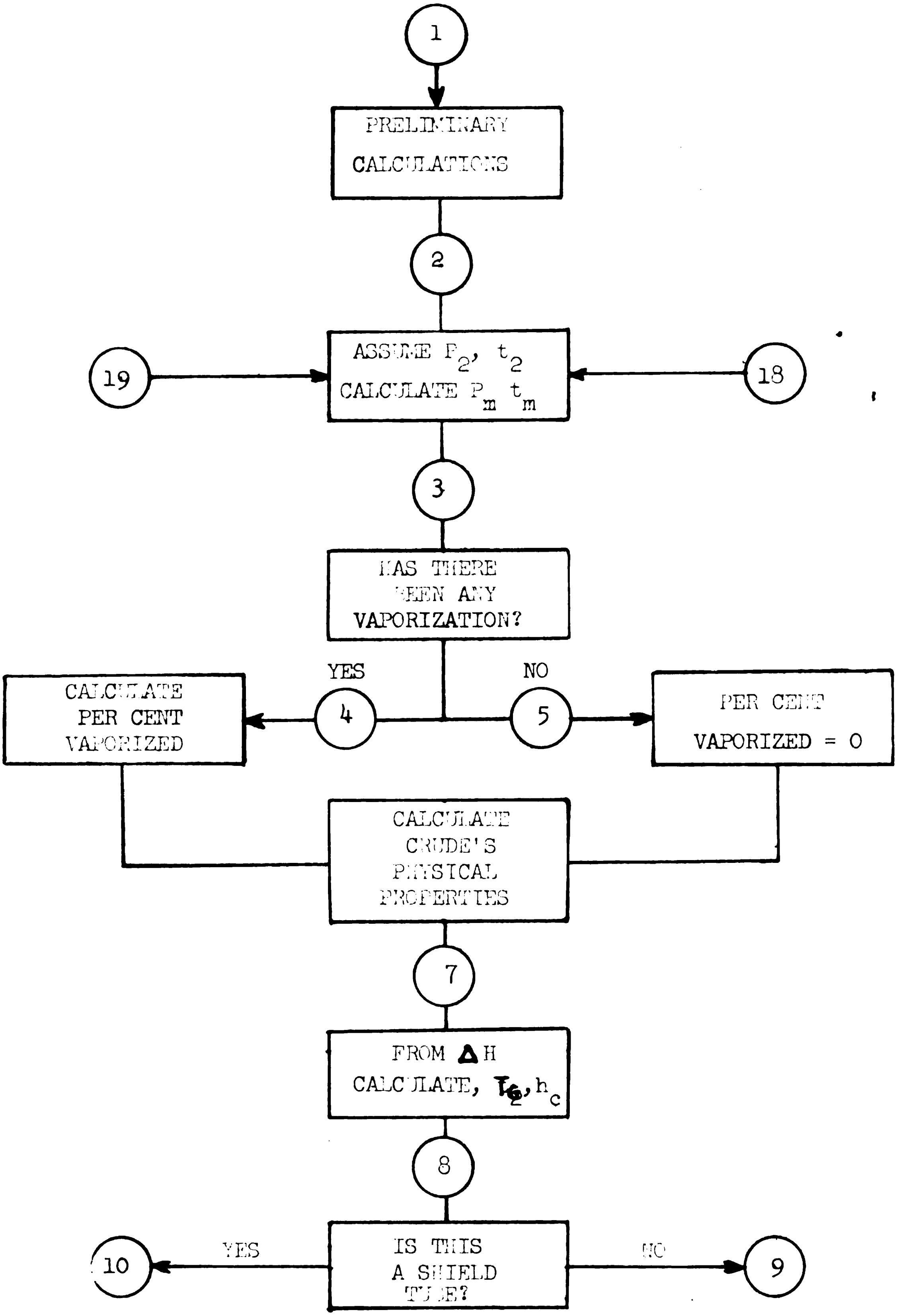

FIGURE 10A. Flow Diagram. Convection Section.

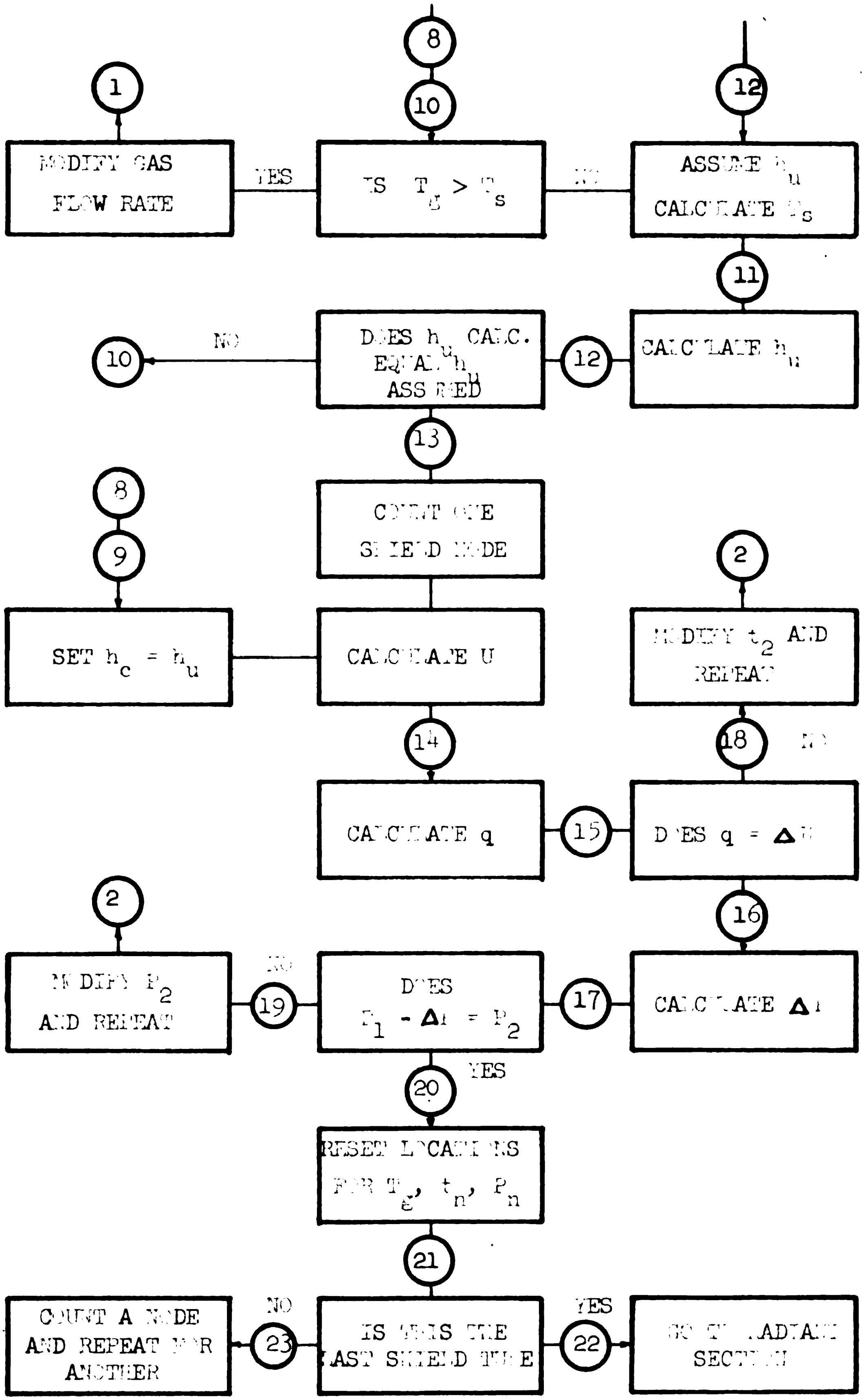

FIGURE 10B. Flow Diagram. Convection Section.

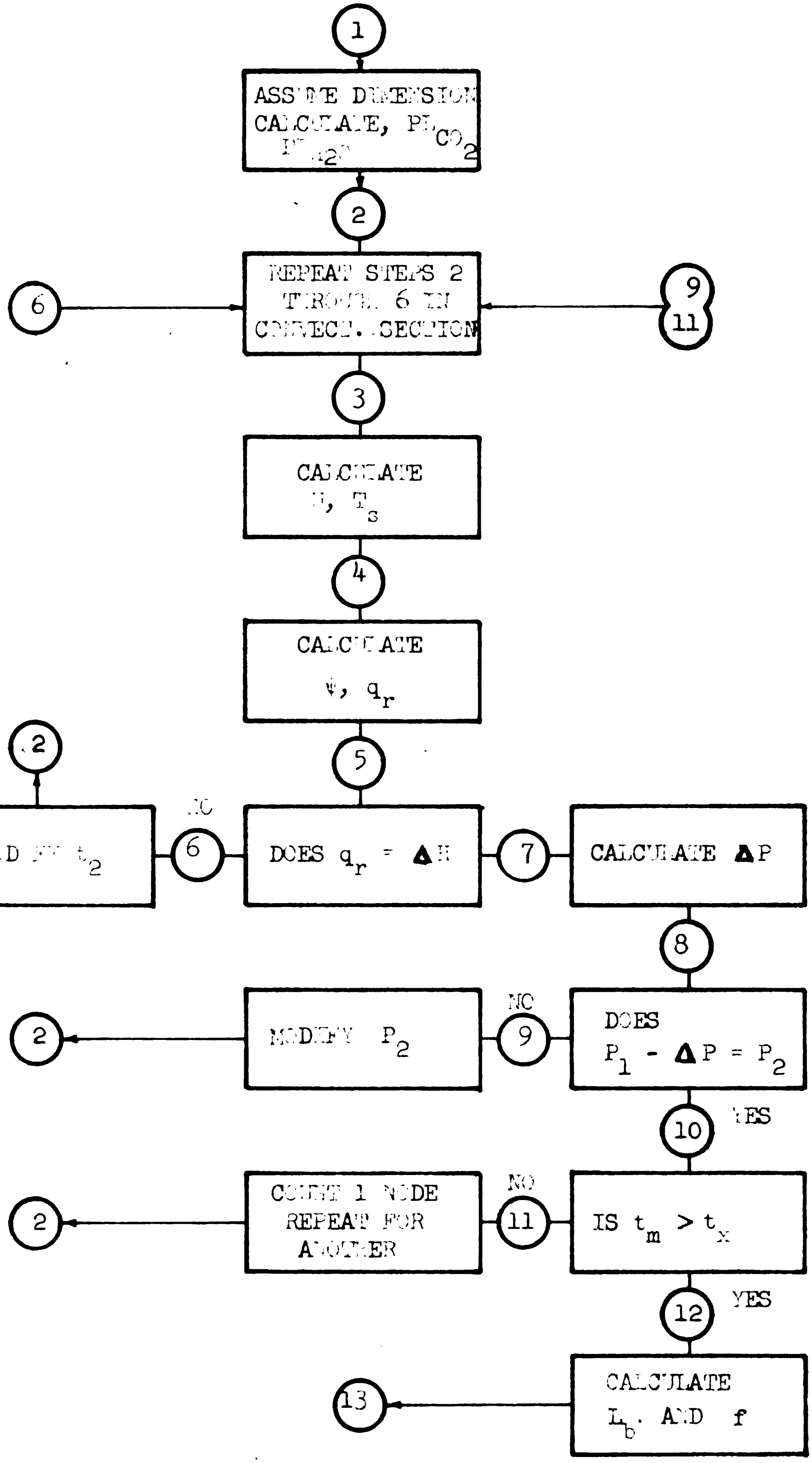

FIGURE 10C. Flow Diagram. Radiant Section.

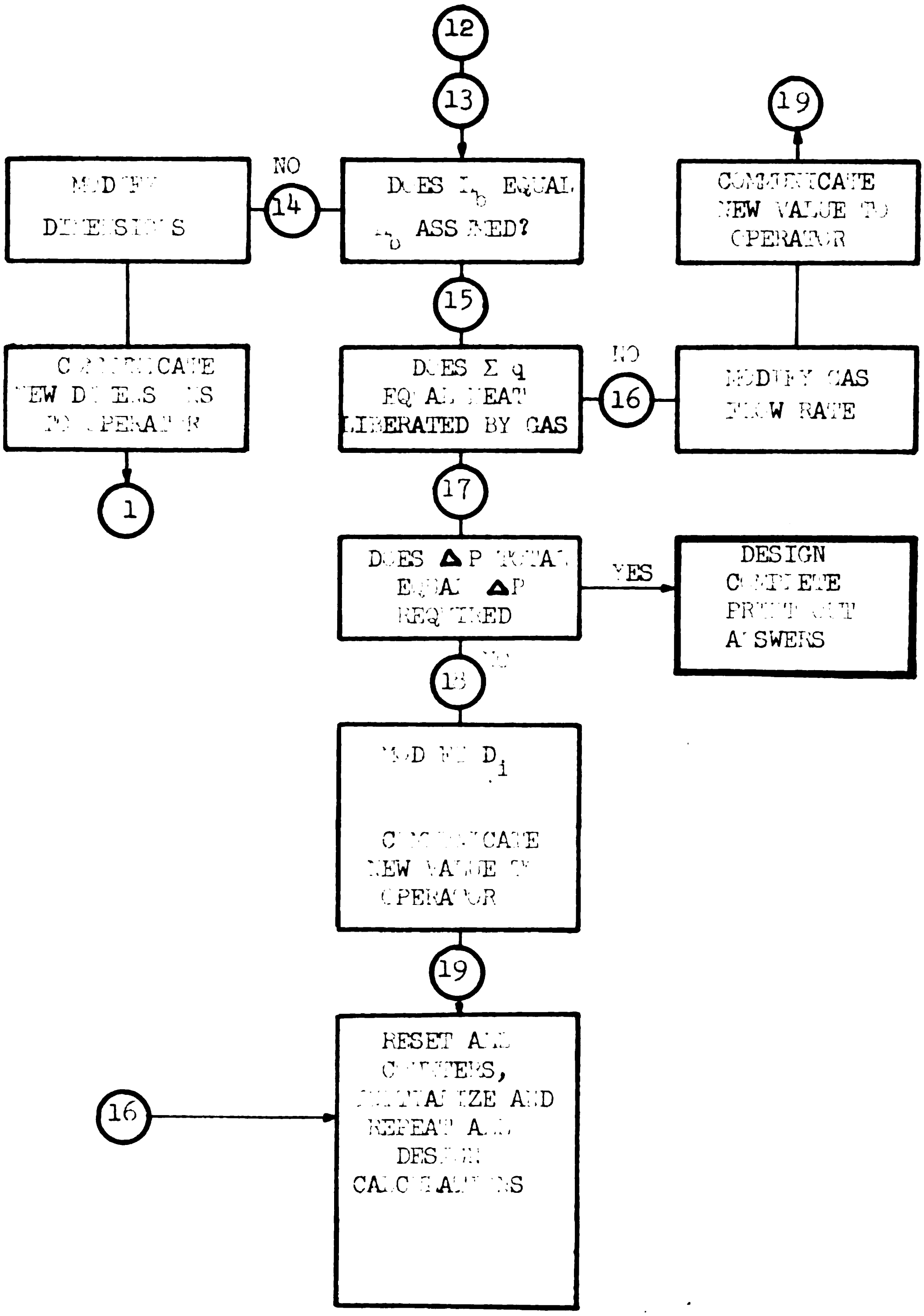

FIGURE 10D. Flow Diagram. Radiant Section.

SAMPLE CALCULATIONS

The following computations serve to illustrate the flow of calculations around a tube node in the convection and radiant sections. It should be clearly understood that this illustration represents only a small fraction of the computations necessary to complete a solution of the furnace design problem.

Furnace data:

Inlet temperature = 400 °F.

Inlet pressure = 240 lb/in^2.

Stack temperature = 870 °F.

Tube spacing = 1.5 tube diameters.

Fuel - CH_4 fired with 50% excess air.

Crude data:

Flow rate = 80,000 lb/hr.

API gravity = 35.7

True boiling point and mid-gravity curves are presented in Figure 11. From the true boiling point curve, a flash curve is drawn at 238 lb/in^2 and is shown in Figure 12.

<u>Calculations</u>

Tube increment = 100 ft.

(2) Assume:

Exit temperature = 409.517 °F.

Exit pressure = 236.29 °F.

Tube diameter (inside) = 0.41 ft.

$\therefore t_m = 404.76$ °F; $P_m = 238.15$ lb/in^2.

(3) Calculate per cent vaporized:

$$d = \frac{t - t_{(b)}}{100m} \qquad \text{(equation for the flash curve)}$$

$$t_{(b)} = t_{(p)} - 50m$$

$t_{(p)}$ is the vapor pressure-temperature equation for a hydrocarbon with an atmospheric boiling point corresponding to the 50% point of the flash curve.

$$t_{(p)} = 426.58 + 3.2P - 0.0113\ P^2 + 15.4 \times 10^{-6}P3$$

Solving for $t_{(p)}$ at p = 238.15 lb/in^2, $t_{(p)} = 759.74$ °F

$\therefore t_{(b)} = 379.74$ °F. Since $t_{(b)} < 404.76$, vaporization has occurred. $d = (404.76 - 379.74) \div (100)\ (7.6) = 0.033$

This result can also be obtained by reading the fraction distilled from the flash curve Figure 12.

(6) Calculate the liquid volume fraction (L.V.F.):

$$L.V.F. = \frac{W_L}{W_L + \frac{W_V \rho_L}{\rho_V}}$$

$$\frac{\left[\frac{W_L}{\rho_L}\right]_{60^\circ F}}{\left[\frac{W_c}{\rho_c}\right]_{60^\circ F}} = (1-d)$$

$W_c = 80,000$ lb/hr $\qquad (SG)_c = 0.8474$

The specific gravity of the liquid can be obtained from the mid-gravity curve (Figure 11), or from the following polynomial at d = 0.033.

$$(SG)_L = 0.599 + 0.965d - 1.487\, d^2 + 1.017\, d^3 = 0.63$$

$\therefore W_L = 57,450$ lb/hr $\qquad W_V = 22,550$ lb/hr

$$\rho_V = \frac{1}{V} = 1728/\left\{\frac{157T}{P} - \frac{1}{T^3}\left[\,7234 + 102P\right] \times 10^7 + 0.52P + 20\right\}$$

$T = 864.76\ ^\circ R \quad P = 238.15$ lb/in^2 $\therefore \rho_V = 2.6$ lb/ft^3

$$(SG)_L = (SG)_L^o + \left(\frac{60-t}{1.8}\right)\left\{\alpha_T + \frac{\beta_T}{1.8}\,[3t - 217]\right\}$$

$$\alpha_T = 0.00122 + 0.00276(SG)_L^o - 0.0079(SG)_L^{o^2} + 0.0046(SG)_L^{o^3}$$

$$\beta_T = 1.92307 \times 10^{-6}(SG)_L^o - 1.5538 \times 10^{-6}$$

$$(SG)_L^o = 0.63$$

$$\therefore \rho_L = (SG)_L\, 62.4 = 29.95$$

$$LVF = \frac{W_L \rho_L}{W_L \rho_V + W_V \rho_L}$$

$$\therefore LVF = 0.018$$

Calculate ΔH:

$$H_L = 15\,(SG)_L - 26 + [0.811 - 0.465\,(SG)_L]t = 0.00029t^2.$$

$$H_V = [215 - 87\,(SG)_L] + [0.415 - 0.104\,(SG)_L]t +$$

$$[3.1 - 0.78\,(SG)_L]t^2 \times 10^{-4} + \frac{1}{9331.7}\Bigg\{$$

$$[20.47P + 0.258P^2 - \frac{P}{T^3} (28.970 + 203.4P)x10^7]\}$$

$$\Delta H = [W_L H_L + W_V H_V]_2 - [W_L H_L + W_V H_V]_1$$

$t_1 = 400°F$; $P_1 = 240$ lb/in^2; $t_2 = 409.517$ °F; $P_2 = 236.29$ °F.

$$\Delta H = 5.35 \times 10^5 \text{ Btu/hr.}$$

(7) Calculate T_g. (Gas temperature above node)

$$\Delta H = \int_{T_1}^{T_2} C_p dT$$

$$C_p = 1.9x10^4 + 1.7T - 9.12x10^{-5}T^2 \qquad \text{Btu/°R}$$

$$5.35x10^5 = 1.9x10^4[T_2 - 1330] + 0.85[T_2^2 - (1330)^2] - 3.04x10^{-5}[T_2^3 - (1330)^3]$$

Newton's method is used to obtain the n^{th} approximation of T_2,

$$T_n = T_{n-1} - \frac{f(T)}{f'(T)} \qquad \text{where } f(T) = \int_{T_1}^{T_2} C_p dT - \Delta H$$

Following this procedure, T_2 is solved for by trial and error.

$$T_2 = 923 \text{ °F.}$$

(9) Calculate $q(= UA\Delta t)$

$$A = \pi D_o \Delta L; \quad ft^2; \quad \Delta t = [tg - tm]$$

$$1/U = \frac{1}{hu} + \frac{D_o(D_o-D_i)}{(D_o+D_i)K_m} + \frac{D_o}{D_i h_i}$$

Km, the thermal conductivity of the tube, is assumed independent of temperature. Km = 26 Btu/ft^2 °F/ft.

$$\therefore \quad hu = \frac{1.6\ G^{2/3} Tg^{0.3}}{D_o'^{1/3}}$$

G = (gas flow rate)/(minimum cross sectional area)

$$= 8.34x10^4/LD_o(\phi-\phi-1)3600 \qquad \text{lb/ft}^2 \text{ sec.}$$

$D_o' = D_o/12$; $Tg = (923 + 870)/2 + 460 = 1356.5$ °R.

$$\therefore \quad hu = 7.8 \quad \text{Btu/hr ft}^2 \text{ °F.}$$

$$\frac{h_i D_i}{K_L} = 0.023 \left(\frac{D\ G}{\mu}\right)_L^{0.8} \left(\frac{C_p \mu}{K}\right)_L^{1/3}$$

$$K_L = \frac{0.813}{12(SG)} [1 - 0.003(t-32)] = 0.095 \quad \text{Btu/hr ft}^2\ {}^\circ\text{F/ft}$$

$$C_p = \frac{\Delta H}{\Delta t} = 0.75 \quad \text{Btu/lb}\ {}^\circ\text{F.}$$

μ_K (kinematic viscosity) $= 5.995 - 0.023t + 3.93\text{x}10^{-4}t^2 - 2.32\text{x}10^{-6}t^3$

$\mu = \mu_K\ (SG)(2.42) = 1.0T$ lb/ft hr.

Solving:

$h_i = 806.9$ Btu/hr ft^2 °F

$u = 7.66$ Btu/hr ft^2 °F.

$q = 7.66\ (\pi)(0.466)(100)[896.5 - 404.76]$

$q = 5.5\text{x}10^5$ Btu/hr.

Since q and ΔH are approximately equal, the assumed t_2 is correct. However, if $q \neq \Delta H$, within tolerance, another value of t_2 would be calculated using the following relationship:

$$t_2^1 = t_2 + (q - \Delta H)(t_2 - t_1)/(\Delta H).$$

and all calculations repeated.

(16) Calculate ΔP.

$$\Delta P = \frac{2\ \phi\ \Delta L\ G^2}{\lambda\ q_c D_i (144)}$$

$\lambda = LVF\ (\rho_L - \rho_V) + \rho_V = 0.18(29.95-2.6) + 2.6 = 7.52$ lb/ft^3

$\phi = 0.0014 + 0.09\ \left(\frac{\mu}{DG}\right)_L^{0.27} = 0.0049$

$\Delta L = 100$ + equivalent length of 4 return bends = 220 ft.

Solving

$\Delta P = 4.2$ lb/in^2

Since $P_2 + \Delta P \doteq P_1$ the assumed pressure is correct.

For $P_2 + \Delta P \neq P_1$ repeat calculations using

$$P_2^1 = P_1 + \Delta P.$$

(10) Shield Section

For tube nodes in the shield section

$$h_u = \frac{(100 + \%\ \text{wall effect})}{100} (h_c + hrg) \quad \text{Btu/hr ft}^2\ {}^\circ\text{F.}$$

$$h_c = \frac{1.6\ G^{2/3} Tg^{0.3}}{D_o'^{1/3}}$$

$$hrg = \epsilon_s \left[\frac{(q_c+q_w)_{Tg} - (q_c+q_w)_{Ts}}{(Tg-Ts)} \right] \frac{(100-\%)}{100}$$

$$\%\ \text{wall effect} = \frac{hrb \times Aw \times 100}{[hc + hrg = hrb]At}$$

$$hrb = 0.00688\ \epsilon_s \left[\frac{T}{100}\right]^3 \quad \text{and } \epsilon_s = 0.95$$

This necessitates a trial and error solution since the tube surface temperature, t_s, is unknown. A solution is obtained by approximating h_u and calculating t_s.

$$t_s = t_m + \Delta t_s$$

$$\Delta t_s = \left[\frac{D_o/Dihi + D_o(D_o-D_i)/(D_o+D_i)Km}{1/h_u + D_o/Dihi + D_o(D_o-D_i)/(D_o+D_i)Km}\right] \Delta t$$

$$\Delta t = tg - tm$$

From t_s, h_u is calculated and compared with the assumed value. If their difference is small, the calculated h_u is used as the correct value. If their difference is not negligible, another value of h_u is assumed and the process repeated.

When a tube node has converged, the calculated values of t_2 and P_2 are used as inlet conditions to the adjacent node and calculations repeated until Tg is within the temperature range assigned to the bridgewall temperature. At this point, calculations for the radiant section begin.

The method of evaluating exit conditions from tube increments in the radiant section is similar to that employed in the convection section. However, q, previously the heat transferred by convection, must be replaced by q_r, the heat transferred to the oil by radiation. The evaluation of q_r is illustrated in the following example.

Length of tube = 25 ft.

Length of node = 50 ft.

Bridgewall temperature = 1391.84 °F.

Inlet temperature = 797.83 °F

Inlet pressure = 51.35 lb/in^2

Per cent excess air = 125%.

(1) Assume:

Dimension of radiant section = 16 x 16 x 25 ft.

Exit temperature = 303.85 °F.

Exit pressure = 53.99 lb/in^2.

(2) Calculate (previously illustrated).

% vaporized = 79.42 %.

ρ_L = 34.086 lb/ft^3.

ρ_V = 0.497 lb/ft^3.

W_V = 6.187x10^4 lb/hr.

W_L = 1.822x10^4 lb/hr.

LVF = 0.00429

ΔH = 4.0839x10^5 Btu/hr.

K_L = 0.0556 Btu/hr ft^2 °F/ft.

μ_L = 0.48 lb/ft hr.

C_p = 0.83 Btu/lb °F.

(3) Calculate t_s

$$\Delta H = u \pi D_o \Delta L [ts-tm]$$

$$1/u = 1/hi + D_o(D_o-D_i)/(D_o+D_i)Km$$

$$\frac{hiDi}{K_L} = 0.023 \left(\frac{DG}{\mu}\right)_L^{0.8} \left(\frac{Cp\mu}{K}\right)^{1/3}$$

D_o = 0.46 ft.; D_i = 0.42 ft.

Solving t_s = 815 °F.

(4) Calculate q_r

$$q_r = \alpha A_{cp} \psi \left\{0.173x10^8 [Tg^4 - Ts^4] + 7 [Tg-Ts]\right\}$$

L_B = 2/3 [25 x 16 x 16]$^{1/2}$ = 12.28 ft.

P_{CO_2} = 0.0466 atms. P_{H_2O} = 0.0892 atms.

$P_L(CO_2)$ = 0.5493; $PL(H_2O)$ = 1.099 atm. ft.

A tube spacing of 1.5 tube diameters corresponds to

α = 0.97 (Figure 3)

$\therefore \alpha A_{cp} = \alpha$ L $\cancel{¢¢}$ D_o = 33.5 ft^2

$$\epsilon_G = \left[\frac{(q_c + q_w)_{Tg} - (q_c + q_w)_{Ts}}{(q_b)_{\bar{T}g} \quad (q_b)_{Ts}.} \right] \left[\frac{100 - \%}{100} \right] \quad \text{Btu/hr.}$$

To obtain q_c and q_w at their respective PL values, it is necessary to interpolate between two polynomials. The following interpolation formula is used.

$$f(x) = f(x_1) + \frac{P - P_1}{P_2 - P_1} [f(x_2) - f(x_1)]$$

$P_1 < P < P_2$ and $f(x_2) > f(x_1)$.

P = desired parameter (PL value), $f(x_1)$ and $f(x_2)$ are the polynomials corresponding to parameters P_1 and P_2 respectively.

Emission due to CO_2 molecules:

$2q_c$ (at PL = 0.4) = $4823.29 - 12.153t + 0.0097198t^2 - 7.351 \times 10^{-5}t^3$

$2q_c$ (at PL = 0.6) = $7588 - 18.258t + 0.013484t^2 - 1.1916 \times 10^{-4}t^3$

Emission due to H_2O molecules:

$2q_w$(at PL=1.0) = $622.9 - 1.6958t + 0.0043397t^2 + 1.848 \times 10^{-4}t^3$

$2q_w$(at PL=1.25) = $2283.7 + 5.2497t - 0.000627t^2 + 3.4113 \times 10^{-4}t^3$

Black body radiation:

$2q_b = -5926 + 27.632t - 0.03172t^2 + 2.550 \times 10^{-3}t^3$

Polynomials are also available for the percent correction. These polynomials are identified by the parameter $S = P_{cL} + P_{wL}$, and evaluated as functions of $R = CO_2/(H_2O + CO_2)$. Interpolation is again necessary.

$P_{cL} + P_{wL} = 1.648$

$$R = \frac{CO_2}{CO_2 + H_2O} = 0.334$$

$2(\%)$, (at S=1) $= 2.5018 + 49.2874R - 92.454R^2 + 102.165R^3 - 52.9670R^4$

$2(\%)$, (at S=2.0) $= 4.232 + 51.983R - 101.885R^2 + 111.444R^3 - 54.630R^4$

Solving for ϵ_G:

$\epsilon_G = 0.394.$

$$A_R/\alpha A_{cp} = \frac{2Z\ (Z + 2L)}{\alpha L\ (2Z + f)} \quad -1$$

If tubes are placed on the Bridge wall, f = 2/3Z. If not, f = 0. The first iteration through the radiant section is made with f = 0. After convergence to the correct discharge temperature, f is calculated from the number of tubes in this section. The magnitude of f determines whether tubes should be placed on the bridge wall.

$$A_R/\alpha A_{cp} = \frac{2 \times 16 [16 + 2 \times 25]}{0.97 \times 25 [2 \times 16 + 0]} - 1 = 1.72$$

$$2\psi, \text{ (at } \epsilon_G = 0.38) = 0.7336 + 0.3505 \left(\frac{A_R}{\alpha A_{cp}}\right) - 0.0495 \left(\frac{A_R}{\alpha A_{cp}}\right)^2 + 0.0024 \left(\frac{A_R}{\alpha A_{cp}}\right)^3$$

$$2\psi, \text{ (at } \epsilon_G = 0.40) = 0.7671 + 0.3792 \left(\frac{A_R}{\alpha A_{cp}}\right) - 0.06246 \left(\frac{A_R}{\alpha A_{cp}}\right)^2 + 0.0037 \left(\frac{A_R}{\alpha A_{cp}}\right)^3$$

Solving

$$\psi = 0.6152.$$

$$q_r = (33.5)\ (0.6152) \left\{ 0.173 \times 10^{-8} \ [(1851.4)^4 - (1275)^4] + 7\ [1851.4 - 1275] \right\} = 4.22 \times 10^5 \text{ Btu/hr.}$$

Since $q_r \doteq \Delta H$, t_2 will not be recalculated. Calculations for ΔP are similar to those performed in the convection section. Calculations are repeated until the exit temperature, t_2, from a tube node is equal to the desired discharge temperature from the still.

(12) Check furnace dimensions:

N_a = number of tubes calculated from assumed dimensions

$$N_a = \frac{2Z + f}{\not{c}\ \not{c}\ D_o} = \frac{(2)\ (16)}{(15)\ (0.46)} = 46 \text{ tubes.}$$

In this example, 48 tubes were counted in the radiant section. Since N_a = 48, the assumed dimension Z is approximately correct and will not be recalculated. If $N_a \neq 48$, N_a in the above equation would be replaced by 48 and Z calculated. This Z could be used during the next iteration through the radiant section.

Calculate f.

$$f = 48\ (\not{c}\not{c})\ D_o - 2Z.$$

(a) If $f < L/2$, tubes are not placed in the bridge wall, consequently f is set equal to zero.

(b) If $f > L/2$, tubes are placed in the bridge wall and f is set equal to $2/3Z$. Since $f = 48\ (1.5)\ (0.46) - 32 = -3.2 < L/2$, the initial assumption, $f = 0$, is correct. Calculations will not be repeated. When case (b) exists, the iteration is repeated using $Z = 3/8\ [N_c \not{c}\not{c}\ D_o]$. N_c = number of tubes counted in the radiant section.

(15) Heat balance:

In this example

heat absorbed in the radiant section = 1.06017×10^7 Btu/hr.

heat absorbed in the convection section = 1.2835×10^7 Btu/hr.

total heat lost = 2.75988×10^7 Btu/hr.

total heat liberated = 5.10355×10^7 Btu/hr.

From the flow rate of the fuel and its net heating value

(584.2 Btu/lb. gas) (83354.9 lb./hr.) = 4.85×10^7 Btu/hr.

Since the total heat liberated as calculated from the heat balance and the net heating value of the fuel are almost equal, it will not be necessary to repeat the calculations.

(18) Tube diameter:

Finally, P_2 is compared with the desired discharge pressure. If their difference is not within tolerance another tube diameter is assumed and all calculations repeated. The assumption is made using the relationship:

$$Di_n = Di_{n-1} \left[\frac{\Delta P_c}{\Delta P_a} \right]^{1/5}$$

ΔP_c = calculated pressure drop after n-1th iterations

ΔP_a = desired pressure drop

Di_n = diameter assumed for the nth iteration

Di_{n-1} = diameter previously used.

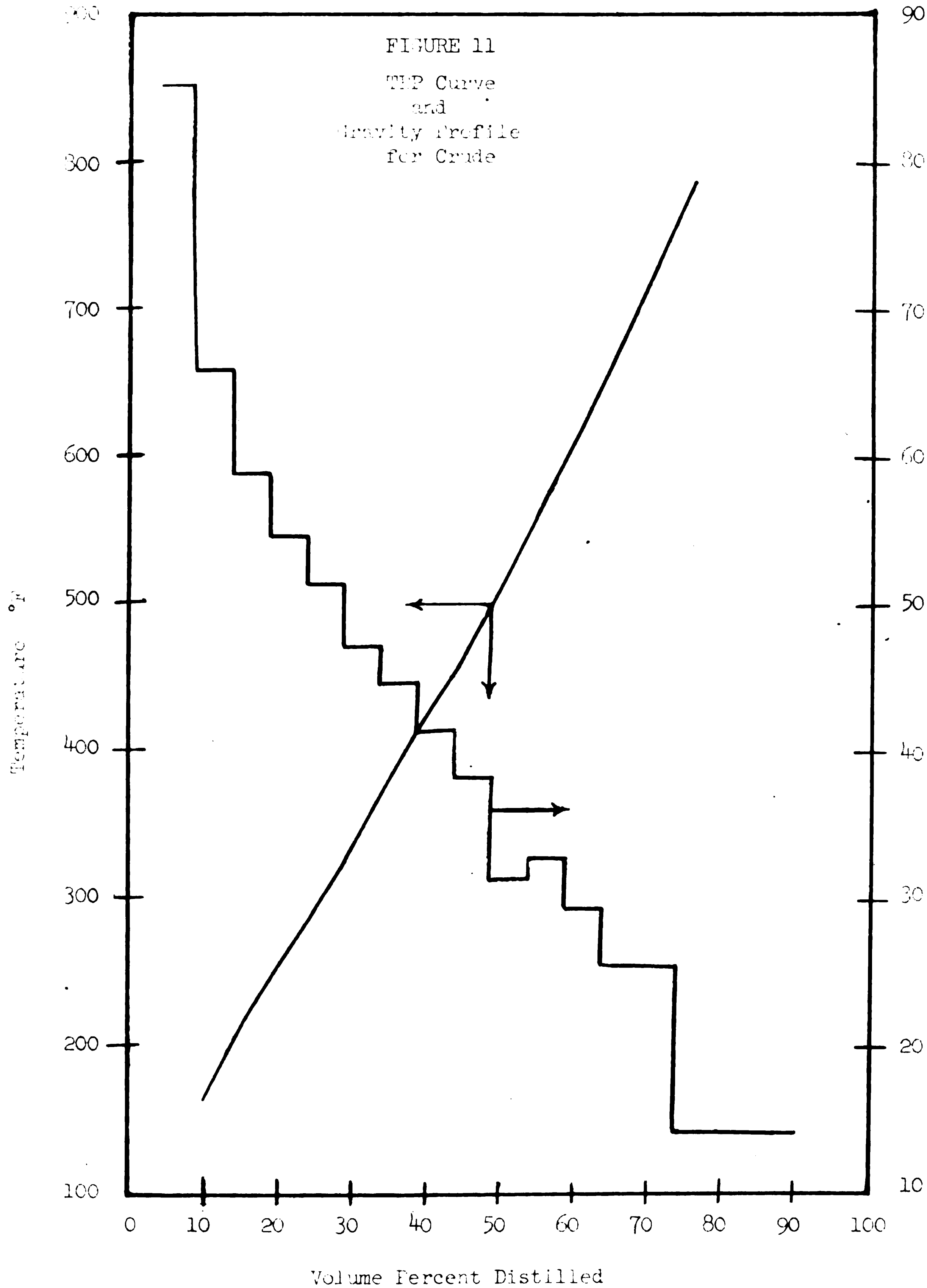

FIGURE 11

TBP Curve
and
Gravity Profile
for Crude

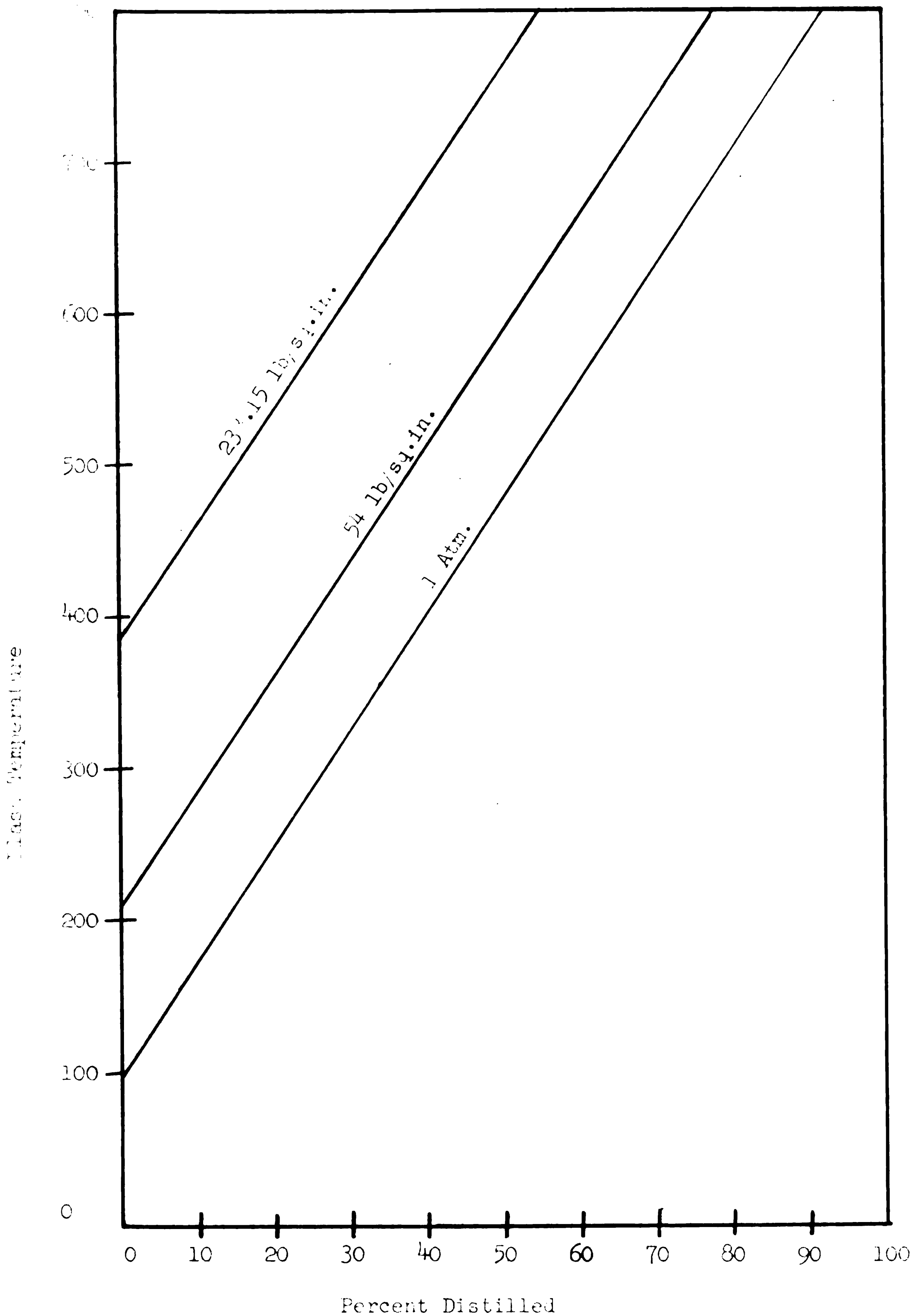
600
500
400
300
200
100
0
54 lb/sq.in.
1 Atm.
0
10
20
30
40
50
60
70
80
90
100
Percent Distilled

FIGURE 12

FIXED POINT ORDERS

ORDER	OPERATION
L5 n	Transfer contents of location n to A.
50 n	Transfer contents of location n to Q.
L4 n	Add contents of n to A.
L0 n	Subtract contents of n from A.
75 n	Multiply Q by contents of n.
40 n	Transfer contents of A to n.
S5	Transfer contents of Q to A.
22 n	Transfer control to right order at location n.
32 n	If contents of A $\geq$ 0, execute 22.
26 n	Transfer control to left order at location n.
36 n	If contents of A $\geq$ 0, execute 26.
F5 n	Transfer contents of n to A and increase the address digits at the right side by 1.

FLOATING POINT ORDERS:

Let F be the floating decimal number in the floating accumulator and let F(n) be the floating decimal number in location n.

ORDER	OPERATION
80 n	Replace F by F - $F_{(n)}$
81 n	Replace F by -$F_{(n)}$
82 n	Transfer control to the right hand interpactive order in n if F$\geq$ 0
83 n	Transfer control to the left hand interpactive order in n if F $\geq$ 0
84 n	Replace F by F + $F_{(n)}$
85 n	Replace F by $F_{(n)}$

ORDER	OPERATION
86 n	Replace F by $F/F_{(n)}$
87 n	Replace F by $F \times F_{(n)}$
88 0	Replace F by one number read from the input tape punched as sign, any number of decimal digits, sign, and two decimal digits to represent the exponent.
89 n	Punch or print F as a sign, n decimal digits, sign, two decimal digits to represent the exponent and two spaces.
8K n	Replace F by n if $0 \leq n < 200$
8S n	Replace $F_{(n)}$ by F
8N n	Replace F by $/F/ - /F_{(n)}/$
8J n	Transfer control to the fixed point order at the left side of location n
8F n	Give a carriage return and line feed and arrange to print a block of numbers having n columns.

If the first function digit of a floating point order is 0, 1, ...7, it refers to one of a set of control registers, or b-registers in the floating decimal routine which are similarly numbered. These are used to count the number of passages through loops and for increasing the addresses of these orders on successive passages. These addresses are increased only if the function digit of the order corresponds to that of the control register currently used.

ORDER LIST WITH $b \neq 8$

The index Cb is used for counting purposes to determine the number of passages through a loop. The index gb is used for advancing the address of floating point orders.

ORDER	OPERATION
b0 n	Replace F by F - F(n + gb)
b1 n	Replace F by -F(n + gb)

ORDER	OPERATION
b2 n b3 n	Replace gb, Cb by gb + 1, Cb + 1 Then transfer control to the right hand (if b2 n) or left hand floating point (if b3 n) order in n if Cb +1 < 0. This transfer is used at the end of the loop.
b4 n	Replace F by F + F(n + gb)
b5 n	Replace F by F(n +gb)
b6 n	Replace F by F/F(n + gb)
b7 n	Replace F by F x F(n + gb)
bk n	Replace gb, Cb by 0, -n. This floating point order is used for preparing L cycle around loop n times
bS n	Replace F(n + gb) by F
bN n	Replace F by F - F(n + gb)
bL n	Replace gb, Cb by gb + n, Cb. This order is used when one wishes to change addresses by some increment other than +1 in a loop. If one places bL 1022 in a loop, the effect will be to decrease addresses by two on each passage.

FURNACE DESIGN PROGRAM

Program Outline

LOCATION	CONTENTS
3	Specifies Floating Accumulator
4	Specifies A1 routine
5	Specifies A3 routine
6	Specifies SA3 (Natural Log) routine
7	Specifies SA2 (Exponential) routine
8	Specifies routine to calculate physical properties
9	Specifies routine which selects correct polynomials to be used in evaluating ψ
10	Specifies location of data
11	Specifies location of generated data
12	Specifies location of data
13	
14	Specifies location of routine to calculate ϵ_G
15	Specifies location of routine to calculate ΔP
16-19	Temporary storages
20-157	Convection section routine
158-270	Radiant section routine
635-660	Continuation of radiant section routine
271-379	Subroutine to calculate physical properties
380-449	Subroutine to select correct polynomials
450-508	Data
509-546	Data (depending on type crude)
547-589	Data (generated)
590-634	Polynomials for ψ
665-678	Routine to calculate ϵ_G

LOCATION	CONTENTS
679-704	Routine to calculate ΔP
705-738	Storages for selected polynomials and their parameters
739-743	Answer storages
739	Counter for shield section
740	Number of rows in convection section
741	Heat absorbed in convection section
742	Heat absorbed in radiant section
743	Number of tubes in radiant section
744	Heat lost from furnace

PROGRAM

003K	Directive Specifying Location Routines
00F 00745F	Floating accumulator
00F 00830F	A1
00F 00773F	A3
00F 00800F	SA3
00F 00747F	SA2
00F 00271F	Calculate physical properties
00F 00380F	Select correct polynomials
00F 00509F	Data
00F 00547F	Data
00F 00450F	Data
00F 00651F	
00F 00665F	Calculate ϵ_G
00F 00679F	Calculate ΔP

LOCATION	ORDER	NOTES
0020K		
0	22 L 50 L	
1	26 S4 0K 45F	Read in and store constants for This routine is later overread
2	88 F 0S 590F	
3	03 2L 8J 999F	Transfer control to continue reading and storing program.
00555K		
0	22 L 50 L	

LOCATION	ORDER	NOTES
1	26 S4	
	OK 23F	Read in and store constants
2	88 F	
	OS 100F	
3	03 2L	
	OK 59F	
4	88 F	
	OS SN	
5	03 4L	
	OK 38F	
6	88 F	
	OS SK	
7	03 6L	
	8K 1F	
8	8K 1F	
	83 20F	Transfer control to location 20
	24 555N	Start executing program at location 555
0020K		Program for the Convection Section
0	85 52SN	Preliminary calculations
	87 SK	
1	84 53SN	
	85 SS	
2	OK 5F	
	05 1SK	
3	OS 1SS	
	02 2L	
4	OK 2F	
	85 45SN	Calculate PL of CO_2 and H_2O for the convection bank
5	87 SS	
	07 21SK	
6	OS 28SS	
	02 4L	
7	8J 8L	
	8J 8L	

LOCATION	ORDER		NOTES
8	41 F5	739F 8L	Entrance to subroutine to select correct polynomials for ϵ_G
9	24 OK	S9 4F	Transfer control to subroutine
10	8K 8S	F 19SK	
11	87 04	3SK 16SK	
12	03 8S	11L 744F	
13	8K 8S	F 36SS	
14	85 80	6SK 1SN	
15	87 87	SS 7SK	
16	8S 85	5F 15SK	
17	86 86	5F SN	
18	8S 87	5F 5F	
19	86 86	SS 36SN	
20	8J 86	S6 3SN	
21	8J 87	S7 4SN	
22	8S 8K	15F 1F	
23	8K 84	1F 14SK	
24	86 87	14SK 6SK	
25	86 8S	41SN 32SS	

LOCATION	ORDER		NOTES
26	85	4SS	Assume P_2
	84	5SS	
27	86	2SN	
	8S	8SS	Store P_m
28	85	1SS	Assume T_2
	84	2SS	
29	86	2Sn	
	8S	7SS	Store t_m
30	OK	4F	
	8K	F	
31	8S	19SK	
	87	3SS	
32	04	16SK	
	02	31L	
33	8S	17F	
	8J	34L	Entrance to subroutine to calculate physical properties
34	22	34L	
	F5	34L	
35	26	S8	Transfer control to this subroutine
	85	21SS	Calculate temp. of flue gases above nth tube (T_g)
36	84	17F	
	8S	5F	
37	81	5F	
	8S	19SK	
38	OK	3F	
	1K	F	
39	05	16SK	
	17	3SN	
40	OS	8F	
	1L	1Q23F	
41	03	39L	
	OK	3F	
42	8K	F	
	87	20SK	Assume T_g
43	04	8F	
	02	42L	

LOCATION	ORDER		NOTES
44	8S	5F	
	OK	4F	
45	8K	F	
	87	20SK	
46	04	16SK	
	02	45L	
47	8S	6F	
	85	24SN	
48	8N	6F	Is this the correct temperature?
	82	52L	Yes: Transfer control to 52L
			No: Modify T_g
49	85	6F	
	86	5F	
50	8S	5F	
	85	20SK	
51	80	5F	
	8S	20SK	
52	82	41L	Repeat calculations using new T_g
	85	20SK	Calculate h_c
53	84	3SS	
	86	2SN	
54	8S	26SS	
	8J	S6	
55	87	44SN	
	8J	S7	
56	87	15F	
	8S	27SS	
57	85	26SS	
	80	10SN	
58	8S	31SS	
	80	1009F	Is this a shield tube?
			Yes: Transfer control to 64L
59	82	64L	No: Continue at 100L without including
	8K	1F	calculations for a shield tube
60	83	120F	
	85	4SS	These orders are used in the radiant section
			$P_1 - P_x \geq 0$?
61	80	1002F	
	84	1008F	

LOCATION	ORDER		NOTES
62	83	250F	Yes: Continue calculations for another node
	89	4F	No: Point out their difference and also the tube diameter. Then change the tube diameter and repeat all calculations.
63	85	SK	
	89	4F	
64	83	645F	Go to 645 to change diameter.
	84	1F	Calculations for shield tubes begin
65	84	739F	Increment counter for shield tubes
	8S	739F	
66	85	SS	Calculate T_s
	84	SK	
67	87	24SK	
	85	5F	
68	85	SS	
	80	SK	
69	87	SS	
	86	5F	
70	8S	5F	
	85	SS	
71	86	SK	
	86	25SS	
72	84	5F	
	8S	5F	
73	8K	1F	
	86	23SK	Assume h_u
74	84	5F	
	8S	6F	
75	85	31SS	
	80	7SS	
76	87	5F	
	86	6F	
77	84	7SS	
	8S	30SS	
78	85	31SS	
	80	30SS	

LOCATION	ORDER		NOTES
79	8S	16F	
	8J	80L	Entrance to subroutine to evaluate $[(q_c + q_w)_{T_g} - (q_c + q_w)_{T_s}] \left[\frac{100-\%}{100}\right]$
80	22	80L	
	F5	80L	
81	26	SF	Transfer control to subroutine
	85	10F	Evaluate h_{rg}
82	86	16F	
	87	46SN	
83	8S	13F	
	8K	100F	Evaluate h_{rb}
84	8S	5F	
	85	30SS	
85	84	10SN	
	86	5F	
86	8S	6F	
	87	6F	Evaluate percent wall correction
87	87	6F	
	87	46SN	
88	87	47SN	
	8S	6F	
89	84	13F	
	84	27SS	
90	8S	7F	
	85	6F	
91	87	32SS	
	86	7F	
92	84	1SN	
	8S	8F	
93	85	13F	
	84	27SS	
94	87	8F	
	8S	34SS	
95	80	23SK	Is the h_u assumed equal to h_u calculated
	8S	5F	
96	85	39SN	
	8N	5F	

LOCATION	ORDER		NOTES
97	83	99L	Yes: Transfer control to 99L
	85	34SS	No: Modify h_u
98	8S	23SK	
	83	66L	repeat calculations using new h_u
99	85	34SS	
	8S	27SS	27SS = h_c if n^{th} tube is not a shield tube
100	85	SS	
	84	SK	27SS = (wall correction)(h_c+h_{rg}) if n^{th} tube is a shield tube
101	87	24SK	
	8S	5F	
102	85	SS	Calculations for **U**
	80	SK	
103	87	SS	
	86	5F	
104	8S	5F	
	85	SS	
105	86	SK	
	86	25SS	
106	84	5F	
	8S	5F	
107	8K	1F	
	86	27SS	
108	84	5F	
	8S	5F	
109	8K	1F	
	86	5F	
110	8S	35SS	
	87	SS	Calculate q = uAΔt
111	87	7SK	
	87	41SN	
112	8S	5F	
	85	31SS	
113	80	7SS	
	87	5F	
114	8S	5F	
	80	21SS	Does q = ΔH?

LOCATION	ORDER		NOTES
115	8S	6F	
	85	1004F	
116	8N	6F	
	83	126L	Yes: continue at 126L
			No: modify t_2
117	85	2SS	
	80	1SS	
118	87	6F	
	86	21SS	
119	84	2SS	
	8S	2SS	
120	8K	1F	If t_2 does not converge after the 7^{th}
	84	40SS	iteration use last calculated value of
			t_2 as the correct value
121	8S	40SS	
	80	1008F	
122	83	126L	No convergence after 7^{th} iteration, proceed
	85	22SN	126L
123	80	739F	Was this a shield tube?
	83	48F	No: repeat calculations using modified t_2
			Yes: reset counter
124	85	739F	
	80	1SN	
125	8S	739F	
	83	48F	Repeat calculations using modified t_2
126	85	7SK	
	86	14SK	
127	8S	38SS	
	85	14SK	
128	8S	39SS	
	8J	3F	Reset counter for number of iterations t_0 t_2
129	JO	130L	
	F5	139L	
130	26	SL	Transfer control to subroutine to calculate ΔP
	85	20SK	at this point the tube node has converged
131	8S	3SS	
	85	9SS	Make a summation of the ΔHn's
132	8S	36SS	
	85	21SS	

LOCATION	ORDER		NOTES
133	84	741F	
	8S	741F	
134	8K	1F	Make a summation of the number of rows (n's)
	84	740F	
135	8S	740F	
	85	6SN	Was this the last tube in the shield section?
136	80	739F	
	83	26L	No: repeat calculations for another tube Yes: Proceed to the radiant section.
137	8K	2F	
	8S	39SS	
003K			
0	41	40SS	Reset counter for the number of iterations
	26	149F	of t_2 to zero.
			Program For the Radiant Section
00158K			
0	85	20SK	Store T_g as the bridgewall temperature
	80	10SN	
1	8S	31SS	
	85	1SS	Store t_1 as the cross-over temperature
2	8S	6SS	
	85	4SS	Store P_1 as the cross-over pressure
3	8S	37SS	
	8K	F	Set counter for ΔH_R and $n_R = 0$
4	8S	742F	
	8S	743F	
5	85	6SS	
	8S	1SS	
6	85	37SS	
	8S	4SS	
7	85	38SS	
	87	1000F	Assume Z, and calculate L_B
8	87	1000F	
	8J	S6	
9	86	3SN	
	8J	S7	

LOCATION	ORDER		NOTES
10	87	2SN	
	86	3SN	
11	8S	5F	Calculate PL of CO_2 and H_2O in the radiant section
	87	21SK	
12	8S	28SS	
	85	5F	
13	87	22SK	
	8S	29SS	
14	8K	2F	Calculate $A_R/\alpha A_{cp}$ for the radiant section
	87	1000F	
15	8S	5F	
	84	42SN	
16	87	38SS	
	87	1001F	
17	8S	6F	
	8K	2F	
18	87	38SS	
	84	1000F	
19	87	5F	
	86	6F	
20	80	1SN	
	8S	32SS	
21	85	38SS	Calculate αA_{cp} per node
	87	39SS	
22	87	1001F	
	87	SS	
23	87	6SK	
	8S	41SS	
24	85	SS	
	84	SK	
25	87	24SK	
	8S	5F	
26	85	SS	
	80	SK	
27	87	SS	
	86	5F	

LOCATION	ORDER		NOTES
28	8S	19F	
	8J	29L	
29	22	29L	
	F5	29L	Select correct polynomials for ϵ_a
30	26	S9	
	85	4SS	Assume P_2
31	84	5SS	Calculate P_m
	86	2SN	
32	8S	8SS	
	85	1SS	Assume t_2
33	84	2SS	Calculate t_m
	86	2SN	
34	8S	7SS	
	8J	35L	
35	22	35L	
	F5	35L	
36	26	S8	Calculate percent vaporized, H_L, H_V, etc.
	8K	1F	Calculate T_s from $q = uA[t_s - t_m]$
37	86	25SS	
	84	19F	
38	8S	5F	
	85	21SS	
39	87	5F	
	86	SS	
40	86	41SN	
	86	39SS	
41	86	38SS	
	84	7SS	
42	8S	30SS	
	8J	43L	
43	22	43L	
	F5	43L	Calculate $[(q_c + q_w)_{T_B} - (q_c + q_w)_{T_s}]\left[\frac{100-\%}{100}\right]$
44	26	SF	
	2K	2F	Calculate $(q_{BB})_{T_B} - (q_{BB})_{T_s}$

LOCATION	ORDER		NOTES
45	OK4F		
	8K	F	
46	27	30SS	
	04	54SN	
47	03	46L	
	2S	5F	
48	23	45L	
	80	5F	
49	8S	6F	
	85	10F	Evaluate ϵ_G
50	86	6F	
	8S	10F	
51	OK	9F	Select correct polynomial for ψ
	1K	F	
52	15	590F	
	80	10F	
53	83	59L	
	8S	718F	
54	3K	4F	
	15	591F	
55	3S	710F	
	1L	1F	
56	32	54L	
	1L	1F	
57	03	52L	
	85	46SN	
58	8S	33SS	
	83	72L	
59	8S	719F	
	3K	4F	
60	15	591F	
	3S	714F	
61	1L	1F	
	33	60L	
62	2K	2F	Interpolate between the polynomials with
	OK	F	values of ϵ_{Gn} closest to ϵ_G

LOCATION	ORDER		NOTES
63	1K	4F	
	8K	F	
64	87	32SS	
	04	710F	
65	OL	1F	
	13	64L	
66	2S	8F	
	23	63L	
67	80	8F	
	8S	7F	
68	85	719F	
	80	718F	
69	8S	5F	
	81	718F	
70	86	5F	
	87	7F	
71	84	8F	
	8S	33SS	Store ψ

00230K

0	2K	2F	Evaluate q_r
	25	30SS	
1	84	10SN	
	8J	S6	
2	87	40SN	
	8J	S7	
3	2S	5F	
	22	L	
4	80	5F	
	87	1018F	
5	8S	6F	
	85	31SS	
6	80	30SS	
	87	1008F	
7	84	6F	
	87	33SS	

LOCATION	ORDER		NOTES
8	87	41SN	
	80	21SS	Is q_r = ΔH?
9	8S	6F	
	85	1004F	
10	8N	6F	
	82	14L	Yes: proceed to 14L NO: modify t_2
11	85	2SS	
	80	1SS	
12	87	6F	
	86	21SS	
13	84	2SS	
	8S	2SS	
14	82	190F	Repeat calculations using new t_2
	8J	15L	
15	J0	16L	
	L5	15L	
16	26	SL	Calculate ΔP
	8K	2F	At this point the tube node has converged.
17	84	743F	Make a summation of n_R's
	8S	743F	
18	85	742F	Make a summation of the ΔH_R's
	84	21SS	
19	8S	742F	
	82	80F	Transfer control to location 80
20	85	1003F	is $t_x - t_2 > 0$
	80	1SS	
21	82	188F	Yes: Repeat calculations for another node
	8K	2F	No: Stop calculating nodes and test for convergence of furnace
22	87	1000F	
	8S	5F	
23	85	743F	
	87	6SK	
24	87	SS	
	8S	9F	
25	80	5F	
	87	2SN	If $2f - dL < 0$ use $f = 0$ $2f - dL > 0$ use $f = 2/3\ Z$

LOCATION	ORDER		NOTES
26	80	38SS	
	83	30L	
27	8K	F	Use f = 0
	8S	42SS	
28	85	9F	
	86	2SN	
29	8S	9F	
	82	33L	
30	85	9F	Use f = 2/3 Z
	87	44SN	
31	86	42SN	
	8S	9F	
32	87	2SN	
	86	3SN	
33	8S	42SS	
	85	1000F	Does Z assumed = Z calculated?
34	80	9F	
	8N	42SN	
35	82	639F	No: proceed to location 639
	85	741F	Yes: Is the assumed flue gas flow rate correct?
36	84	742F	
	84	744F	
37	86	1004F	
	8S	12F	
38	86	15SK	
	80	1SN	
39	8S	5F	
	8K	1F	
40	83	635F	Continue these calculations at 635
	83	635F	

00635K

0	85	39SN	
	8N	5F	
1	82	6L	Yes: Continue at 641
	8F	2F	No: print out N_R last calculated and flue gas flow rate last used

LOCATION	ORDER		NOTES
2	85	743F	
	89	3F	
3	85	12F	
	89	6F	
4	83	3SJ	Proceed to modify flue gas flow rate
	85	9F	Modify Z
5	8S	1000F	
	89	5F	Print out modified Z
00641K			
0	82	161F	Repeat calculations in radiant section using
	85	4SS	Does $P_2-P_x = 0$ within limits? new Z
1	80	1002F	
	89	3F	Print out difference
2	8S	5F	
	85	1008F	
3	8N	5F	
	82	8L	Yes: Furnace design complete, proceed to 8L
4	85	14SK	No: modify tube diameter
	80	4SS	
5	86	1005F	
	8J	S6	
6	86	1006F	
	8J	S7	
7	87	SK	
	8S	SK	Using new diameter, repeat all calculations. First, reset all counters to zero at 7SJ.
8	82	7SJ	
	85	1000F	Furnace design complete. Print out answers.
9	89	8F	
	OK	6F	
10	05	739F	
	89	8F	
11	03	10L	
	8J	12L	
12	OF	F	Stop all calculations
	OF	F	

LOCATION	ORDER			NOTES
13	86	15SK	(3SJ)	Modify gas flow rate
	8S	10F		
14	OK	3F		
	05	16SK		
15	87	10F		
	OS	16SK		
16	02	14L		
	85	12F		
17	8S	15SK		
	OK	6F		Set all counters to zero
18	8K	F		
	OS	739F		
19	03	18L		
	83	20F		Repeat all calculations from beginning
				Subroutine for Physical Properties
00271K				
0	42	106L		
	50	L		
1	26	S4		
	OK	4F		Calculate percent vaporization
2	8K	F		
	87	SS		
3	04	26SK		
	02	2L		
4	8S	5F		
	85	8SK		
5	87	5SN		
	8S	6F		
6	85	5F		
	80	6F		
7	80	7SS		Was there any vaporization in this node?
	8S	5F		
8	83	107L		No: Go to 107L. and pick up previously calculated percent
	81	5F		Yes: Is the amount of vapor negligible?

LOCATION	ORDER		NOTES
9	86	6F	
	86	2SN	
10	8S	9SS	
	85	9SK	
11	80	9SS	
	83	15L	Yes: Set percent vaporized = 0
			No: Evaluate the specific gravity as a function of the percent vaporized.
12	OK	4F	
	8K	F	
13	87	9SS	
	04	30SK	
14	03	13L	
	82	16L	
15	8K	F	Set percent vaporized = 0
	8S	9SS	
16	85	10SK	
	8S	10SS	Calculate the density of the crude at t_m
17	85	10SS	
	87	11SK	
18	84	12SK	
	86	6SN	
19	8S	5F	
	85	7SS	
20	87	3SN	
	80	7SN	
21	87	5F	
	8S	5F	
22	OK	4F	
	8K	F	
23	87	10SS	
	04	48SN	
24	03	23L	
	84	5F	
25	8S	5F	
	85	8SN	
26	80	7SS	
	86	6SN	

LOCATION	ORDER		NOTES
27	87	5F	
	84	10SS	
28	87	9SN	
	8S	11SS	
29	85	1SN	Calculate w_L
	80	9SS	
30	87	13SK	
	87	10SS	
31	86	10SK	
	8S	12SS	
32	85	13SK	
	80	12SS	Calculate $w_V = w - w_L$
33	8S	13SS	
	85	7SS	Calculate the vapor density.
34	84	10SN	
	8S	5F	
35	87	11SN	
	86	8SS	
36	8S	8F	
	85	8SS	
37	87	13SN	
	84	12SN	
38	87	17SN	
	86	5F	
39	86	5F	
	86	5F	
40	8S	7F	
	85	8SS	
41	87	14SN	
	84	15SN	
42	84	6F	
	80	7F	
43	8S	6F	
	85	16SN	
44	86	6F	
	8S	14SS	

LOCATION	ORDER		NOTES
45	85	13SS	Calculate the liquid volume fraction
	87	11SS	
46	86	14SS	
	84	12SS	
47	8S	6F	
	85	12SS	
48	86	6F	
	8S	15SS	
49	85	18SN	Calculate H_L and H_V
	87	10SS	
50	80	19SN	
	8S	10F	
51	85	20SN	
	87	10SS	
52	80	21SN	
	8S	9F	
53	85	22SN	
	8S	8F	
54	1K	2F	
	OK	3F	
55	8K	F	
	17	1SS	
56	04	8F	
	02	55L	
57	1S	16SS	
	12	54L	
58	OK	3F	
	05	23SN	
59	87	10SS	
	8S	6F	
60	05	26SN	
	80	6F	
61	OS	8F	
	02	58L	
62	1K	2F	
	OK	3F	

LOCATION	ORDER		NOTES
63	8K	F	
	17	1SS	
64	04	8F	
	02	63L	
65	1S	18SS	
	12	62L	
66	0K	2F	
	85	30SN	
67	07	4SS	
	84	29SN	
68	07	4SS	
	8S	6F	
69	85	32SN	
	07	4SS	
70	84	31SN	
	07	4SS	
71	87	17SN	
	8S	7F	
72	05	1SS	
	84	10SN	
73	8S	8F	
	85	7F	
74	86	8F	
	86	8F	
75	86	8F	
	8S	9F	
76	85	6F	
	80	9F	
77	86	33SN	
	04	18SS	
78	0S	18SS	
	02	66L	
79	85	7SS	Calculate the thermal conductivity of the crude.
	80	37SN	
80	87	34SN	
	8S	6F	

LOCATION	ORDER		NOTES
81	8K	1F	
	80	6F	
82	87	35SN	
	86	36SN	
83	86	10SS	
	8S	20SS	
84	85	17SS	Calculate ΔH
	80	16SS	
85	87	12SS	
	8S	6F	
86	85	19SS	
	80	18SS	
87	87	13SS	
	84	6F	
88	8S	21SS	
	85	2SS	
89	80	1SS	
	8S	6F	
90	85	17SS	
	80	16SS	
91	86	6F	
	8S	22SS	
92	0K	4F	Calculate the viscosity of the vapor
	8K	F	
93	87	7SS	
	04	34SK	
94	03	93L	
	87	11SS	
95	86	9SN	
	87	38SN	
96	8S	23SS	
	85	40SN	Calculate $(R_e)_L$
97	87	12SS	
	86	41SN	
98	86	SK	
	86	23SS	

LOCATION	ORDER		NOTES
99	8S	24SS	
	8J	S6	Calculate h_i
100	87	42SN	
	8J	S7	
101	87	43SN	
	8S	5F	
102	85	22SS	
	87	23SS	
103	86	20SS	
	8J	S6	
104	86	3SN	
	8J	S7	
105	87	5F	
	8S	25SS	
106	8K	1F	
	82	[ ]F	Transfer control to main routine
107	85	36SS	
	8S	9SS	
108	8K	1F	
	82	10L	

Program to Select Correct Polynomials for

This program will select, from tape, two polynomials with PL values closest to polynomial desired. It will also interpolate between these polynomials to obtain ϵ_G

00380K			
0	42	26L	
	50	L	
1	26	S4	
	2K	2F	
2	3K	2F	
	35	28SS	Store PL CO_2, and then PL H_2O
3	8S	706F	
	8J	4L	
4	22	4L	
	F5	4L	

LOCATION	ORDER		NOTES
5	26	408F	Transfer control to 408
	OK	4F	Store polynomials in the locations designated for them.
6	05	710F	
	2S	720F	
7	05	715F	
	2S	724F	
8	2L	1F	
	03	6L	
9	85	709F	
	3S	736F	
10	2L	4F	
	32	2L	Repeat to select and evaluate q_{CO_2}. At T_B and T_S
11	85	28SS	Calculate parameter for percent correction
	84	29SS	Parameter = $PL_{CO_2} + PL_{H_2O}$
12	8S	706F	
	8J	440F	T. C. to 440
13	85	2SN	If $2 - (PL_{CO_2} + PL_{H_2O}) < 0$ let
	80	706F	$(PL_{CO_2} + PL_{H_2O}) = 1.8$
14	83	16L	
	85	6SN	
15	8S	706F	
	83	16L	
16	85	28SS	
	86	706F	
17	8S	10F	
	8J	18L	
18	22	18L	
	F5	18L	
19	26	408F	Select polynomial
	OK	5F	Evaluate
20	8K	F	
	87	10F	
21	04	715F	
	02	20L	
22	8S	5F	
	8K	100F	

LOCATION	ORDER		NOTES
23	8S	6F	
	80	5F	
24	86	6F	
	8S	738F	
25	8J	442F	Transfer to 442
	8K	1F	
26	8K	1F	
	82	[ ]F	

<u>00408K</u>

LOCATION	ORDER		NOTES
0	42	12L	
	50	L	
1	26	S4	
	8J	446F	Transfer control to 446
2	85	707F	Is $PL_m - PL > 0$?
	80	706F	
3	82	5L	Yes: Proceed to 5L
	8S	708F	No: Store difference and proceed to 421
4	8J	421F	
	8K	1F	
5	82	1L	Read in and check another polynomial
	8J	429F	Transfer to 429
6	8J	448F	Transfer to 448
	85	58SN	Was this the last polynomial in the set?
7	80	705F	
	82	9L	Yes: Use the two polynomials in memory
			No: Proceed to 436
8	8J	436F	
	8K	1F	
9	83	6L	
	85	708F	Interpolate between the PL values available.
10	84	706F	
	80	707F	
11	86	708F	
	8S	709F	
12	8K	1F	
	82	[ ]F	Transfer control to 385

LOCATION	ORDER		NOTES
0040K			
0	F5	5L	Prepare to operate on 4^{th} degree polynomials
	40	5L	
1	26	29S4	
	001F		
2	L5	5L	Reset to operate on 3^{rd} degree polynomials
	L04L		
3	40	5L	
	26	29S4	
4	00F		
	001F		
5	00F		
	004F		
6	81	40F	Read a sexadecimal character from tape
	40	707F	(representing a value of PL) and store in 707
7	22	7L	
	26	29S4	Transfer control to the order following last 8J order executed
8	81	40F	Read in and store the next PL on tape
	40	705F	
9	22	9L	
	26	29S4	Proceed to order following the last 8J order executed
00421K			
0	22L		
	L5	6L	
1	42	2L	
	41	7L	
2	81	40F	Read in and store a polynomial
	40	710F	
3	F5	2L	
	42	2L	
4	F5	7L	
	40	7L	
5	L0	445F	
	36	29S4	Proceed to last 8J order executed
6	26	2L	
	00	710F	

LOCATION	ORDER		NOTES
7	00	F	
	00	F	
8	L5	9L	Read in and store polynomial associated with PL last checked
	42	10L	
9	41	7L	
	LO	715F	
10	81	40F	
	40	715F	
11	F5	10L	
	42	10L	
12	F5	7L	
	40	7L	
13	LO	445F	
	36	29S4	Proceed to order following 8J order last executed
14	26	10L	
	26	10L	
15	41	7L	
	81	40F	Read in and dump a polynomial
16	F5	7L	
	40	7L	
17	LO	445F	
	36	29S4	Transfer to order following the last 8J order executed
18	22	15L	
	22	15L	

Subroutine to Evaluate Polynomials for and to interpolate between them

00665K		
0	42	13L
	50	L
1	26	S4
	OK	2F
2	1K	F
	2K	2F
3	3K	2F
	4K	4F
4	8K	F
	07	30SS

LOCATION	ORDER		NOTES
5	14	720F	
	1L	1F	
6	42	4L	
	3S	5F	
7	32	3L	
	80	5F	
8	26	736F	
	84	5F	
9	2S	8F	
	23	3L	
10	84	8F	
	0S	10F	
11	03	2L	
	80	10F	
12	87	738F	
	8S	10F	
13	8K	1F	
	82	[ ]F	

00679K			Subroutine to Evaluate ΔP
0	46	22L	
	50	L	
1	26	S4	
	85	24SS	
2	8J	S6	
	87	1014F	
3	8J	S7	
	8S	5F	
4	85	15SS	
	80	1015F	
5	83	19F	
	85	15SS	
6	87	5F	
	84	1016F	
7	87	37SN	
	87	13SK	

LOCATION	ORDER		NOTES
8	87	13SK	
	OK	5F	
9	86	SK	
	03	9L	
10	86	1017F	
	8S	5F	
11	85	11SS	
	80	14SS	
12	87	15SS	
	84	14SS	
13	8S	6F	
	85	1011F	
14	87	SK	
	84	38SS	
15	87	39SS	
	87	5F	
16	86	6F	
	84	5SS	Does P_2 assumed = P_2 calculated?
17	80	4SS	
	8S	10F	
18	8N	39SN	
	82	22L	No: Modify P_2 and repeat calculations Yes: Replace P_1 by P_2
19	85	5SS	
	8S	4SS	
20	85	2SS	Replace t_1 by t_2
	8S	1SS	
21	84	40SN	
	8S	2SS	
22	82	[ ]F	Transfer control to program
	85	5SS	
23	80	10F	
	8S	5SS	
24	84	4SS	
	86	2SN	
25	8S	8SS	
	82	1S8	

LOCATION	ORDER		NOTES
0019K			
0	85	1015F	
	83	685F	

Table 5. Polynomials for the Evaluation of ψ

Per cent Correction:

$2\% = a + bR + cR^2 + dR^3 + eR^4$; $R = CO_2/(CO_2 + H_2O)$

Parameter = PL	a	b	c	d	e
0.01	0.3857	4.3900	20.1857	-33.8957	9.7907
0.25	0.3558	21.9324	-16.2491	5.5128	-8.6094
0.5	4.5758	7.3524	18.9710	-19.1826	-6.4339
0.75	1.0423	51.7054	-98.6457	106.2914	-53.2739
1.0	2.5018	49.2875	-92.4543	102.1649	-52.9669
1.5	3.2912	55.1680	-109.9346	120.0117	-58.3493
2.0	4.2321	51.9834	-101.8847	111.4446	-54.6296

Radiation due to Carbon Dioxide:

$2E = a - bt + ct^2 - dt^3$

Parameter = PL_{CO_2}	a	b	c x 10^3	d x 10^7
0.001	51.7010	0.1513	0.2078	-0.2919
0.002	54.0670	0.1802	0.3150	0.3364
0.003	211.8020	0.5723	0.6414	0.7937
0.004	173.6900	0.6191	0.8155	1.0992
0.005	30.0980	0.3734	0.7349	0.8113
0.006	35.6890	1.0696	1.2016	1.4618
0.008	252.8100	0.9766	1.2920	1.4689
0.01	651.9700	1.9534	2.0423	2.6610
0.015	754.5600	2.2446	2.3347	2.7522
0.02	1278.9000	3.5058	3.2896	4.2098
0.03	1503.5400	4.1913	3.9372	5.0236
0.04	1103.5000	3.3869	3.5666	3.7936
0.06	1346.5700	4.1166	4.2543	4.3270
0.08	668.4400	2.5749	3.2179	1.4470
0.10	1817.8000	5.3986	5.3289	5.2498
0.15	3927.0900	10.0878	8.3592	9.5874
0.20	3990.4400	10.2342	8.4211	8.1694
0.30	5781.0000	14.3987	1.1211	11.8823
0.40	4823.2900	12.1531	9.7198	7.3506
0.6	7588.0000	18.2576	13.4843	11.9161
1.00	6246.9900	16.0636	12.7290	8.7444
2.00	8057.2900	19.8252	15.1587	9.2603
4.00	6090.0000	16.0543	13.1966	1.7166

Radiation Due to Water Vapor:

$2E = a - bt + ct^2 - dt^3$

Parameter $= PL_{H_2O}$	a	b	$c \times 10^3$	$d \times 10^7$
0.01	55.5228	0.1204	0.2066	0.0937
0.015	-110.9520	-0.2520	0.0261	-0.4142
0.02	25.6710	0.0940	0.3523	0.1163
0.025	321.7550	0.7316	0.7960	0.7295
0.03	265.0900	0.6268	0.8141	0.6336
0.04	229.8430	0.5751	0.8985	0.5156
0.05	430.1100	1.0786	1.3918	1.1563
0.06	25.3160	0.2532	1.0233	0.1046
0.08	288.6660	0.7717	1.3913	0.3541
0.10	139.8600	0.3633	1.2072	-1.1428
0.15	537.3800	1.3519	2.2078	0.6104
0.20	1908.1100	4.3802	4.4417	1.7536
0.25	8.4300	0.3724	2.2078	-3.4239
0.30	1196.2760	3.1741	4.4304	-0.5380
0.40	453.3800	1.1607	3.0303	-0.6590
0.50	-90.2000	-0.0144	2.5594	-9.9998
0.60	631.0400	1.9334	4.1433	-9.2052
0.80	2025.3000	4.6138	5.8862	-11.4342
1.00	622.9000	1.6958	4.3397	-18.4798
1.25	-2283.7000	5.2497	-0.6273	-34.1129
1.50	1408.0000	0.0252	3.8746	-31.9589
2.00	5548.0000	10.9464	8.7011	-32.9599
3.00	1362.3000	-0.3637	-0.5046	-64.4918

Black Body Radiation:

$2E_B = -5926.00 + 27.6324t - 31.7241t^2 + 255.0130t^3$

Overall Exchange Factor:

$2\psi = a + bR - cR^2 + dR^3$; $R = A_R/\alpha A_{cP}$

Parameter =	a	b	$c \times 10^2$	$d \times 10^4$
0.2	0.3875	0.3009	0.3655	0.1787
0.22	0.3934	0.3569	0.5188	0.2995
0.24	0.4655	0.3292	0.4177	0.2023
0.26	0.4981	0.3467	0.4726	0.2480
0.28	0.5460	0.3480	0.4765	0.2437
0.30	0.5827	0.3634	0.5380	0.2990
0.32	0.6170	0.3740	0.5750	0.32811
0.34	0.6583	0.3651	0.5726	0.3038
0.36	0.6830	0.3818	0.6028	0.3486

Table 5 (cont.)

Parameter =	a	b	$c \times 10^2$	$d \times 10^4$
0.38	0.7336	0.3505	0.4950	0.2447
0.40	0.7671	0.3791	0.6236	0.3728
0.45	0.8699	0.3668	0.6148	0.3657
0.50	0.9567	0.3472	0.5738	0.3290
0.55	1.0559	0.3379	0.6076	0.3765
0.60	1.1404	0.3245	0.6003	0.3765
0.65	1.2232	0.3068	0.5993	0.3929
0.70	1.2863	0.2976	0.6122	0.4172

Table 6. Molar Heat Capacities of Flue Gas Components

$C_p = a + bT + cT^2$; T = OK

Compound	a	$b \times 10^3$	$c \times 10^6$
Oxygen	6.0954	3.2533	-1.0171
Carbon dioxide	6.3930	10.100	-3.405
Water vapor	7.219	2.374	0.267
Nitrogen	6.4492	1.4125	-0.0807