

**FIRED HEATERS
OR
FURNACES**

INDEX

	<u>Pages</u>
I. REASON FOR DISCUSSION	
A. General	1
B. Failure 1	1 & 2
C. Failure 2	3
D. Failure 3	3
E. Failure 4	3
F. Failure 5	3
G. Failure 6	3
H. Failure 7	3 & 4
II. HEATER NOMENCLATURE	4
III. HEATER CLASSIFICATION	
A. By Service	4 & 5
B. By Type	5
IV. FUNCTION OF HEATER	
A. General	5 & 6
B. Historical Development	6 - 20
V. CONSIDERATIONS IN DESIGN	
A. General	20 & 21
B. Modes of Heat Transfer	21 & 22
C. Radiant Section - Lobo Evans Method	22 - 43
D. Convection Section	44 - 73
E. Stack Design	74 - 76
F. Furnace Rating Procedure	77 - 84
VI. SAMPLE CALCULATIONS	85 - 99

I. REASON FOR DISCUSSION

A. Why do we want to discuss or talk about this equipment?

1. It is one of the most neglected pieces of equipment in a unit.
2. It is usually a piece of equipment purchased as the least cost item which probably means the most tightly designed piece of equipment.
3. It is the only place in the unit where there are exposed fires used to process flammable fluids.
4. It usually is pushed to over design during operation without operator knowledge that this is occurring.
5. Operator judgements may or may not be bad but these happenings are usually the result of errors.
6. When the above happens the results can be disastrous and typical examples of these occurrences are shown on Figure 1, 2, 3, 4, 5, 6, etc.

B. Figure 1a, 1b, 1c

This heater is a two service heater i.e. Charge service and a Reboiler service. The heater also has an air preheat system.

(Description of this heater operation not written.)

The fire started when tube no. 14 (hip section) of H-E2 (Reboiler) ruptured, Fig. 1a, during a start-up. The liquid drained on to the floor from the convection zone and into the common air ducting which caused the fire to spread underneath and over the whole heater, stacks and forced-air system. The heater was damaged essentially from an explosion after the tube failure. The fire damages were minimal.

The fractionator reboiler heater, H-H2, had never been used alone.

It is apparent that soon after the unit was commissioned, the operating group may have assumed that the H-H2 was too large, due to an over-sight of the duty of the convection tubes. Consequently, in future start-up's, the reboiler radiant section may not have been fired together with the reactor charge section of the heater. Thus the possible contaminated condensate in the cooler H-H2 parallel radiant coil tubes vaporized, in the zones shown on sketch no. 1a, from the hot flue gases of the reactor charge coils section.

Tube wall thinning was definitely on the inside. Note sketch no. 1a "Section BB". Externally, the tubes indicated that they had never been exposed to excessive temperatures.

It was obvious that the corroded/eroded surface was caused by vapors contaminated with traces of chlorides which was being influenced by the flue gas temperatures from the reactor charge section of the heater. As mentioned above, the U-bends were not attacked.

Tube material was inferior in quality making it most susceptible to even weak corrosive environments of this nature.

The unit is about 7 years old and at each shut-down it is completely steamed-out. By means of a flow sheet of the unit it was shown how small traces of chloride could reach the H-H2 coils from the stripper column, if the unit was not properly water flushed and drained before each start-up. Figures 1b and 1c are the results of this occurrence. Delay in returning the unit to operation will be about one year.

C. Figure 2

A typical aftermath of a heater firebox explosion. Leaking fuel gas valves allowed an explosive build-up of gas during the purging period. The operator's torch caused ignition while still three feet away from the firebox.

D. Figure 3

A typical common breeching situation. A fire which originated in the center heater damaged both of the others and severely damaged the common stack.

E. Figure 4

One result of congestion - Note the final resting place of the toppled heater stack impaled on the small tower.

F. Figure 5

The result of a tube failure in the convection section which is inaccessible for inspection.

G. Figure 6a, 6b, 6c, 6d, 6e, 6f, 6g are the results of a tube rupture at high pressure with back flow resulting in erosion of the exterior elbow. This caused most of the external damage. The fire in the heater was contained in the heater.

H. Figure 7a, 7b, 7c, 7d, 7e

Again results of a tube rupture and the results turned out to be spectacular. The major fire was contained in the firebox and only the tube bank needed replacement.

These are just a few of the occurrences that happen with this equipment and it shows that more attention to the equipment and better equipment should be considered in your units.

There are many more happenings that could be discussed but time is limited and we must go on.

II. HEATER NOMENCLATURE

Figures 8 through 13 describe roughly the nomenclature used by furnace or heater designers. These are just a couple of samples but will give you an idea as to what the terms mean and location of the items so named.

III. HEATER CLASSIFICATION

A. By Service

1. No decomposition

a. Atmospheric distillation

b. Vacuum distillation

c. Catalytic processes

(1) Hydrodesulfurization

(2) Catalytic cracking

(3) Catalytic reforming

(4) Other (Hydrocracking, Steam-methane, Styrene, etc.)

d. Reboilers

e. Miscellaneous

2. Partial conversion

a. Thermal cracking

(1) Single coil

(2) Selective

b. Continuous coking

3. Total conversion

a. Thermal reforming

b. Viscosity breaking

c. Pyrolytic cracking

(1) Light hydrocarbons

4. Combined duties in a single heater

B. Classification as to Type

1. By predominate heat transfer means

a. Convection

b. Radiant

c. Radiant - Convection

2. By general arrangement

a. Box

b. Cellular

c. Circular

FUNCTION OF HEATER

A. General

Most refining processes require a source of heat to raise the temperature of the streams to be processed so that the proper processing function can be accomplished. With the more thorough and sophisticated processing more attention has developed in design of the equipment.

Petroleum heaters can be divided into three main groups, according to the amount of decomposition obtained

1. Heaters used for heating only with little or no decomposition.
2. Heaters where in addition to the heating, substantially all of the decomposition desired for the refining process is obtained in the heater.
3. Heaters where only partial decomposition is obtained in the heater, with the remainder of the reaction being carried out in the reaction chambers or soaking drums, usually not heated externally.

Heaters of the first group are designed to obtain minimum time-temperature effect with the maximum temperature employed and are used for non-destructive distillation where no chemical change of the overhead and residual products is desired, such as topping, atmospheric or vacuum distillation, rerunning, etc.

Heaters of the second group are designed to give maximum time-temperature effect at the highest operating temperature employed, and are used with cracking units for the lighter overhead stocks where no reaction chamber is employed. These heaters are usually so designed that the heating surface nearest the heater outlet is subjected to lower rates of heat input than the rest of the tube bank to provide a reaction or "soaking" section in the heater, where very little or no increase in temperature is obtained.

Heaters of the third group are the most difficult to design, particularly for residual cracking stocks which are highly sensitive to thermal treatment. These heaters must be designed for a time-temperature effect that will permit highest coil outlet temperatures, and consequently highest reaction chamber temperatures to assure effective operation of the chamber, without excessive decomposition and the resultant coke deposition in the coil.

3. Historical Development

The earliest heaters for petroleum refining consisted of cylindrical shells set over a firebox, with the lower portion of the cylinder absorbing heat from the flame, the refractories and the products of combustion that swept the lower rear portion of the shell on their way to the stack.

Because of a large volume of oil per unit of heat-absorbing surface and low velocities with which the surface was washed by the oil being heated, shell heaters were limited to low rates of heat input. The permissible maximum was determined by the rate of free convection or convection induced by steam jets within the body of the oil, which even with boiling fluids was too low to permit rates much in excess of 3,000 B.t.u. per square foot per hour without over-heating the oil in contact with the heated portion of the shell.

Coke deposition and burning out of still bottoms almost invariably followed attempts to force shell heaters much above these rates. The addition of internal flues to shell heaters to simulate a fire tube boiler materially improved their operation and increased their capacity. However, shell heaters were primarily batch heaters, and continuous operation, which gradually displaced batch operation in the larger refineries, could be accomplished with them only by operating a battery of shell heaters in series, each shell operating at a slightly higher temperature than its predecessor in the series. With large scale operation in some fields, batteries of shell heaters became too cumbersome.

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 of shell heaters by tubular heaters for large-scale refining operations.

The introduction of cracking, which called for heaters that could withstand higher pressures and higher metal temperatures than could be withstood by an externally heated shell, contributed greatly

to the development of tubular heaters and definitely demonstrated their superiority over the shell type.

The earlier tubular heaters varied little in their setting from shell heaters, the main difference being that the shell was displaced by a bank of tubes placed over the firebox. However, since a definite velocity was imparted to the oil by pumping in continuously through the tube bank and the ratio of heating surface to volume was greatly increased, average rates of 5,000 - 6,000 B.t.u. per hour per square foot were obtained without particular difficulties in the early tubular heaters in spite of the normally present zones of local over-heating, or "hot spots" in some sections of the tubes directly over the fires with actual rates of 15,000 - 20,000 B.t.u. per square foot.

As the average rates of heat input and the oil temperatures were increased, the hot spots became more pronounced and troubles were encountered with failure of tubes in these hot spots, particularly in heaters, where counter-current flow of oil and gases was intended. The troubles were first attributed to radiation from the flames in the firebox, and a number of tubular heaters were built with the tube bank separated from the firebox by solid or perforated walls to protect the tubes from this radiation. Because of more uniform heat distribution, greater average rates of heat input were obtained in these heaters than in heaters where the tube bank was directly over the fires in the combustion chamber, but it was soon found that even with the bank shielded from the visible flames, the tubes first coming in contact with the invisible products of combustion became easily over-heated unless the gas temperature was reduced below a certain minimum for the operation carried out.

Reduction of the temperature of the gases entering the tube bank was the next step in the development of tubular heaters. This was done in two different ways. In the so-called "convection" heaters, where only "fluid" heat from gases flowing over the tubes was used, the products of combustion were diluted first with excess air and later with returned or recirculated flue gases, to obtain the desired temperature of the gases entering the tube bank or the "heating chamber".

In other heaters, "radiant" tubes were installed in the combustion chamber to cool the products of combustion by absorbing radiant heat from the flame, the gases and the refractories of the combustion chamber before the gases entered the "main" or "convection" bank of the heater, which was intended to supply the major portion of the total heat, and which was usually separated by a bridge wall from the combustion chamber. Particular care was taken in these heaters to place the "radiant" tubes out of the path of the products of combustion. To reduce the rates of heat input which would be absorbed by the relatively small number of tubes that were subjected to the full radiation in the combustion chamber, the tubes were shielded from this radiation by surrounding either the tubes or the flames with refractory enclosures, which cut these rates approximately in two.

High maintenance and operating costs of flue gas recirculating equipment and frequent failures of the enclosures around the tubes or the flame, together with comparatively satisfactory operation of some open-flame "radiant" heaters without flue gas recirculation and without shielding either the tubes or the flame, led to the gradual

abandonment of the flue gas recirculating and the shielded "radiant tube" heaters and their substitution with the open-flame type.

Improvement in heat generation and distribution and better understanding of the laws of radiant and convection heat transfer, greatly contributed to the development of open-flame heaters and their almost universal adoption for oil refining. The proportion of radiant heat input has been gradually increased from 25 per cent of the total in the early heaters to as high as 80 per cent of the total in the latest designs.

Fig. 14a A & B shows a flue gas recirculating heater of an early type widely used in petroleum refining. In this heater the fresh products of combustion generated in the combustion chamber are mixed with flue gases withdrawn from the flue to the stack by means of a flue gas recirculating fan, and distributed along the length of the heater through openings in a distributing duct.

Mechanical problems with this style of heater were numerous. There are other styles of flue gas recirculation systems that were used but time does not permit discussion.

Fig. 14a B shows an early shielded radiant heat oil heater in which a roof bank was installed to absorb sufficient radiant heat from the products of combustion before they enter the "main" or the convection bank of the heater, to permit their safe use in this bank without flue gas recirculation or high amounts of excess air. To protect the relatively small number of the tubes in the roof bank from the full radiation in the combustion chamber, the portion of the tubes over the combustion chamber where radiation was the most intense was covered with ceramic enclosures. Cast-iron enclosures were used on the portion over the main bank where radiation was

comparatively low. Comparatively high amounts of excess air (100 - 150 percent) had to be used in this heater to reduce the temperature of the gases entering the main bank, when the heater was operated at high enough capacities effectively to utilize all of the surfaces in the main bank.

Fig. 14a, C shows another type of an early shielded radiant heat oil heater where shielding enclosures or "muffles" were placed around the flames and the products of combustion to protect from direct radiation the comparatively small amount of radiant heat-absorbing surface used. Because of the high temperature to which the enclosures were subjected and to obtain comparatively high rates of heat transfer through these enclosures to the radiant tubes, carborundum was used for the enclosures. The heater was intended for operation with low excess air to obtain high flame-burst temperatures and the highest radiation through the carborundum.

Fig. 14b, D shows an early open flame radiant heat oil heater with a much larger combustion chamber than that of the heater shown in Fig. 14a, A, and with approximately one-half of the total tube bank located along the roof of the combustion chamber to permit the use of bare radiant heating tubes and open flames.

Fig. 14b, E shows an open flame heater with radiant heat-absorbing tubes along the roof and at least two walls of the combustion chamber.

Fig. 14b, F shows a partially shielded flame heater where the muffles shield only a small portion of the flames and act more for promoting and completing combustion than as shields between the flames and the tubes. Approximately 75 percent of the total surface is exposed to radiation and the floor and the back walls of the

heater, in addition to the roof, are covered with tubes.

Fig. 14b, G shows an early down-draft heater consisting of a vertical cylindrical section with the tubes placed along the whole circumference of this section, a secondary horizontal cylindrical section with tubes located along the lower half of this section, and a convection section. No air preheater was generally used. The later design down-draft heaters consist only of a vertical cylindrical section and an air preheater.

Fig. 14c, H shows an updraft heater of the same general type as Fig. 14b, G. The convection section when used is very small, and an alloy air preheater is usually employed to reduce the temperature of the gases going to the stack.

The heaters shown on Figs. 14a, B to 14c, H inclusive represent the complete range of proportions of radiant heat-absorbing surface to total surface used. The heaters shown on Fig. 14a, B & C have too little radiant heat-absorbing surface, so that shielding of the surface was required to protect these tubes from excessive rates, overheating of the tubes, coke deposition in the tubes and their eventual burning out. The heater shown on 14c, H has too much radiant heat-absorbing surface in proportion to the furnace envelope, with the result that the effectiveness of the surface is comparatively low, unless the gases are made to wash the upper portion of the tubes to supply an appreciable amount of purely convection heat to these tubes.

Fig. 14c, I shows a heater with approximately 50 per cent of the total heat-absorbing surface located along the roof and the floor of the combustion chamber. The tube banks in the combustion chamber form approximately two-fifths of the area of the furnace envelope.

With single rows of tubes used in the combustion chamber, the optimum effectiveness of the circumferential area of tubes irradiated from one side is obtained. As a general rule, the floor tubes of the heater are the tubes of the highest rates of heat input.

Heaters shown previous to Fig. 14c, I, had fixed heat distributions and were rather inflexible as to a change in the type of service it was to perform i.e. design strictly for a certain process operation.

The heater on Fig. 14c, I can be operated to obtain a limited control changing the character of the flame and therefore the relative rates of heat input into the convection section, the roof tubes and the floor tubes of the heater. The burners fire through three compartment firing tunnels, of which the center tunnel F is the firing tunnel proper and the upper and lower tunnels A and B are for admission of secondary air to the fuel after its ignition in tunnel F. With all the air passing through compartment F as primary air, and short non-luminous flames, the rates of heat input into the floor tubes and the roof tubes near the firing wall are proportionally lower, and the rates of heat input into the convection section and the roof tubes over the bridge wall and near the back wall of the heater are proportionally higher than if only a portion of the air is sent through compartment F and the rest of the air used as secondary air through compartments A and B to produce luminous flames.

Restricting the flow of air through compartment F and admitting a large portion of the air as secondary through compartment A, produces semi-luminous flames with a hotter face toward the floor

tubes. Admitting the secondary air through compartment B gives a semi-luminous flame with a hotter face toward the roof tubes.

Fig. 14c, J shows an updraft rectangular heater in which the radiant heat absorbing surface is located along the roof and the side wall of the heater, with the tubes in the roof subject to both radiant heat and fluid heat from the gases passing to the convection section. Special tile is placed over the top row of the roof tubes to act as distributing baffles and to direct the gases at high velocities along the shielded faces of these tubes, so as to impart a high convection component to the shielded faces and partially equalize the heat pickup along the circumference of the top row of the roof tubes. The parallel flow arrangement is used for large capacities. Series flow arrangement is shown by the dashed lines.

By changing the type and rate of firing of the two sets of burners in the combustion chamber a wider variations can be obtained in this heater than in the heater shown on Fig. 14c, I. If a separating wall is installed in this heater, as shown in dotted lines, to divide the combustion chamber into practically two separate compartments a wide range of heating curves may be obtained in this heater. Because of the relative flexibility of firing, this type of heater once installed may be used without changes for a wider range of charging stocks that the heaters shown on Figs. 14a, B - 14c, I inclusive, since the heating curve in this heater may be changed to obtain the desired time-temperature effect for the stock processed.

Fig. 14c, K shows a 2-compartment heater often referred to as the double end-fired heater. In this heater the two combustion chambers

are completely separated and the floor and roof tubes of one chamber function wholly independent of those in the other chamber, providing more definite control of the heating curve than that which could be obtained in the heater shown on Fig. 14a, B - 14c, J.

This heater is used with either parallel flow, as shown in solid lines, for processing two streams of oil, or with a single flow in series through both compartments, as shown in dotted lines.

In the early 1930's, UOP developed the Equiflux Heater. This development basically was due to overcome all the deficiencies of the earlier box type heaters. It was and still is the best heater for the services of that day, namely, thermal cracking and was developed for that service. Controlability of the heating curve was the main reason for its use; the one drawback was the cost of the heater as it did indeed cost more than the box type. This heater is shown as Fig. 15. The first designs were the down-fired as shown in 15A & B. Later in the 1930's, the updraft type, Fig. 15C was designed.

The cellular structure of the heaters permits the use of any number of streams with each stream subjected to the optimum thermal treatment. The cellular structure also permits maintaining the optimum dimensional relationship of the heater structure for the best gas hydraulics within the structure irrespective of the total heat requirements of the process.

The radiant tubes were spaced farther apart than in other types of heaters. The radiant tube spacing for other types of heaters was usually twice the tube outside diameter. The Equiflux Heater spaced radiant tubes on 2.6 diameters. The radiant tubes were also placed

in the middle of the firebox. This combination of wider tube spacing and a location in the center of the firing zone allowed the tubes to receive almost equal radiation around the whole circumference of the tube and hence the name, "Equiflux". There is only about 5% variation in heat input rates between the radiant tubes at the top of the furnace and those at the bottom the box type heater had about 20% variation in radiant rates.

When thermal cracking black oil, the Equiflux radiant tubes were usually designed for 15,000 Btu/Hour/external exposed sq. ft. of surface.

The Equiflux type of heater cost more than other types of heaters of similar size. The refractory (brick work) was not protected by tubes like the walls in the box type heater. Since the radiant tubes were on wider than normal spacing, less radiant surface could be installed in a given combustion space in spite of the fact that the allowable radiant heat input rates were higher. The tube supports had to be high nickel alloy and the walls had to be super-refractory firebrick.

In about 1936, Figure 16, UOP built the center wall updraft heater and shortly thereafter, the first updraft without a center wall was installed. This has many features which have been changed from the original centerwall.

The average radiant rates in the center wall updraft heater were usually 11,000 - 11,500 Btu/Hour/external exposed sq. ft. and 10,000 - 10,500 in the heater without the center wall. These floor fired types were good heaters when gas was fired but not so good for oil firing. This is one of the reasons why the updraft heater without the center wall was designed, Fig. 17. It could be fired from the

ends horizontally or vertically from floor. UOP then built a side fired center wall updraft heater and made it wider than previously. The two heaters shown above were about 10'-6" wide. The side fired heater was about 19 ft. wide and was built in about 1955.

In about 1945, Petrochem built a circular heater with vertical tubes. There was a previous heater designed by Sinclair Ref. Co. called the "De Florez Heater" but Sinclair was about the only refiner who used a circular heater in thermal cracking service. The Petrochem heater was rarely used for thermal cracking service. It was primarily used for Crude, Vacuum and Reboiler heaters. It was cheaper than most heaters, took up less space, but was less efficient thermally. The early models had fins on the upper section of the tubes and used a cone to increase the flue gas velocity.

The average heat rates in crude service were about 10,000 Btu/Hour/Sq.Ft. The center wall updraft in crude service used 11,500 - 12,000 Btu/Hour/Sq.Ft.

In about 1960, Phillips Pet. Co. built the "Selas Heater". It looks similar in shape and tube bank arrangement to the UOP updraft Equiflux. It had gas burners arranged on each side wall to control the shape of the heating curve.

The above and the better heaters are shown as Figure 18. They may be called the following although the names are not standardized:

- A. End fired down-convection
- B. Large double cell box-type
- C. Separate-convection (Lummus)
- D. A-frame (Kellogg)
- E. Double upfired centerwall (UOP)

- F. Floor fired horizontal (BORN)
- G. Equiflux downfired (UOP)
- H. Circular (De Florez)
- J. Isoflow (Petrochem)
- K. Isoflow (Intregal tube)(Petrochem)
- L. Radiant wall (Selas)
- M. Inverted wicket (UOP)

There are other shapes but this is only to give a rough idea of availability.

Although all of the stills of Fig. 18 are useful for many services, there are certain features in which each excels or may be deficient.

1. Flame Impingement. Too large a flame can cause impingement or overheating of certain tubes in nearly all of the stills, but tubes that are vulnerable are those just above the bridgewall in (A) down-convection, and the lowest tubes of (E) double-upfired centerwall or (F) straight-up. The difficulty disappears in stills (F) and (E) when the capacity is large, i.e., the stills can be wider. Note the black tubes in Fig. 18.
2. Hot Tubes. The rate of heat absorption tends to be high at the entrance to the convection section because heat is delivered by both radiation and convection (black tubes in type (A), (B), (D), (E), and (F)). The feed, because it is cooler, is sometimes introduced through the tubes (a screen bank), and in some instances steam is superheated or water is heated in the screen bank. Likewise, the tubes situated closest to the flame receive the largest amount of heat (Black tubes in (E), (F), (E), (K), and (M)).

3. Oil Firing. Oil fuel tends to produce larger flames especially because larger burners are used. Stills (A), (B), (C), and (D) are best suited for oil firing although the larger sizes of the other stills encounter little difficulty.
4. Heat Distribution. One phase of poor heat distribution has been mentioned in item (2) above, but in addition the rate of heating varies widely in different parts of the boxes or cavities of such stills as (A) and (B), although most of the other vertical stills can attain a fairly uniform distribution of heat.
5. Two-coil Heating. Still (A) is not suited for two-coil flow although by experiment it is possible to use it for two streams. Symmetrical two-coil arrangements can be devised for most of the other stills, but if a different rate of heat input is needed in each coil, the stills with a center bridgewall are most effective.
6. Control of Rates. Stills (G) and (L) are eminently suitable for process control of the rate of heat input and for operation at exceedingly high (1000 to 1500°F) temperatures. Very low rates of heat absorption can be attained in the circular types (H), (J), and (K); and in other types if enough radiant surface is provided.
7. Capacity. Stills (A), (F), (G), (K), and (L) can be built for the lowest capacities, and types (B), (C), (D), and (E) are best suited for large capacities.
8. Stacks. Stills (A), (B), (C), and (G) require tall stacks, but scarcely any stack (except to release the gas at a high level) is needed with the other stills because the hot furnace itself produces a draft.
9. Type M is well suited for extremely low pressure drops.

Air preheat is indicated in Fig. 18-H (DeFlores), and it can be employed in any still that is mechanically tight from gas leakage. The recirculation of flue gas through the combustion chamber by means of a fan is not indicated in any of the stills. The flue gas is reheated by the flame, causing a lower flame temperature and thus a milder radiation rate. At the same time the convection-section duty is increased, and a larger convection section must be provided. Thus, flue-gas recirculation is particularly suited to the older convection-type stills. When it is necessary to alter or regulate the rate of radiant absorption, no method other than flue gas recirculation is available except such expensive methods as the use of large amounts of excess air, a reduction in capacity, or possibly to obtain a fuel which radiates less. Flue-gas-recirculation stills were widely used during the development of thermal cracking processes; but, now the engineers know how to design stills for specific radiant-absorption rates recirculation stills are no longer installed.

CONSIDERATIONS IN HEATER DESIGN

General

The fired heaters are basically large heat exchangers. The heat input is provided by burning fuel, usually gas or oil, in the combustion chambers. Heat is transferred from the hot gas to the fluid contained in tubes. As with any heat exchanger, the rating involves a heat balance between the heat releasing and heat absorbing streams and a rate relationship. But in contrast to the usual heat exchanger, much of the heat is transferred by radiation instead of convection.

The radiant section provides space for the fuel to be completely mixed with air and burned. It contains the radiant heat absorbing tubes which remove a large percentage of the heat from the flue gas before it passes out to the convection section.

The convection section recovers additional heat from the flue gas, at a lower temperature level than can be economically obtained in the radiant section.

We will use the problem developed in the process design of the Crude Unit for sample calculations.

Modes of Heat Transfer

1. Conduction. Heat may be transferred by conduction from one molecule to the next within a substance. If one end of a steel rod is inserted into a furnace, the other end of the rod will become hot after a time due to conduction of heat through the metal rod.
2. Convection. Heat is transferred between solids and fluids by a process known as convection. This is really a combination of conduction and fluid motion. For example, heat moves from the outside surface of a hot oven wall to the adjacent air molecules by conduction, and then the air moves away carrying the heat with it. The motion of the fluid (air in this case) may be due to the natural buoyancy of the heated molecules (hot air rises) in which case the process is termed free or natural convection, or the motion of the fluid may be mechanically produced (as by a fan) in which case it is termed forced convection. Two special forms of convection which behave quite differently from all others are boiling and condensation.

3. Radiation is entirely different from the above-mentioned modes of heat transfer. Light is a form of heat radiation, but all heat radiation is not necessarily visible (light). For example, a person can feel the heat radiation from a furnace wall or a hot piece of metal even though neither may be red hot (that is, neither is radiating light). Radiant heat travels in straight lines and may pass through a vacuum, air, some gases, some liquids, and a few solids such as glass and quartz.

C. Radiant Section - Lobo Evans Method

Extensive investigations have been made of the radiant heat transfer between solid surfaces in various arrangements, and between hot gases and solids. Lobo and Evans applied the basic radiation concepts to furnace design, and developed a rating method that was generally applicable without excessive complication. The method described here follows their basic pattern. It has been further simplified, however, by elimination of some minor variables and inclusion of additional general correlations.

The basis for radiant heat transfer is the Stefan-Boltzman equation. A black body at absolute temperature T radiates energy at a rate W_B , given by the relation:

$$W_B = \sigma T^4 \quad (1)$$

W_B = Radiant Energy Rate

σ = Stefan-Boltzman Constant =
 0.173×10^{-8} Btu/Hr, Sq.Ft., $^{\circ}R^4$

T = Absolute Temp., $^{\circ}R$

The Stefan-Boltzman constant σ has a value of 0.173×10^{-8} Btu/sq. ft., hr, $^{\circ}\text{R}^4$.

For radiant heat transfer between two real surfaces at temperatures T_a and T_b , the relation becomes equation (2):

$$q_r = \sigma AF(T_a^4 - T_b^4) \quad (2)$$

q_r = Radiant Heat Transfer, Btu/Hr

A = Area, Sq. Ft. (one of the surfaces)

F = Radiant Exchange Factor

T_a = Temp., $^{\circ}\text{R}$ of One Surface

T_b = Temp., $^{\circ}\text{R}$ of Other Surface

Here A is the area of one of the surfaces, and F is an exchange factor which depends on the relative area and arrangement of the various surfaces, and on the emissivity and absorptivity of each. Either the heat-losing or heat-absorbing surface may be used as a basis for determining q_r . However, the value of the exchange factor depends on which surface is used.

1. Equivalent Cold Plane Surface

In a furnace, the heat-absorbing surface is usually well defined. Furthermore, the rate of heat transfer per unit of absorbing area is important to the design. Therefore, it has become accepted practice to use the heat-absorbing, or "cold", surface as a basis for computing the radiant heat transfer.

The usual heat-absorbing surface consists of a number of parallel cylindrical tubes in front of a refractory wall. Part of the radiation from the hot gas strikes the tubes directly and is absorbed. The remainder passes through to the refractory, and is reradiated back into the furnace. Again, part of the

reradiated energy is absorbed by the tubes, and the remainder passes through.

This complicated situation is handled in the calculations by replacing the tube bank with an equivalent plane surface, A_{cp} . It equals the number of tubes, times their exposed length, times the center-to-center spacing, Equation (3). The tube bank, however, will not absorb all the energy radiated to the cold plane area. So the total cold plane area must be corrected by an absorption efficiency factor, α . Hottel has published values for the factor α as a function of tube arrangement and spacing. His curves for one or two rows of tubes in front of a refractory wall are reproduced in Figure 19.

$$A_{cp} = N \times L \times S \quad (3)$$

A_{cp} = Cold Plane Surface

N = Number of tubes

L = Length of tubes, effective, ft.

S = Center-to-center spacing, ft.

The product of the actual cold-plane area and the absorption efficiency factor, αA_{cp} , is called the equivalent cold-plane area. It is the area of an ideal black plane having the same absorbing capacity as the actual tube bank.

$$A_{cp} = N \times L \times S \times \alpha \quad (4)$$

A_{cp} = Equivalent Cold Plane Surface

α = Hottel Factor or Efficiency Factor

The shield tubes require special handling. These tubes are not backed by a reradiating refractory wall. Instead, any energy that is not directly absorbed by the shield tubes passes

on to the tubes in the convection section, and is absorbed there. Therefore, from the standpoint of the radiating gas, the factor α for the shield tubes is unity. So in calculating the total equivalent cold-plane area in a furnace, the applicable value of α is applied only to the refractory-backed tubes. The cold-plane area of the shield tubes is taken at full value. It might be argued that the shield tubes do not pick up all this heat, so the computed heat absorption in the radiant section will be too high. However, actual experience shows that because of the high gas velocities past the shield tubes, convective transfer to them is higher than to the normal radiant tubes, and more than compensates for the loss in radiant transfer. The assumption that α equals unity for these tubes is a reasonable allowance for the added convective effects, and simplifies the calculations.

Exchange-Factor.

The remaining term to be evaluated in Equation 2 is the exchange-factor, F . First, we must account for the fact that the gas in the firebox is far from an ideal black body. The only constituents in normal flue gas that contribute significantly to the radiant emission are carbon dioxide and water. Figure 20 shows the amount of water vapor and carbon dioxide in a typical fuel oil on a wet basis. Figure 21 shows the products on a dry basis. The emissivity of a body of gas depends on concentration of each of these two components, the furnace dimensions, and the temperatures of the gas and of the absorbing surface. Lobo and Evans have shown that the composition and dimensional effects can be accounted for by a single term, the

partial pressure of carbon dioxide plus water multiplied by the mean beam length. Figure 22 shows the partial pressure, P , of the radiating components as a function of excess air for the usual hydrocarbon fuels, Table 1 gives the mean beam length, L , for various furnace shapes. It has been found that the tube wall temperature has only a minor effect. Therefore, the emissivity can be correlated as a function of the PL product and gas temperature, as shown in Figure 23. Variations in tube wall temperature between 400 and 1000°F can be seen by the deviation from these curves as 23a thru d.

The exchange-factor is also dependent on the amount of re-radiation from exposed refractory. Energy striking such refractory is, in effect, reflected back toward the tubes. Thus a furnace with a large amount of exposed refractory will transfer more heat per unit of tube surface than one with most of the walls covered by tubes. This effect can be correlated on the basis of the ratio of exposed refractory area to equivalent cold-plane area, as shown in Figure 24. The exposed refractory area is defined as the area which would be exposed if the tube bank were replaced by the equivalent cold plane. That is, it is the total envelope area of the firebox, less the equivalent cold-plane area of all the tubes.

Figure 24 also allows for the fact that the tubes do not completely absorb all the radiant energy that strikes them. The curves are based on a tube-surface absorptivity of 0.9, which is a commonly accepted value for oxidized metal surfaces.

The working equation for calculating radiant heat transfer in the firebox thus becomes Equation (5):

$$q_{R_r} = \sigma \alpha A_{cp} F (T_g^4 - T_t^4) \quad (5)$$

q_{R_r} = Radiant heat transfer by radiation in Radiant Section.

F = Exchange Factor.

α = Efficiency Factor (Hottel)

T_g = Flue Gas Temp., °R

T_t = Tube Wall Temp., °R

3. Convective Transfer

Although radiation accounts for most of the heat transfer in the radiant section, convection cannot be neglected. The relative amount of heat transferred by convection may be from five to twenty percent of the total radiant duty, or more, depending on the temperature level. The relation for convective heat transfer is shown as Equation (6).

$$q_{Rc} = h_{Rc} A_{Rt} (T_g - T_t) \quad (6)$$

q_{Rc} = Heat transfer by convection in Radiant Section, Btu/Hr.

h_{Rc} = Coefficient of heat transfer by convection in Radiant Section

A_{Rt} = Tube Area in Radiant Section

Since convective heat transfer is not the major contributor, we make some simplifying approximations. For the usual furnace, h_{Rc} is about 2.0 Btu/Hr., sq. ft., °F, and A_{Rt} is about two times αA_{cp} . Making these substitutions into Equation 6, we can put it into a form similar to Equation 5 and have Equation 7.

$$\begin{aligned}
 q_{Rc} &= h_{Rc} A_{Rt} (T_g - T_t) \\
 \text{since } h_{Rc} &= \sim 2.0 \text{ BTU/hr ft}^2 \text{ } ^\circ\text{F} \\
 A_{Rt} &= \sim 2 \propto A_{cp} \\
 q_{Rc} &= 2(2 \propto A_{cp})(T_g - T_t) = 4 \propto A_{cp}(T_g - T_t) \quad (7)
 \end{aligned}$$

4. Total Heat Transfer Rate

The total heat absorbed in the radiant section is the sum of the radiant and convective transfer.

$$q_R = q_{Rr} + q_{Rc} = \sigma \propto A_{cp} F (T_g^4 - T_t^4) + 4 \propto A_{cp} (T_g - T_t) \quad (8)$$

dividing both side by $\propto A_{cp} F$

$$\frac{q_R}{\propto A_{cp} F} = \sigma (T_g^4 - T_t^4) + \frac{4(T_g - T_t)}{F}$$

For the usual furnace, the exchange factor, F , is approximately 0.57. Substituting into the right hand side of the above equation:

$$\frac{q_R}{\propto A_{cp} F} = \sigma (T_g^4 - T_t^4) + \frac{4}{0.57} (T_g - T_t)$$

or

$$\frac{q_R}{\propto A_{cp} F} = \sigma (T_g^4 - T_t^4) + 7.02 (T_g - T_t) \quad (8a)$$

This relation is shown in Figures 25 a-c or Figure 28.

Is there any way the convective constituent can be altered in the radiant section?

We find that a tube placed one tube diameter from a re-radiating wall will absorb approximately 27% more heat than a tube directly against a wall (see Figure 25d).

The movement of gases to the walls of a furnace provides both convective heat transfer to the tubes and radiant heat transfer from the refractory walls to the tubes. Moving gases not only transfer convective heat directly to the tubes but also transfers heat by convection to the walls. The heated walls, in turn, radiate this heat to the tubes.

This phenomenon is important to consider where the refiner intends to add insulation to an existing furnace. If insulation is placed over existing insulation on walls shielded by tubes, the distance between tube and wall will decrease. This, in turn, may reduce the heat absorbing capacity of these radiant tubes. Instead of conserving heat with insulation, the furnace is forced to fire harder to account for the loss in heat transfer to the tubes, wasting fuel and potentially causing other difficulties.

.....But how can we determine the Tube Wall Temperature (T_t)

ΔT Across the Tube Metal (ΔT_m)

The basic relation of heat flow by conduction is Fourier's Law:

$$\frac{dq}{dA} = -K \frac{\partial T}{\partial t}$$

where A = Area of isothermal surface

t = Distance measured normally to surface

q = Rate of heat flow across surface in direction normal to surfaces

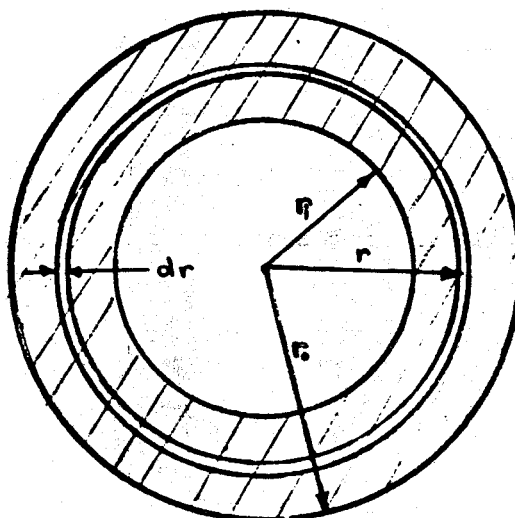
T = Temperature

K = Proportionality constant

In steady state operation, assuming K is independent of T and heat losses are negligible, we have:

$$\frac{q}{A} = -K \frac{dT}{dr} \quad \text{or} \quad dT = -\frac{q}{KA} dr$$

Let us look at heat flow through a tube:



Where r_i = inside radius

r_o = outside radius

r = radius of very thin cylinder between r_i and r_o

dr = thickness of very thin cylinder of radius r

T_i = temperature of inside surface

T_o = temperature of outside surface

L = length of tube

We will assume dr is so small when compared to the thickness of the cylinder (r) that the lines of heat flow may be considered to be parallel.

Our equation for steady conduction can be rearranged as follows:

$$\frac{q}{A} = -K \frac{dT}{dr} \qquad q = -KA \frac{dT}{dr}$$

applying this relation to our cylinder:

$$q = -K (2\pi rL) \frac{dT}{dr}$$

$$\text{where } A_{\text{cyl}} = 2\pi rL$$

$$dr = dr$$

rearranging and integrating

$$\int_{r_i}^{r_o} \frac{dr}{r} = \frac{2\pi LK}{q} \int_{T_o}^{T_i} dT$$

$$\ln r_o - \ln r_i = \frac{2\pi LK}{q} (T_i - T_o)$$

$$q = \frac{K(2\pi L)(T_i - T_o)}{\ln(r_o/r_i)}$$

for convenience let's place this equation in another form for simplification

$$q = \frac{K \bar{A}_L (T_i - T_o)}{r_o - r_i}$$

in order for the equation to be correct

$$\frac{K \bar{A}_L (T_i - T_o)}{r_o - r_i} = \frac{K (2\pi L)(T_i - T_o)}{\ln(r_o/r_i)}$$

solving for $\bar{A}_L$

$$\bar{A}_L = \frac{2\pi L(r_o - r_i)}{\ln(r_o/r_i)}$$

note that the logarithmic mean radius, $\bar{r}_L = \frac{r_o - r_i}{\ln(r_o/r_i)}$

therefore $\bar{A}_L = \frac{A_o - A_i}{\ln(A_o/A_i)} = \text{logarithmic mean area}$

rearranging our basic equation

$$\Delta T_m = T_i - T_o = \frac{q}{\bar{A}_L} \left(\frac{t}{K} \right)$$

where $t = r_o - r_i$

To put the heat flux, q/A in terms of the outside circumferential area, we note

$$\bar{A}_L = \frac{A_o - A_i}{\ln(A_o/A_i)} = 2\pi L \bar{r}_L = \pi \bar{D}_L L$$

where A_i = inside circumferential area

A_o = outside circumferential area

$\bar{D}_L$ = logarithmic mean diameter

and for tube sizes we work with, we can fairly accurately assume

$$\bar{A}_L = \frac{A_o + A_i}{2} \quad \text{and} \quad \bar{D}_L = \frac{D_o + D_i}{2}$$

where D_i = inside diameter

D_o = outside diameter

therefore

$$\frac{D_L}{D_o} = \frac{A_L}{A_o} \quad \text{or} \quad A_L = A_o \left(\frac{D_L}{D_o} \right)$$

returning to our basic equation

$$\Delta T_m = \frac{q}{A_L} \left(\frac{t}{K} \right) = \frac{q}{A_o} \left(\frac{D_o}{D_L} \right) \left(\frac{t}{K} \right)$$

since $D_i = D_o - 2t$

therefore

$$\bar{D}_L = \frac{(D_o - 2t) + D_o}{2} = D_o - t$$

substituting

$$\Delta T_m = \frac{q}{A_o} \left(\frac{D_o}{D_o - t} \right) \left(\frac{t}{K} \right)$$

ΔT Across the Inside Film (ΔT_f)

The heat transfer coefficient, h , is defined generally by the following equation:

$$h = \frac{dq/dA}{T_f - T_w}$$

Where q = rate of heat flow across surface in direction normal to surface

A = area of isothermal surface

T_f = bulk temperature of fluid

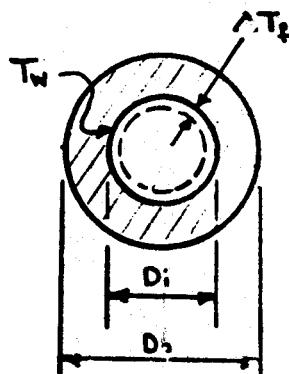
T_w = temperature at wall

Rearranging gives

$$(T - T_w) = \Delta T_f = \frac{dq}{dA} \left(\frac{1}{h} \right)$$

or

$$\Delta T_f = \frac{q}{A} \left(\frac{1}{h} \right)$$



Expressing the flux rate on an outside tube area basis and correcting the flux rate to its location on the inside tube area:

$$\text{Since } \frac{D_o}{D_i} = \frac{A_o}{A_i}$$

$$\Delta T_f = \frac{q}{A_o} \left(\frac{1}{h} \right) \left(\frac{D_o}{D_i} \right)$$

Tube Metal Temperature (T_t or TWT)

$$TWT = T_f + \Delta T_f + \Delta T_m$$

$$= T_f + \frac{q}{A_o} \left(\frac{1}{h} \right) \left(\frac{D_o}{D_i} \right) + \frac{q}{A_o} \left(\frac{D_o}{D_o - t} \right) \left(\frac{t}{K} \right)$$

Please Note:

If there are coke deposits and scale deposits the equation becomes

$$TWT = T_f + \Delta T_f + \Delta T_{\text{coke}} + \Delta T_{\text{scale}} + \Delta T_m$$

Individual Heat Transfer Coefficients

Earlier, we touched upon the subject of heat transfer coefficients in calculating the temperature drop across the inside tube film. We should note the heat transfer coefficient is an artificial property used to determine convective heat transfer.

If a fluid in turbulent flow flows through a tube, there are three separate regions. There is a thin layer of fluid at low velocity at the wall and a turbulent region at the center of the tube. Between the two regions is a third, a transition region. The velocity gradient is largest at the wall and smallest at the tube center. The temperature gradient is affected similarly; it is large at the wall and through the viscous sublayer, in rapid change through the transition zone and small in the turbulent core.

The viscous sublayer is of primary concern with respect to heat transfer. Heat transfer through this layer may be described as approximating conduction due to its low velocities. Other fluid regions may be considered to be fairly equal in temperature as a result of rapidly moving eddy currents within them.

The overall resistance to the flow of heat is a result of the two viscous layers on either side of the tube wall and the wall itself. The overall heat transfer coefficient is best approximated by analyzing it in terms of the three aforementioned resistances. This approach requires the use of individual heat transfer coefficients for the fluid streams on either side of the wall.

The individual heat transfer coefficient, h , is defined generally by the equation:

$$h = \frac{dq/dA}{T_f - T_w}$$

It is not a measurable quantity as is length and volume but is a theoretical quantity produced empirically through experimentation. Throughout the literature, individual heat transfer coefficient correlations differ with one another considerably. One can open 5 separate engineering books only to discover each has its own correlation for a specific substance. Many employ the Reynolds, Prandtl and Nusselt numbers.

For our purposes, we recommend the Sieder-Tate method for determining the individual inside heat transfer coefficient for all systems except hydrocarbon vapor (one phase). The Sieder-Tate equation is as follows:

$$h = \frac{K}{D} \left[\frac{CZ}{K} \right]^{1/3} \left(\frac{Z}{Z_w} \right)^{0.14} \phi \left(\frac{DG}{Z} \right)$$

See figure 45 for the nomenclature.

When determining the individual heat transfer coefficient in hydrocarbon vapor phase systems we use the following correlation by McAdams:

$$H = \frac{16.6 C_p G^{0.8}}{D_i^{0.2}}$$

Where C_p = heat capacity, Btu/lb°F

G = mass velocity, lb/ft²Sec

D_i = inside diameter of tube, in.

Heat Distribution in the Radiant Section

Heater tubes in the radiant section of refinery furnaces do not utilize their available surface to the fullest. Picture a heater tube intersected by a plane perpendicular to the centerline of the tube. Let us center our attention on the intersection of the plane and the outside circumferential area.

Ideally at all points on this circumference, the heat flux will be equal. However, due to the geometry of the heater and the burner and wall orientation we find the circumferential flux distribution to vary considerably. Figure 25e demonstrates the radiant circumferential heat flux distribution for a heater tube receiving heat from a radiant source on one side of the tube (refractory wall on the other) and a heater tube receiving heat from radiant sources on both sides.

Diagram A of Figure 25e displays the circumferential flux distribution of a heater tube shielding a refractory wall. The area of maximum heat transfer is immediately opposite the radiating plane while the point of minimum heat transfer is 180° away, immediately opposite the refractory wall. The minimum point does not "see" the radiating plane directly and only receives radiant heat indirectly from radiant rays rebounding off the refractory walls.

Similarly, we observe the circumferential flux distribution for a tube accepting radiant heat from sources on either side in Diagram B of Figure 25e. As we can see, the surface area

is better utilized but still is not perfect. We note the direct radiation profile on either side of the tube mirrors that of the direct radiation profile of the hot side of the shielded (single fired) wall tube. In comparison, if we set a maximum point heat flux in each of the two examples, we can see that the overall heat transferred would be greater in the double fired (radiating planes on both sides) tubes.

This may be again evidenced in Figure 25f, a chart provided in the second edition of API RP 530. If we compare shielded tubes vs. double fired tubes, we find that the ratio of maximum flux rate to the average at a centerline to centerline spacing divided by the tube outside diameter of 1.8 is 1.90 for the former and 1.22 for the latter. What this means is that if we are, for some reason, limited by a certain maximum circumferential flux, a double fired tube will have an average flux rate more than 55% ($1.9/1.22$) greater than the single fired tube or, to put it in other terms, the total surface area of the single fired tube will be capable of absorbing 55% more heat.

Figure 25f also demonstrates the effect of increasing the distance between the tubes. As the centerline to centerline distance increases, the ratio of the maximum to average flow decreases. This is more pronounced in single fired tubes. With single firing, more heat is transferred to the tube by re-radiation of the refractory wall as the tube spacing is increased; as tube spacing increases, direct radiant rays formerly hitting a tube will miss the tube, hit the refractory and rebound to another tube.

The discussion, so far, made no mention of the convective constituent of the overall heat transfer in the convection section. The heat distribution with the convective component further rounds out each curve, in particular, the single fired system. As we have seen before, the distance of the tube from the wall will affect the overall heat absorption of the tube.

In many furnaces, there is a longitudinal flux distribution as well as a circumferential flux distribution. Unfortunately, this variation is not as easily placed upon paper. This longitudinal variation usually rests between 1.0 and 1.5 and may vary considerably between different sections of the furnace.

5. Radiant Heat Balance

To determine the temperature of the flue gas in the fire-box, we must now consider the heat balance around the furnace. Heat is put into the radiant section from three sources - the net heat of combustion q_n , the sensible heat of the combustion air q_a , and the sensible heat of the fuel q_f . Heat is removed in three ways: that absorbed by the tubes q_R , the losses q_L , and the sensible heat of the exit flue gas q_{g2} . Expressed as Equation 9:

$$q_n + q_a + q_f = q_R + q_L + q_{g2} \quad (9)$$

OR

$$q_R = q_n + q_a + q_f - q_L - q_{g2} \quad (10)$$

q_n = Net heat of combustion

q_a = Sensible heat of combustion air

q_f = Sensible heat of fuel

q_R = Absorbed by tubes

q_L = Losses

q_{g2} = Sensible heat of exit flue gas

Equation 10 can be modified to a form similar to Equation 8a.

$$\frac{q_R}{\alpha A c_p F} \left[1 + \frac{q_a}{q_n} + \frac{q_f}{q_n} - \frac{q_L}{q_n} - \frac{q_{g2}}{q_n} \right] = \frac{q_n}{\alpha A c_p F} \quad (11)$$

The net heat release q_n is established as soon as the total heater duty and efficiency are established. The values of q_a and q_f are fixed by the temperature of the fuel and combustion air. The loss q_L is usually from one to three percent of the net heat release, and can generally be estimated

from past experience. This leaves q_{g2} as the only variable on the righthand side of Equation 11. It turns out that the fraction of net heat-release retained in the flue gas is primarily a function of temperature and excess air, for all common liquid and gaseous fuels. This relation is shown in Figure 26a - d for a typical oil. Data for sixteen fuel gases and seventeen fuel oils agree with the curves, Fig. 27, within two percent.

6. Temperatures

Figure 28 shows the rate of heat transfer between a mass of gas at one uniform temperature, and a tube surface at another uniform temperature. In most actual furnaces, however, neither of these temperatures is constant. Average effective temperatures must be selected in order to use Figure 28.

Tube wall temperature depends on the temperature of the fluid inside, the intube transfer coefficient, the thermal resistance of the tube wall, and the total heat flux. Fluid temperature can almost always be taken as the arithmetic mean of the radiant section inlet and outlet. The difference between fluid and tube wall temperatures can, if necessary, be determined from the usual intube film-coefficient correlations. Since radiant absorption rate is fairly insensitive to tube wall temperature, however, it is usually accurate enough to add 100°F to the average fluid temperature.

The average effective gas temperature depends on the general shape of the furnace and the firing conditions. Experience has shown that for heaters of the type with an approximately square cross-section and no large areas of refractory with direct flame impingement, the average gas temperature is very close to that of the exit gas. For some other types, however, such as a cylindrical furnace with a high length/diameter ratio, the average firebox gas is appreciably hotter than the exit gas. The magnitude of this difference must be determined empirically from experience with similar designs.

7. Radiant Section Absorption

Equation 8a gives the heat transferred to the tubes in the radiant section as a function of average gas temperature. Equation 11 gives the heat removed from the flue gas as a function of exit temperature. Simultaneous solution of the two equations gives the firebox temperature and the amount of heat transferred.

The solution is most easily obtained graphically by superimposing a plot of Equation 11 on Figure 28. The value of t_g is read where the plot of Equation 11 crosses the applicable tube wall temperature. Then, from a heat balance, q_r is readily determined. Note that it is not necessary to plot Equation 11 over the full temperature range. Only that part which crosses the curves in Figure 28 is required. Furthermore, since Equation 11 is nearly linear with respect to gas temperature, two calculated points close to the intersection, joined by a straight line, are usually sufficient.

D. Convection Section

1. General

As in the radiant section, heat is transferred in the convection section by both radiation and convection. A firm basis for calculating convection-section heat transfer was first outlined by Monrad. He accounted for convection heat radiation from the gas, and radiation from the refractory walls. The method outlined here follows Monrad's theory, but with revised correlations to include later experimental results.

2. Convective Heat Transfer

The correlation offered by Monrad has been revised to include later and more extensive data for film transfer coefficients.

$$h_{Cc} = \frac{2.14 (T_f)^{0.28} (G_{max})^{0.6}}{d^{0.4}} \quad (12)$$

h_{Cc} = Convection Coefficient of Heat Transfer in the Convection Section

T_f = Temperature of film, °R

G = Mass Velocity flue gas, lbs/sec/sq.ft.

d = Diameter, O.D. tube inches

Figure 29 shows the convection coefficient for several common tube sizes. The average gas film temperature is defined as the average intube temperature plus one-half the log mean temperature difference from flue gas to fluid. The flue gas rate is at the minimum cross-section in the convection section. It has been found that flue gas rates for all ordinary gaseous and liquid fuels can be correlated

on the basis of net heat release and percent excess air. The relation is shown in Figure 27.

3. Radiation from Flue Gas

A method for calculating the heat transfer by radiation from flue gas to a bank of tubes has been presented by Hottel. His basic relation was solved using typical tube spacing of about two tube diameters, a tube surface absorptivity of 0.9, and a typical partial pressure of carbon dioxide plus water. The results are shown on Figure 30. The average gas temperature is defined as the average intube fluid temperature plus the log mean temperature difference from the flue gas to fluid. The average tube wall temperature may be taken as average fluid temperature plus 100°F.

4. Radiation from Refractory Walls

Following Monrad's pattern, we express the effective coefficient for wall-to-wall transfer as a function of tube wall temperature, as shown in Figure 31. This coefficient, together with the relative wall and tube areas in the convection section, determines a correction factor f , Equation 13:

$$f = \frac{h_{Cw}}{h_{Cc} + h_{Cr} + h_{Cw}} \frac{A_{Cw}}{A_{Ct}} \quad (13)$$

h_{Cw} = heat transfer coefficient in Convection Section from refractory walls

h_{Cr} = heat transfer coefficient in Convection Section due to radiation

A_{Cw} = Refractory wall area in convection section

A_{Ct} = tube area in Convection Section.

5. Overall Heat Transfer Coefficient

The total apparent gas film coefficient is determined from the individual coefficients by the following Equation 14:

$$h_c = (1 + f)(h_{Cc} + h_{Cr}) \quad (14)$$

The intube coefficient is calculated by standard methods, based on the flow-rate and physical properties of the fluid. The resistance of the tube wall is small, and is usually neglected. The over-all transfer coefficient in the convection section then becomes Equation 15.

$$V_c = \frac{h_c(h_{Ci})}{h_c + h_{Ci}} \quad (15)$$

In many rating problems, particularly during preliminary work, the detailed calculations outlined above are not justified. For these cases, the curves in Figure 32 can be used. These represent overall convection section coefficients, assuming usual tube arrangements, fluid temperatures, and intube film coefficients.

If we refer to the literature on heat transfer by convection the overall heat transfer coefficient is given by the following:

$$V_o = \frac{1}{\frac{D_o}{D_i h_{di}} + \frac{D_o}{D_i h_i} + \frac{t_w}{k_m} \left(\frac{D_i}{D_L} \right) + \frac{1}{h_o} + \frac{1}{h_{do}}}$$

or

$$V_i = \frac{1}{\frac{1}{h_{di}} + \frac{1}{h_i} + \frac{t_w}{k_m} \left(\frac{D_i}{D_L} \right) + \frac{D_i}{D_o h_o} + \frac{D_i}{D_o h_{do}}}$$

where V_i = overall heat transfer coefficient based upon inside tube area

V_o = overall heat transfer coefficient based upon outside tube area

D_i = inside tube diameter

$\bar{D}_L$ = logarithmic mean tube diameter

D_o = outside tube area

h_{di} = fouling factor for deposits on inside of tubes

h_{do} = fouling factor for deposits on outside of tubes

h_i = inside heat transfer coefficient

h_o = outside heat transfer coefficient

K_m = thermal conductivity of tube

t_w = tube wall thickness

If we look at this overall coefficient more closely and make the following assumptions and observations:

- 1) The ratio of the diameters is small and approaches unity,
the overall coefficient becomes

$$V_o = \frac{1}{\frac{1}{h_{di}} + \frac{1}{h_i} + \frac{t_w}{K_m} + \frac{1}{h_o} + \frac{1}{h_{do}}}$$

- 2) The resistance due to the tube wall is negligible in comparison with the other resistances

$$V_o = \frac{1}{\frac{1}{h_{di}} + \frac{1}{h_i} + \frac{1}{h_o} + \frac{1}{h_{do}}}$$

- 3) Fouling is negligible

$$V_o = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}}$$

If we multiply both the numerator and the denominator of the right hand side by $h_0 h_1$, we have

$$V_o = \frac{h_0 h_1}{h_0 + h_1} \text{ or equation (15), } V_o = \frac{h_c (h_{ci})}{h_c + h_{ci}}$$

Ideally, this equation should be satisfactory but in cases where fouling may be substantial, we must include fouling resistances.

6. Extended Surface

Extended surface on the tubes in the convection section is often used as a means of obtaining additional heat transfer per unit of prime tube surface. Convective transfer coefficients for such tubes depend upon the exact size and arrangement, and must be determined for the particular type being considered. It should be noted that the extended surface increases only the convective transfer. Radiant transfer must be computed on the basis of prime tube surface. With most types of extended surface now being used, radiant transfer to the convection section tubes is so small that it could be neglected.

EXTENDED SURFACE

A. Finned Surface

1. history

In order to lay a foundation for our discussion, some background on the history of high temperature fin tubes will be discussed. An early method of applying extended surface to a tubular product involved cast iron rings which were shrunk to the tube in such a fashion as to transfer heat from the tube. This type of extended surface, known as "gill rings", was very expensive, extremely heavy, and has been replaced by more modern methods of attaching extended surface. Early contributions to the industry included the brazing of both

longitudinal and square plate-type extended surface to tubular products. This also remained a costly and less attractive method of attachment.

At this point, two methods of attaching a spiral-wound fin to a tubular product-were introduced to the industry and received immediate acceptance and widespread use. One involved grooving a tubular product and rolling a smooth, continuous fin into the groove, resulting in a tight mechanical bond. The other method used cut-type fins with a rolled underfoot that was spirally wound around the tube and spotwelded by resistance-welding equipment to the tubular product. At about this same time, longitudinal type fins were being attached to tubular products in the same manner as the spiral wound product. In addition to the spiral-wound type fin tubes, a "stud" type fin is also available which is resistance welded to the tube. This product has found application in vertical-type furnaces and in some high temperature services. However, owing to its low external surface ratio, it does not compete in general applications with the spiral-wound type fins. Another spiral-wound fin tube employs a heli-arc weld at the base of the fin stock providing a continuous fillet weld.

A more recent development in spiral-wound fin tubes involves a smooth continuous fin which is tension wrapped around the tube and high-frequency welded as it is wrapped onto the tube.

The latest of the spirally wrapped fin tubes is an advancement or improvement in two previously mentioned type of tubes.

The first is a continuously welded cut type fin in lieu of the old spaced type weld. The second is a serrated high frequency welded fin tube.

B. Serrated Fins (Fig. 48)

1. Rating Instructions

All fin tube manufacturers have rating instructions. We feel that there are no short cut methods that are sound as they contain flaws and limit their scope and seriously affect their accuracy.

2. Type of weld

The most widely accepted type of weld in recent years is the continuous weld. The flange of the fin is continuously welded to the tube by a resistance weld which leaves the tube unchanged metallurgically; hence, normalizing after welding is unnecessary, regardless of alloy. The width of this weld is always greater than the thickness of the fin. This provides a broad heat transfer path from the fin tip to the inside of the tube. If space welding is to be used, the tip of each weld should not be greater than 1/8" apart.

3. Cleanability

Soot blower manufacturers have demonstrated that the serrated fin is easier to clean than the continuous fin. The reason is that the space between the segments permits lateral flow of the gas during operation (as well

as the steam during cleaning). This keeps buildups between the fins from accumulating rapidly and permits more effective steam cleaning.

C. Solid Fin (Fig. 48)

1. Strength

A few operating companies feel that the solid fin is easier to handle and replace in the field. The continuous fin apparently slides through the tube sheets and supports easier.

2. Thicker Fins

The main advantage of a solid fin which used the high frequency welding process is the fact that the fins can be made much thicker than the standard frequency process. Fin thicknesses can go all the way up to .1875". By increasing the fin thickness, one raises the fin efficiency enabling one to use fins at higher temperatures. In other words, at temperatures previously too high for fin tubes.

One of the important considerations for higher type of fin tube is the recommended fin tip temperatures for a given material should be used.

The most commonly used materials and the maximum fin tip temperatures are as follows:

Carbon Steel	950°F
11% Chrome	1250°F
18-8	1650°F
25-12 or 25-20	2000°F

3. Type of Weld

The base of the fin is continuously welded to the tube by a special, high frequency resistance welding process.

The width of this continuous weld will be no less than 90% of the thickness of the fin. This provide a broad heat transfer path from the fin tip to the inside of the tube. This is actually a forge type weld of the I type (no flange). There is a thickening at the base of the fin which is the fin metal.

D. Figures 49 & 50 - Tube Failures

E. Studs (Fig. 51, 52 & 53)

The studding followed essentially the same development as the finning, however, it was developed for boilers and then adapted to process heaters.

F. Calculations

1. General

In the foregoing example we will replace the bare tube convection bank with a solid fin bank that is suitable for firing fuel oil.

As the cost of fuel increases, it is increasingly important to improve the operating efficiency of refinery heating equipment. One way to improve the efficiency of fired heaters is to increase the heat recovery in the convection section by using finned tubes. In actual practice, the sample we are using would be totally redone but for now we will just substitute tube banks.

2. Rating Method

This method is basically for finned-tube convection sections. The rating problem is divided into three parts:

- a. The heat transfer for flow of gases across banks of finned tubes.

- L. The radiation leakage from the combustion chamber to the convection section
- C. The gas pressure drop across the bank of finned tubes

The total heat transferred to the finned tubes is the radiation leakage through the shield coil (if any) plus the heat transferred by convection and radiation within the tube bank.

$$q_c = q_{sL} + U_c A_t \text{ (LMTD)} \quad (20)$$

q_c = Convective heat rate, Btu/Hr

q_{sL} = Heat Rate Shield leakage

U_c = Convective overall heat transfer coefficient, Btu/Hr/Sq.Ft./°F

A_t = tube area, Sq. Ft.

Heat Transfer for Cross Flow of Gases

The classic reference on rating fired-heater convection sections is a paper Monrad published in 1932. In this paper, Monrad developed the relations covering heat transfer to the convection section by convection from the gases, radiation from the gases, and radiation from the refractory walls. He used convection heat transfer data for smooth tubes only. He did not consider the additional heat input due to radiation leakage from the combustion chamber when the convection section can "see" the radiant section through the shield coil.

After Monrad's paper was published, Colburn developed a more satisfactory relation for the convection coefficient.

$$\frac{h_c D_o}{k} = 0.36 \left(\frac{c_p \mu}{k} \right)^{1/3} \left(\frac{D_o G_{max}}{\mu} \right)^{0.6} \quad (21)$$

h_c = Convective Film heat transfer coefficient
and BTU/hr/sq. ft. °F

D_o = Outside diameter, ft.

k = Thermal Conductivity, BTU/hr sq. ft., °F/ft.

c_p = Specific Heat, BTU/#

μ = Viscosity, Lb/ft.hr.

G = Mass Velocity, Lb/sq. ft./sec.

The constants used here are not those originally proposed by Colburn, but are revised values in accord with extensive data published later by Huge, Pierson and Tucker.

Colburn's equation and the data of Huge, Pierson and Tucker are for plain tubes. Gardner developed a general relationship for the effectiveness of a fin which can be used to compute the film coefficient for a finned tube from the coefficient for a plain tube of equal root diameter. Gardner's general relationship for rectangular cross-sectional, helical fins is:

$$E = \frac{\tanh(u_b)}{u_b} \quad (22)$$

E = fin efficiency

u_b = at base of fin

where

$$u = x \sqrt{\frac{2 h_c}{KY}} \quad (23)$$

x = fin height, in.

Y = cross sectional width of fin, in.

The film coefficient for the finned tube is related to the coefficient for a plain tube by the following equation.

$$h_f = h_c \frac{(EA_f + A'_t)}{A'_t} \quad (24)$$

h_f = film heat transfer coefficient,
Btu/Hr, Sq. Ft., °F

A_f = Fin Area, Sq. Ft./Row

A'_t = Total Surface Area of plain tube, Sq. Ft./Row

The fin efficiency, E, depends on the fin dimensions, the thermal conductivity of the fins and the convective heat transfer film coefficient. In developing Equation (22), Gardner made the following assumptions:

- a. The heat flow and temperature distribution throughout the fin is independent of time; i.e., heat flow is steady.
- b. The fin material is homogeneous and isotropic.
- c. There is no heat source in the fin itself.
- d. The heat flow to or from the fin surface at any point is directly proportional to the temperature difference between the surface at that point and the surrounding fluid.
- e. The thermal conductivity of the fin is constant.
- f. The heat transfer coefficient is the same over the entire fin surface.
- g. The temperature of the fluid surrounding the fin is uniform.
- h. The temperature of the fin base is uniform.

- i. The joint between the fin and tube is assumed to offer no bond resistance.
- j. The fin thickness is so small compared with its height that temperature gradients across the width of the fin may be neglected.
- k. The heat transferred through the outermost edge of the fin is negligible compared with the passing into the fin through the sides.

Figures 54 and 55 give the graphical solution of Equation (22) for annular fins and spines respectively. Note that the use of a fin efficiency factor makes Equation (24) useful for both annular fins and spines.

Monrad's basic reference for radiant heat transfer from non-luminous gases is a paper by Hottel published in 1927. Hottel has since published the material, with later data, as a chapter in McAdams' book on heat transmission. The apparent heat transfer coefficient for radiation from the hot gases to the convection section is

$$h_r = \frac{(\sigma) \frac{(1 + e_t)}{2} (e_g T_g^4 - e_t T_t^4)}{(T_g - T_t)} \quad (25)$$

σ = Stefan-Boltzman Constant

e_t = Emissivity of tube at tube wall temperature

e_g = Emissivity of gas at radiating temperature

T_g = Temperature of gas, °R

T_t = Temperature of tube wall, °R

To account for radiation from refractory walls to the tubes, Monrad expressed the radiant heat transmission as a fraction of the heat transferred directly to the tubes.

$$B = \frac{h_w}{(h_c + h_r + h_w)} \frac{A_w}{A_t} \quad (26)$$

B = Factor for wall radiation in convection section

h_w = Refractory wall heat transfer coefficient,
Btu/Hr,Sq.Ft., °F

h_r = Radiant heat transfer coefficient,
Btu/Hr,Sq.Ft., °F

A_w = Refractory wall area, Sq.Ft.

Four factors must be considered in order to evaluate the heat transfer for cross flow of gases. (1) The convective heat transfer coefficient. (2) The fin efficiency. (3) The radiation from the gas. And (4) the radiation from refractory walls.

There are a number of tube arrangements possible, but for the purpose of this presentation equilateral triangular pitches of 1.5 and 2.0 times the nominal diameter of the tube are chosen as standard. Therefore, when a tube size is selected, the tube spacing is fixed by the chosen standard of tube arrangement. The tube size, the number of tubes per row, the area per row and the minimum cross-sectional area for the flue gas flow are considered in this rating presentation. Usually, in fired-heaters, the tube size in the radiant section is the same as in the convection section, and the number of tubes per row is such that the flue gas velocity is about or below 15 ft/sec. The area per row is the

exposed area per tube times the number of tubes. The minimum cross-section area for flue gas flow is the total cross-sectional area of the convection section less the projected area of one row of tubes.

The design conditions for the convection section are set by the overall design conditions and by the radiant section rating. The flue gas rate, the stack heat content, the stack temperature and the intube fluid inlet temperature must satisfy the overall design conditions. The flue gas inlet temperature and the intube fluid outlet temperature must satisfy the over-all design conditions; and they must fit in with the radiant section rating, if the exchanger is a fired-heater. The design temperature data are used to calculate the log mean temperature difference, the average gas film temperature and the average gas temperature.

For simplicity, the convection coefficient correlation proposed by Colburn is used with revised coefficients based on the data of Huge, Pierson and Tucker. Equation (21) is simplified as follows, for use in computing flue gas film heat transfer coefficients:

$$h_c = \frac{2.14 T_f^{0.28} (G_{max})^{0.6}}{d_o^{0.4}} \quad (21)$$

T_f = flue gas film temp., °R

The effective heat transfer coefficient for a finned tube is given by Equation (24).

The method for calculating the amount of heat transferred by radiation from the flue gas within the convection section was presented by Hottel. For finned tubes, the gas radiation coefficient is based on the root diameter of the tube. The area used to compute the radiant heat transfer is the area of the bare tube. The basic relation for standard tube-to-tube clearances (1.5 and 2.0 diameters apart) was solved by assuming a tube surface emissivity of 0.90, and a constant partial pressure of carbon dioxide and water. The apparent heat transfer coefficient for radiation from the flue gas within the convection section is given by Equation (25). Figure 56 is a graphical solution of Equation (25). The average gas temperature is defined as the average intube temperature plus the log mean temperature difference from gas to intube fluid. The average tube wall temperature is approximately the average intube temperature plus 100°F.

Radiation from the refractory walls is still another method of transferring heat to the convection section tubes. It is assumed that the convective and radiative coefficients for heat transfer to the refractory are the same as the coefficients for heat transfer to the tubes, and any heat loss from the convection section is neglected. Monrad expressed the indirect radiant transfer as a fraction of heat transferred directly to the tubes in Equation (26). He assumed that T_w is equal to $1.25 T_t$, that the tubes and walls form infinite parallel planes, and the wall emissivity is 1.0. On this basis, Equation (28) is plotted in Figure 57.

$$h_w = 9.46 \left(\frac{T_t}{1000} \right)^3 \quad (28)$$

The overall heat transfer coefficient for the convection section is calculated from the apparent gas film coefficient and the intube film coefficient. This apparent gas film coefficient is

$$h_G = (1 + \beta)(h_f + h_r) \quad (29)$$

Radiation Leakage from Combustion Chamber

The Lobo and Evans method for calculating the heat absorption in furnace radiant sections is the basis for Wimpres' method which is used here to calculate the radiant heat which leaks through the shield coil into the convection section. The difference between the incident radiant energy and the direct absorption is the radiation leakage absorbed by the convection section

$$q_{sL} = C(1 - \alpha) A_{cp} F (T_g^4 - T_t^4) \quad (30)$$

where A_{cp} is the cold plane area of the shield coil, F is the radiant exchange factor of the radiant section, T_g is the radiating temperature in the combustion chamber, T_t is the shield coil tubewall temperature, and C is a factor of comparison for direct radiation. Figure 58 is a plot of Equation (30) rearranged as

$$\frac{q_{sL}}{(1 - \alpha) A_{cp} F} = C(T_g^4 - T_t^4) \quad (31)$$

In evaluating the shield coil leakage, the tubewall temperature at the cross-over between the convection section and the radiant section is used. Radiation leakage through the shield coil to the convection section does not occur if the convection section does not "see" the radiant section

through the shield coil.

Pressure Drop Across the Bank of Finned Tubes.

Gunter and Shaw presented the normal form of the friction factor for a specific tube bank as

$$f = \Delta P \frac{2g_c \rho_g}{(G_{\max})^2} \left(\frac{D_v}{L} \right) \left(\frac{\mu}{\mu_w} \right)^{0.14} \left(\frac{D_v}{S_T} \right)^{-0.4} \left(\frac{S_L}{S_T} \right)^{-0.6} \quad (32)$$

ΔP = Pressure Drop, inches H_2O

g_c = Gravitational constant, 32.17 lbm ft/lbf sec²

ρ_g = Density of gas, lbs/ft³

D_v = Volumetric hydraulic diameter = $\frac{4 \text{ (Net Free Volume)}}{\text{Friction Surface}}$, ft.

L = Mean length of radiant beam

μ_w = Viscosity water, lb/ft hr.

S_T = Transverse pitch, center-to-center distance from one tube to another in the same row, ft.

S_L = Longitudinal pitch, center-to-center distance from tube in one row to the tube in the next row, ft.

Equation (32) is based on the intensive data of Seider and Scott, Hoge and Pierson. The ratio of viscosities to the 0.14 power is approximately 1.0, and for equilateral spacing, S_L equals S_T . Then

$$P = \frac{f (G_{\max})^2 L}{2g_c \rho_g D_v} \left(\frac{D_v}{S_T} \right)^{0.4} \quad (33)$$

Now, combining the constants together and converting the pressure drop into inches of water

$$\Delta P = 0.003 \frac{f (G_{\max})^2 L}{\rho_g D_v} \left(\frac{D_v}{S_T} \right)^{0.4} \quad (34)$$

The friction factor, f , is plotted as a function of the Reynold's number in Figure 59.

COMPARISON OF BARE TUBE VS. FINNED TUBE CONVECTION SECTION

Taken from:

- 1) Rating Fired Heaters by R. N. Wimpres
Hydrocarbon Processing, Oct. 1973, V42, No. 10
- 2) How to Rate Finned-Tube Convection Section in Fired Heaters
by J. L. Schweppe & C. A. Torrijos
Hydrocarbon Processing, June 1964, V43, No. 6

Process Conditions:

98,640 lb/hr flue gas in at 1610°F, out at 820°F
Process fluid in at 380°F, out at 483°F

$$h_i = 100$$

Duty required = 23.4×10^6 BTU/HR

Use 6, 4" tubes per row, 8" C to C spacing,
triangular spacing, Eff L = 38.5 ft.

- 1) Bare using Wimpres' method
- 2) Finned with 1" high, 2 - 0.05" thick fins per inch
using the Schweppe and Torrijos method

Temperature Differences:

$$\begin{array}{rcl} \text{Hot} - 1610 & \text{Cold} - 820 & \text{LMTD} - \frac{1127-440}{\ln\left(\frac{1127}{440}\right)} = 730^\circ\text{F} \\ -483 & -380 & \\ \hline 1127^\circ\text{F} & 440^\circ\text{F} & \end{array}$$

Average Temperatures:

$$\begin{array}{rcl} \text{Oil} & = (380 + 482)/2 & = 432^\circ\text{F} \\ \text{Tube Wall} & = 432 + 100 & = 532^\circ\text{F} \\ \text{Gas} & = 432 + 731 & = 1163^\circ\text{F} \\ \text{Gas Film} & = 432 + 731/2 & = 798^\circ\text{F} \end{array}$$

WIMPRESS

SCHWEPPE & TORRIJOS

Gross Width	6.5 (8) = 52	6.5 (8) = 52"
Projected Width	6 (4.5) = 27 25" = 2.08'	6 [4.5 + 2 (0.05)(1)(2)] = 28.2 23.8
Free Width		= 1.98'
Mass Velocity	$\frac{98640}{3600(38.5)(2.08)} = \frac{0.342 \text{ lb}}{\text{ft}^2 \text{ Sec}}$	$\frac{98640}{3600(38.5)(1.98)} = \frac{0.359 \text{ lb}}{\text{ft}^2 \text{ Sec}}$
Surface	$A_t = \frac{\pi (4.5) (38.5) (6)}{12}$ = 272 ft ² /row	$A_f = \left[\frac{\pi (6.5^2 - 4.5^2)}{4} \right] (2) (12) (38.5) (6)$ = 1390 ft ² /row
Radiation leakage through shield		(1-α) AcpF = (1-0.76) (154) (0.63) = 25.4 ft ² Tg = 1610°F Assuming TWT = 483 + 100 = 583°F from Fig. 6 $\frac{qSL}{(1-\alpha)AcpF} = \frac{29600 \text{ Btu}}{\text{ft}^2 \text{ Hr}}$ qSL = 29600 (25.4) = 0.75 x 10 ⁶ Btu/Hr
h _{cc}	$h_{cc} = \frac{2.14(798+460)^{0.28}(0.342)^{0.6}}{4.5^{0.4}}$ = 4.54	$h_{cc} = \frac{2.14(798+460)^{0.28}(0.359)^{0.6}}{4.5^{0.4}}$ = 4.60 fin efficiency: $\sqrt{\frac{2h}{KY}} = \frac{1}{12} \sqrt{\frac{2(4.68)}{31(0.05/12)}}$ = 0.705

WIMPRESS

SCHWEPPE & TORRIJOS

		$d_f/d_o = 6.5/4.5 = 1.445$ $E = 0.84$ $h_f = 4.68 \left[\frac{0.84(1390)+272}{272} \right]$ $= 24.77$
h_{cr}	From Figure 10, $h_{cr} = 2.25$	From Figure 4, $h_{cr} = 2.25$
h_{cw}	$h_{cw} = 9.46 \left[\frac{(532+460)}{1000} \right]^3$ $= 9.23$	$h_{cw} = 9.46 \left[\frac{(532+460)}{1000} \right]^3$ $= 9.23$
Wall Area	$2 \left(\frac{8}{12} \right) (0.866) (38.5) = 44.45 \text{ ft}^2$	$2 \left(\frac{8}{12} \right) (0.866) (38.5) = 44.45 \text{ ft}^2$
f	$\beta = \frac{9.23}{4.54+9.23+2.25} \left(\frac{44.45}{272} \right)$ $= 0.0942$	$\beta = \frac{9.23}{24.77+9.23+2.25} \left(\frac{44.45}{272} \right)$ $= 0.0416$
h_c	$h_c = (1+0.0942) (4.54+2.25)$ $= 7.43$	$h_c = (1+0.0416) (24.77+2.25)$ $= 28.14$
Overall heat transfer coefficient	$V_c = \frac{7.43 (100)}{7.43 + 100}$ $= 6.92$	$V_c = \frac{28.14 (100)}{28.14 + 100}$ $= 22.0$
Total Surface Area Required	$A_T = \frac{q_c}{V_c (LMTD)}$ $= \frac{23.4 \times 10^6}{6.92 (731)}$ $= 4626 \text{ Ft}^2$	$A_T = \frac{q_c - q_{SL}}{V_c (LMTD)}$ $= \frac{(23.4-0.75) \times 10^6}{22.0 (731)}$ $= 1408 \text{ Ft}^2$

	<u>WIMPRESS</u>	<u>SCHWEPPE & TORRIJOS</u>
No. of Rows	$N = \frac{4626}{272}$ $= 17.007$ Use 17	$N = \frac{1408}{272}$ $= 5.176$ Use 6

The Schweppe and Torrijos article, when compared with Wimpres's article, includes radiant losses through the shield tubes to the convection section. Wimpres ignores these losses which should be considered. Nevertheless, Schweppe and Torrijos are in error when they approach the convection section.

In both articles, the radiant section is assumed to be a well stirred reactor. Each article employs a heat balance around the radiant section. However, the radiant heat balance does not include the radiant losses through the shield tubes to the convection section. With respect to Schweppe and Torrijos, although they do consider these radiant losses, they never correct the flue gas temperature to the convection section to account for these losses.

Schweppe and Torrijos determine the flue gas temperature leaving the radiant section irrespective of the radiant losses through the shield tubes. They then determine the quantity of heat absorbed in the convection section due to these radiant losses. They proceed to calculate the tube surface required in the convection section by convection without adjusting the flue gas temperature. Although the convective heat absorption is reduced by the radiant leakage, the enthalpy change due to the radiant leakage has not been considered in the flue gas. The result is a flue gas temperature entering the convection section higher than need be. This higher temperature defies an enthalpy balance and produces a calculated convection section surface area lower than required; it is necessary to subtract the radiant shield losses from the sensible heat of the flue gas and determine a new flue gas temperature to the convection section.

Can we theoretically justify this correction? No justification is necessary in the convection section but our radiant section calculations are now subject to question. One might ask the question, "Since there are radiant losses through the shield tubes, must not the radiant firebox temperature be reduced"?

The answer is again "no". The radiant section calculations employ a series of estimations and assumptions (re: the assumption that the radiant section is a well mixed reactor) to best place, in empirical terms, a description of the behavior in the radiant section. Since this method has been demonstrated to be acceptably accurate, there is no need for alteration.

Recommended Method

Let the 2 bottom convection rows be bare and the remaining rows shall be finned as in 2).

A) Calculate the radiation leakage through the shield tubes

$$(1 - \alpha) A_{cp} F = (1 - 0.76)(154)(0.635) = 25.4 \text{ ft}^2$$

assuming TWT = 483 + 100 = 583°F, from Figure 6, Scheweppe and Torrijos:

$$\frac{q_{SL}}{(1 - \alpha) A_{cp} F} = 29600 \frac{\text{BTU}}{\text{ft}^2 \text{hr}}$$

$$q_{SL} = 29600 (25.4) = 0.75 \times 10^6 \text{ BTU/HR}$$

B) Determine the flue gas temperature to the first row of bare tubes (correcting the flue gas enthalpy for the loss of radiant heat through the shield tubes).

Heat content of flue gas @ 1610°F + 25% excess air
Net heat release

$$= 0.45 \text{ (Wimpress, Figure 7)}$$

Heat content of flue gas to the first row of bare tubes
Net heat release

$$= 0.45 - \frac{0.75 \times 10^6}{97.5 \times 10^6} = 0.45 + \frac{q_{SL}}{q_n}$$

$$0.44$$

From Wimpress, Figure 6, ordinate = 0.44

$$T = 1585^\circ\text{F}$$

G) Determine convective heat transfer to the 2 bare rows.

$$\begin{aligned}\text{Gross width} &= 6.5 (8) = 52 \\ \text{Projected width} &= 6 (4.5) = 27 \\ \text{Free width} &= \frac{27}{25''} = 2.08'\end{aligned}$$

$$\text{Mass velocity} = \frac{98640}{3600(38.5)(2.08)} = 0.342 \frac{\text{lb}}{\text{ft}^2\text{sec}}$$

$$\text{Surface} - A_t = \frac{\pi(4.5)(38.5)(6)}{12} = 272 \text{ ft}^2/\text{row}$$

1) Assume convective heat transfer = 6×10^6 BTU/HR

a) Calculate flue gas T leaving 2nd bare row:

$$\frac{\text{Heat content of flue gas}}{\text{Net heat release}} = 0.45 - \frac{(0.75 + 6.00) \times 10^6}{97.5 \times 10^6} = 0.38$$

$$\text{flue gas T out} = 1398^\circ$$

b) Calculate fluid temperature in

$$T_{fl.in} = 483 - \frac{(6 + 0.75)(483 - 380)}{23.4} = 453^\circ\text{F}$$

c) Temperature Difference:

$$\begin{array}{rcl} \text{hot} & = & 1585 \\ & - & 483 \\ \hline & & 1102 \end{array} \quad \begin{array}{rcl} \text{cold} & = & 1398 \\ & - & 453 \\ \hline & & 945 \end{array} \quad \text{LMTD} = \frac{1102 - 945}{\ln \left(\frac{1102}{945} \right)} = 1021.5$$

d) Average Temperatures

$$\begin{aligned}\text{Oil} &= (483 + 453)/2 = 468 \\ \text{Tube Wall} &= 468 + 100 = 568 \\ \text{Gas} &= 468 + 1022 = 1490 \\ \text{Gas Film} &= 468 + 1022/2 = 979\end{aligned}$$

e) Calculate surface required

$$h_{cc} = \frac{2.14 (979 + 460) 0.28 (0.342) 0.6}{4.50.4} = 4.27$$

$$h_{cw} = 2.93 \text{ From Figure 10, Wimpres (Fig. 4, S + T)}$$

$$h_{cw} = 9.46 \left[\frac{(568 + 460)}{1000} \right]^3 = 10.28$$

$$\text{Wall Area} = 2 \left(\frac{8}{12} \right) (0.866)(38.5) = 44.45 \text{ ft}^2$$

$$\beta = \frac{10.28}{4.72 + 10.28 + 2.93} \left(\frac{44.45}{272} \right) = 0.0984$$

$$h_c = (1 + 0.0984)(4.72 + 2.93) = 8.40$$

$$V_c = \frac{8.40(100)}{8.40 + 100} = 7.75$$

$$\text{Surface Area Required} = \frac{6 \times 10^6}{7.75(1021.5)} = 758$$

$$\text{No. of rows} = \frac{758}{272} = 2.79 \quad - \quad \underline{\text{Too High}}$$

Lower estimate of heat absorption

2) Assume convective heat transfer = 4.45×10^6 BTU/HR

a) Calculate Flue Gas T leaving 2nd bare row:

$$\frac{\text{Heat Content of Flue Gas}}{\text{Net Heat Release}} = 0.45 - \frac{(4.45 + 0.75) \times 10^6}{97.5 \times 10^6} = 0.40$$

$$\text{flue gas T out} = 1447^\circ$$

b) Calculate Fluid Temperature in

$$T_{f1.in} = 483 - \frac{(4.45 + .75)(483 - 380)}{23.4} = 460$$

c) Temperature Differences

$$\begin{array}{rcl} \text{hot} & = & 1585 \\ & - & 483 \\ & \hline & 1102 \end{array} \quad \begin{array}{rcl} \text{cold} & = & 1445 \\ & - & 460 \\ & \hline & 985 \end{array} \quad \text{LMTD} = \frac{1102 - 985}{\ln \frac{1102}{985}} = 1042$$

d) Average Temperatures

$$\begin{array}{rcl} \text{Oil} & = & (460 + 483)/2 = 472 \\ \text{Tube Wall} & = & 472 + 100 = 572 \\ \text{Gas} & = & 472 + 1042 = 1514 \\ \text{Gas Film} & = & 472 + 1042/2 = 993 \end{array}$$

e) Calculate Surface Required

$$h_{cc} = \frac{2.14 (993 + 460)^{0.28} (.342)^{0.6}}{4.5^{0.4}} = 4.73$$

$$h_{cr} = 3.05 \text{ (Fig. 10, Wimpres or Fig. 4, S\&T)}$$

$$h_{cw} = 9.46 \left[\frac{(572 + 460)}{1000} \right]^3 = 10.40$$

$$\beta = \frac{10.40}{4.73 + 10.40 + 3.05} \left(\frac{44.45}{272} \right) = 0.0935$$

$$h_c = (1 + 0.0935)(4.73 + 3.05) = 8.51$$

$$V_c = \frac{8.51(100)}{8.51 + 100} = 7.84$$

$$\text{Surface Area Required} = \frac{4.45 \times 10^6}{7.84(1042)} = 544.7$$

D) Determine the remaining number of finned rows required using 1" high, 0.05" thick carbon steel fins, 2 fins per inch.

$$\begin{aligned} \text{Gross Width} &= 6.5(8) = 52'' \\ \text{Projected Width} &= 6 \left[4.5 + 2(0.05)(2) \right] = \frac{28.2''}{23.8''} = 1.98' \end{aligned}$$

$$\text{Mass Velocity} = \frac{98640}{3600(38.5)(1.98)} = 0.359 \frac{\text{lb}}{\text{ft}^2 \text{sec}}$$

$$\begin{aligned} \text{Surface of Fins} &= \frac{\pi}{4} \left[\frac{6.5^2 - 4.5^2}{144} \right] (2) (2) (12) (38.5) (6) \\ &= 1390 \text{ ft}^2/\text{row} \end{aligned}$$

Temperature Differences:

$$\begin{array}{rcl} \text{hot} & = & 1447 \\ & - & 460 \\ & \hline & 987 \end{array} \quad \begin{array}{rcl} \text{cold} & = & 820 \\ & - & 380 \\ & \hline & 440 \end{array} \quad \text{LMTD} = \frac{987 - 440}{\ln \frac{987}{440}} = 677$$

Average Temperatures:

$$\begin{array}{rcl} \text{Oil} & = & (460 + 380)/2 = 420 \\ \text{Tube Wall} & = & 420 + 100 = 520 \\ & = & 420 + 677 = 1097 \\ & = & 420 + 677/2 = 759 \end{array}$$

$$h_{cc} = \frac{2.14(759 + 460)^{0.28}(0.359)^{0.6}}{4.5^{0.4}} = 4.64$$

fin efficiency

$$x \sqrt{\frac{2h}{KY}} = \frac{1}{12} \sqrt{\frac{2(4.64)}{31(0.05/12)}} = 0.705$$

$$\frac{d_f}{d_o} = \frac{6.5}{4.5} = 1.445$$

from Figure 2, Schweppe + Torrijos

$$\text{Efficiency} = 0.84$$

$$h_c = 4.64 \left[\frac{0.84(1390) + 272}{272} \right] = 24.56$$

$$h_{cr} = 2.07 \quad (\text{Fig. 10, Wimpres or Fig. 4, S\&T})$$

$$h_{cw} = 9.46 \left[\frac{(520 + 460)}{100} \right]^3 = 8.90$$

$$\beta = \frac{8.90}{24.56 + 8.90 + 2.07} \left(\frac{44.45}{272} \right) = 0.0409$$

$$h_c = (1 + 0.0409)(24.56 + 2.07) = 27.72$$

$$v_c = \frac{27.72(100)}{27.72 + 100} = 21.70$$

$$\text{Surface Area Required, } A_t = \frac{23.4 - \left[(4.45 + 0.75) \right] \times 10^6}{21.70(677)} = 1239$$

$$\text{No. of finned rows} = \frac{1239}{272} = 4.55 \quad \text{Use 5 finned rows}$$

E. Stack Design

1. General

In a natural-draft heater, the stack must create sufficient draft to overcome frictional drop through the burners, the convection section, the damper, and the stack.

2. Burners

Pressure drop across the burners is often required to draw in secondary air. Its magnitude is usually set by the burner manufacturer, and depends on the type of burner and the fuel. A normal value is about 0.25 inch of water.

3. Convection Section

The pressure drop across bare horizontal tubes in the convection section, is about one-half velocity-head per row. The velocity-head in inches of water is given by the following.

$$P_v = \left(\frac{v^2}{2g} \right) \left(\frac{\rho_g}{\rho_w} \right) (12) = 0.0030 v^2 \left(\frac{\rho_g}{\rho_w} \right) = .003 \frac{v^2}{g} \quad (16)$$

v = Velocity, ft/sec.

g = Gravitational constant, 32.17 ft/sec²

ρ_g = Density of flue gas, lbs/cu.ft.

ρ_w = Density of water, lbs/cu.ft.

Flue gas densities are shown in Figure 33. In the convection section, the gas density is evaluated at the same average gas temperature that was used to calculate the convection section transfer coefficient.

4. Stack and Damper

The various remaining losses are expressed in terms of velocity-head in the stack.

Stack entrance	0.5 velocity-heads	H, Height, Ft.
Damper	1.5	D, Diameter, Ft.
Stack friction	H/50D	
Stack exit	1.0	
TOTAL	3.0 plus H/50D	

Equation 16 and Figure 33 are used to calculate the velocity-head, just as for the convection section.

5. Draft

The draft effect depends on the difference in density between the hot flue gas and the surrounding air.

$$\Delta P = H \left(\frac{P_a - P_g}{P_w} \right) \quad (12) = 0.187H(P_a - P_g) \quad (17)$$

Available draft per 100 feet of height is shown in Figure 34. To allow for heat loss from the stack, its draft should be computed on the basis of a temperature 100°F lower than that of the flue gas leaving the convection section.

6. Furnace Pressure

It is usually desirable for all sections of the heater to be below atmospheric pressure. Then any leaks through peep-holes, tube-openings, or cracks in the setting will be outside air moving in, rather than hot flue gas moving out. In heaters with tall radiant sections, such as vertical cylindrical types, the firebox alone may develop more draft than is required by the burners. In such cases, the stack height must be increased as necessary to maintain sub-atmospheric pressure at the convection section inlet.

7. Stack Size

In many installations, minimum stack heights are set by fume dispersal requirements, particularly if there are tall structures nearby. If such is the case, the stack is sized so that the

total friction drop is no greater than the available draft. If there are no such outside limitations, the most economical combination of diameter and height can be selected. Usual designs have flue gas velocities in the stack of about 25 - 30 feet per second.

F. Furnace Rating Procedure

1. General

We now come to the problem of applying the principles outlined above to a specific furnace design. The following paragraphs describe general concepts that are normally used, and present the outline of a procedure that has been found to require a minimum of trial and error computation.

2. Basic Design Considerations

The function of a process furnace is to heat a prescribed amount of some material from one temperature to another, using a particular fuel. In addition to this requirement, other limitations are generally imposed. Some are needed to fit additional process requirements, and others are set on the basis of past experience. The usual restrictions are discussed in the following paragraphs.

Over-all furnace efficiency determines the basic operating cost of the heater. The optimum efficiency for a particular application depends on a number of factors, such as the cost of fuel, the temperature of the fluid being heated, the type of tube material required, and the user's policy on investment payout. For process heaters without air preheat, over-all efficiencies are usually in the 70 to 80 percent range, based on net heating value.

Table II shows furnace efficiencies based on fuel oil and exit flue gas temperatures. Each fuel will have its

own efficiency chart, however, variations are not extreme and this chart is sufficient for fuel oil firing. Another general curve for estimating efficiency for design or analysis is shown as Figure 35. Each fuel has its own characteristics and to be truly correct a curve should be developed for your fuel.

Excess combustion air has a significant effect on efficiency. In general, the lower the excess air, the smaller the heater for a specific job. However, too little excess air can cause operating difficulties from incomplete combustion, poor flame characteristics, and erratic control. Usual excess air rates are 15 to 25 percent for gas-firing and 25 to 40 percent for oil.

The maximum absorption rate in the radiant section is usually specified by the user on the basis of past experience. High radiant rates generally result in smaller, less expensive heaters. Lower rates give lower tube wall and refractory temperature, thus tending toward lower maintenance. Normal design figures for clean service with good intube transfer coefficients are in the range of 10,000 to 15,000 Btu per hour per square foot of tube surface.

Maximum heat release per unit firebox volume may be specified. Unit heat release is a rough measure of the probability of flame impingement on the tubes. However, current practice is to judge the possibility of flame

impingement from the furnace shape and the type of burner, rather than to set an arbitrary heat release limit.

Maximum pressure drop of the process fluid is usually dictated by the user. This places some restrictions on the tube size and number of parallel passes.

3. Preliminary Design Steps

Before starting the detailed rating, certain aspects of the design can be established from the process requirements and the other limitations described above. First, calculate the net heat release, using the specified absorption duty and efficiency.

Then select an excess air percentage and determine the flue gas rate from Figure 27 or 26c & d.

Normal heaters will handle about 70 percent of the total duty in the radiant section. On this basis, estimate the radiant heat absorption q_R . From enthalpy or specific heat data for the process fluid, estimate the temperature at the crossover between the convection and radiant sections, and the average fluid temperature in the radiant section. Add 100°F to the average fluid temperature to get the average tube wall temperature. Using the specified allowable radiant transfer rate, calculate the total radiant tube surface.

The convection bare tube surface is usually about equal to that in the radiant section. On this basis, select a tube size and pass arrangement that will give the required total surface and meet the specified pressure drop limitation.

4. Radiant Section Rating

Choose a center-to-center spacing for the radiant tubes which is compatible with the selected tube size. Wide tube spacing permits high radiant absorption rates with relatively low firebox temperatures, and gives good circumferential heat distribution. Closer spacing permits more tubes to be installed in a given size firebox. The usual spacing is about twice the nominal pipe size corresponding to the outside tube diameter, for example, 8-inch spacing for 4.5-inch OD tubes.

Using the approximate radiant tube surface determined above, choose firebox dimensions that will accommodate the required total tube length. The exact proportions depend on judgment and past experience. Long furnaces minimize the number of return bends required, thus decreasing total cost. Shorter and wider fireboxes, on the other hand, usually give more uniform heat distribution and lessen the probability of flame impingement on the tube surface.

For the selected firebox dimensions, calculate the equivalent cold-plane surface, the effective refractory surface, and the mean beam length. Read the partial pressure of carbon dioxide plus water from Figure 22 and calculate the PL product.

If preheated air or fuel is used, calculate the heat content of each stream above 60°F, and its ratio to the total net heat release. The heat content of any of these streams below 100°F can usually be neglected. Estimate the heat loss through the setting.

Assume a value for the average temperature of the flue gas in the radiant section, t_g . At this temperature, read the gas

emissivity from Figure 23c & d, and then the exchanger from Figure 24. Calculate the ratio $q_n / \dot{Q} A_{cp} F$.

If the furnace design is one for which experience has shown a significant difference between average effective gas temperature and exit temperature, apply the appropriate correction to determine the exit temperature t_{g2} . Read the corresponding value of q_{g2}/q_n from Figure 37. Then, using Equation 10, calculate $q_R / \dot{Q} A_{cp} F$ and plot it against average furnace temperature on Figure 28.

If the assumed value of t_g was correct, the computed point will be on the absorption curve for the average tube wall temperature. Usually, of course, there will be a discrepancy. Then select another value of t_g on the other side of the absorption curve, and repeat the above procedure. When two points on opposite sides of the curve are obtained, join them by a straight line. The point of intersection of this line with the absorption curve is the correct firebox temperature. For this temperature, calculate t_{g2} , read q_{g2}/q_n from Figure 37, and calculate q_R from Equation 9. This is the heat actually absorbed in the radiant section.

It is now necessary to check the computed radiant absorption to be sure it meets the design limitations. First, divide the heat absorption by the total exposed tube area to get the average heat flux. If it is higher than the allowable maximum, a new furnace with more tube surface must be selected, and the rating repeated. If the actual heat flux is considerably below the allowable value, a smaller furnace should be considered. Then, from enthalpy data on the process fluid, calculate the crossover

temperature from convection to radiant section. If the computed temperature is significantly different from that assumed in starting the rating, it may be necessary to go back and repeat with a new value of average tube wall temperature. Tube wall temperature, however, has such a small effect on the radiant absorption rate that such refinement is usually unnecessary.

There is one point to watch in regard to the cross-over temperature. The fluid temperature determined as described above corresponds to the transition from true convection tubes to shield tubes. Many heaters include the shield tubes as part of the convection section, and the crossover piping is actually between the shield and radiant tubes. For such cases, the temperature in the crossover piping will be higher than that calculated in the rating, by an amount corresponding to heat absorption in the shield tubes.

Convection Section Rating

The convection section must pick up the difference between the heat absorbed in the radiant section and the total heater duty. The temperatures of the flue gas entering the convection section and the process fluid leaving it have already been calculated as part of the radiant section rating. The incoming fluid temperature is set by the original design conditions. The flue gas temperature going to the stack is determined by an over-all heat balance and Figure 37.

$$\frac{q_s}{q_n} = \frac{q_a}{q_n} + \frac{q_f}{q_n} - \left(\frac{q_R + q_c}{q_n} \right) \quad (16)$$

$$\frac{q_s}{q_n} = 1 + \frac{q_a}{q_n} + \frac{q_f}{q_n} + \left(\frac{q_R + q_c}{q_n} \right)$$

Using these temperatures, calculate the average fluid temperature, the log mean temperature difference, the average gas temperature, and the average gas-film temperature.

Select a convection tube arrangement that will give a maximum flue gas mass-velocity of about 0.3 to 0.4 pounds per square foot per second. Calculate the mass-velocity G . For precise ratings, calculate the gas-film coefficient using Figures 29, 30 and 31, and Equations 13 and 14. Calculate the intube coefficient using any applicable method. Then calculate the over-all transfer coefficient from Equation 15. For less critical ratings, read the over-all design coefficient from Figure 32. In either case, finally determine the convection area from the usual heat-transfer equation

$$A_c = \frac{q_c}{U_c (\text{LMTD})} \quad (19)$$

The calculated convection surface will usually not correspond to an integral number of tube rows. In many cases, the computed surface is simply rounded up to an even number of tube rows, and the excess considered as a margin of safety. If a more exact rating is desired, however, it is necessary to assume a new over-all efficiency and repeat the entire rating. The results can then be interpolated to the desired number of tubes.

Finally, calculate the flue gas pressure drop through the convection section, using Equation 16. Evaluate the flue gas density at the same average temperature that was used in calculating the transfer coefficients.

6. Stack Design

The stack must provide enough draft to hold the firebox at a negative pressure, and to overcome frictional losses through the convection section and stack. First, calculate the draft caused by the firebox itself. Determine whether the limiting factor is pressure drop across the burners or the necessity of maintaining negative pressure throughout the firebox.

Assume a stack temperature 100°F lower than the flue gas leaving the convection section. Select a stack diameter to give a stack velocity of about 25 to 30 feet per second, and calculate the velocity head. Estimate the stack height, and calculate the friction loss through it, allowing 1.5 velocity heads for the inlet and exit losses, 1.5 velocity heads for the damper, and 1 velocity head for each 50 diameters of stack height.

Read the stack draft per 100 feet from Figure 34. Calculate the draft that the stack must produce, and the required height. If additional height is required for fume dispersal, the stack diameter may be reduced.

TABLE 3 - Altitude Correction Factors for Gas-Density and Draft

<u>Altitude Above Sea Level</u>	<u>Multiple Density and Draft By</u>
0	1.000
1000	0.964
2000	0.930
3000	0.896
4000	0.863
5000	0.832
6000	0.801
7000	0.772
8000	0.743

VI. SAMPLE CALCULATIONS

To better understand the foregoing discussion, we will do a sample problem. I have chosen that which was developed in your Crude Unit process heat and weight balance. The data from the heat and weight balance is as shown on Figure 38a and the flash curve as Figure 38b. There really isn't much information in the attached data so that we will generate data to try to match or come close to that given. A UOP 9.6 profile was used and we obtained the following curves, Figure 40a and Figure 40b.

Check Ht. Content

Assume 180 psig pressure drop

at outlet

$$T_o = 685^{\circ}\text{F} \quad P_o = 15 \text{ psig} \quad H_o = 438 \text{ Btu/Lb.}$$

at inlet

$$T_i = 490^{\circ}\text{F} \quad P_i = 195 \text{ psig} \quad H_i = 245 \text{ Btu/Lb.}$$

H Btu/Lb

193 Btu/Lb.

$$\text{Duty} = 1.240.000 \times 193 = 239.32 \times 10^6 \text{ Btu/Hr.}$$

237.92 (Figure 38a)

Vaporization at outlet = 68%

Process Specs $\epsilon = 55$

We will use the higher vapor point as it will be safer from pressure drop considerations.

We have decided that we would use the heater structure similar to that shown on Figure 42. It also is necessary to decide on a radiant heat transfer rate and as a guide we will use the number from Table 4 Figure 43 for Crude service of 12,000 Btu/Hr/Sq.Ft.

We will burn fuel oil so that we must establish an efficiency on the basis of the inlet fluid temperature. From Figure 39 or Figures 35 or 36 with a bare tube convection bank we obtain the following assuming a 300°F approach temperature between the exit flue gas and the inlet fluid temperature with 30% excess air.

$$\text{Gross Heater Efficiency} = 71.25\%$$

$$\text{Net Heater Efficiency} = 76.25\%$$

$$\text{Net Heat Release} = \frac{239.32 \times 10^6}{.7625} = 313.86 \times 10^6 \text{ Btu/Hr}$$

From Figure 27 there are 1,080 Lbs. flue gas/MM Btu Net Release

Actually, rather than using a general chart, a chart similar to that shown as Figure 41 should be used for your specific fuel. It is developed quite easily and it should be done. We will use the general correlation of Figure 27.

$$\text{Total Flue Gas} =$$

$$1080 \times 313.86 = 338,971 \text{ Lbs/Hr}$$

$$94.2 \text{ Lbs/Sec.}$$

Radiant Section

Assume that the radiant section will absorb 70% of the total absorption required.

$$\text{Radiant Duty} = .7 \times 239.32 \times 10^6 = 167.52 \times 10^6 \text{ Btu/Hr.}$$

$$\text{With } 12,000 \text{ Btu/Hr/Sq.Ft., the radiant surface required} = \frac{167.52 \times 10^6}{12,000} \text{ (Table 4)}$$

$$= 13,960 \text{ Sq. Ft. From Table 5, Figure 44 we will select 6" I.P.S.}$$

Schedule 40 thickness, then the number of passes required at approxi-

$$\text{mately 3 ft/sec. cold velocity would be } \frac{99,900 \text{ B/D}}{1,000 \times 8 \text{ passes}} \times .31 = 3.8 \text{ ft/sec.}$$

It would be too complicated to use more passes in this style of box we have chosen. A recommended velocity for this service is about 3.5 ft/sec.

Surface per foot of tube (Table 5) = 1.734 Sq.Ft./Ft.

Length of tubing required = $\frac{13,960}{1.734} = 8051$ ft.

Assume tube length of 60 ft.,

then No. tubes required = $\frac{8051}{60} = 134.2$ tubes

tubes/pass = $\frac{134.2}{8} = 16.8$ say 17 tubes/pass

We normally use 50 equivalent diameters for a return bend (Table 5).

Equivalent length per tube = $60 + 25.3 = 85.3$ ft.

Equivalent length per pass = 1450.1 ft.

$G = \text{Mass velocity} = \frac{1,240,000 \text{ Lbs/Hr}}{3,600 \times 8 \times 0.2008} = 214.4 \text{ lbs/sec. sq.ft.}$

satisfactory - not less than 150

Cross Over Temperature

Now let us estimate the crossover temperature accordingly with the following procedure:

Outlet conditions: 685°F & 15 psig $\longrightarrow H = 438$ Btu/Lb

Crossover conditions:

$$\Delta H_{co} = \frac{\text{Radiation section heat absorbed, Btu/Hr}}{\text{Fluid heated, lb/hr}}$$

$$\Delta H_{co} = \frac{167.52 \text{ MM Btu/Hr}}{1.24 \text{ MM Lb/Hr}} = 135 \text{ Btu/Lb}$$

$$\Delta E_{co} = H_o - H_{co} \qquad \begin{matrix} H_{co} & H_o - \Delta H_{co} \\ H_{cc} & 438 - 135 = 303 \text{ Btu/Lb} \end{matrix}$$

Pressure assumed = 140 psig, so $T_{co} = 554^\circ\text{F}$ (Fig. 40a)

Estimated Pressure Drop

$$\frac{\Delta P}{100'} = \frac{.517 f G^2}{ID \times \rho}$$

$f = 0.004$ for this service

$ID = 6.065$ in. (pipe size ID)

$G = 214.42$ lb/sec. sq.ft.

$\rho = @$ Process Conditions

$$\rho = \frac{1}{S.V} = \frac{1}{1.8} \text{ ft}^3/\text{lb}$$

then:

$$\begin{aligned} \frac{\Delta P}{100'} @ \text{ outlet cond.} &= \frac{(.517)(.004)(214.42)^2}{(6.065) \left(\frac{1}{1.8} \right)} \\ &= (15.6765)(1.8) = 28.22\#/100' \end{aligned}$$

$$\frac{\Delta P}{100'} @ \text{ crossover cond.} = (15.6765)(S.V @ \text{ Cross over cond.})$$

specific volume = .125 @ $P_{co} = 140$ PSIG & $T_{co} = 554^\circ\text{F}$

$$\frac{\Delta P}{100'} @ \text{ crossover cond.} = (15.6765)(.125) = 1.96\#/100'$$

$$LM \Delta P = \frac{28.22 - 1.96}{\ln \frac{28.22}{1.96}} = \frac{26.26}{\ln 14.4} = \frac{26.26}{2.6672} = 9.846\#/100'$$

$$\text{Pressure calculated} = \frac{9.846\#}{100'} \times 1450.1' = 142.8\# \text{ vs. } 140.0\#$$

Accepted deviation: 10 - 20%

T_{co} = cross over temperature = 554°F

$$\text{Average fluid temp.} = \frac{T_{co} + T_o}{2} = \frac{554 + 685}{2} = \frac{1239}{2} = 620^\circ\text{F}$$

Tube Wall Temperature Check

Now let us estimate the tube wall temperature by means of Sieder & Tate equation as follows:

$$h = \frac{K}{D} \left(\frac{CZ}{K} \right)^{1/3} \left(\frac{Z}{Z_w} \right)^{0.14} \cdot f \left(\frac{DG}{Z} \right)$$

h = Film Coeff. Btu/Hr/Sq.Ft./°F

K = Film thermal conduct. Btu/Hr/Sq.Ft./°F/Ft)

L = Heated tube length, ft.

D = Inside tube diameter, inches

Z_w = Absolute viscosity, centipoises at average tube wall temperature

G = Mass velocity, lb/sec/sq.ft.

C = Specific heat of fluid at average fluid temp., Btu/Lb/°F

Z = Absolute viscosity, centipoises at average temperature of fluid

Crude oil properties will be determined at an average temperature of 620°F.

Crude properties: API 34.8 Charact. factor 11.9

$$K = 0.0654$$

$$L = 1450.1 \text{ ft.}$$

$$D = 6 \text{ in. (6.065 I.D.)}$$

$$(Z/Z_w)^{0.14} \approx 1.0$$

$$Z = .3 \text{ Centistokes @ 620°F}$$

$$G = 214.42$$

$$C = 0.76 \times 1.005 = .7638$$

Corrected by UOP K factor accordingly to UOP Chart F-1

$$(Z = Z_{cp}/S.G. \quad , \quad Z_{cp} = (Z)(S.G.) = (.3)(.625)$$

$$S.G. @ 620°F = .625 \text{ from UOP PD-194}$$

$$\text{Modified Reynolds Number} = \frac{DG}{Z} = \frac{(6)(214.42)}{(.1875)} = 6861$$

$$\text{From Fig. 45: } \phi \left(\frac{DG}{Z} \right) = 22,000$$

From Equation 1:

$$h = \frac{(.0654)}{(6)} \left[\frac{(.7638)(.1875)}{(.0654)} \right]^{1/3} (1)^{.14} (22,000) = 239.8 [2.1898]^{1/3}$$

$$H = (239.8)(1.3) = \underline{311.74}$$

The temperature difference through the oil film for a clean tube, is:

$$\Delta T_{\text{film}} = 12000 \left(\frac{6.625}{6.065} \right) \frac{1}{311.74} = 42.0$$

The temperature difference through the tube wall, is:

$$\Delta T_{\text{metal}} = 12000 \left(\frac{0.28}{190} \right) \left(\frac{6.625}{6.625 - .28} \right) = 18.5$$

$$\text{Total } \Delta T = \Delta T_{\text{film}} + \Delta T_{\text{metal}} = 42 + 18.5 = 60.5$$

$$\text{Average Tube Wall Temperature} = \frac{554 + 685}{2} + 60.5 = 680^{\circ}\text{F}$$

Radiation Section Lay-out

The lay-out of the heater is shown on Fig. 46. The heater dimensions are:
34 feet wide by 36 feet high.

Total number of tubes: 136

Tube Long Overall: 60 ft.

Exposed Length: 58.5 ft. (1.5 ft available for support and bank)

Center-Center distance: 1 ft.

Shield Tubes: 8

Number of Passes: 8

External-Internal Tube Wall Ratio: 2-3

External-Internal Tube Wall High Diff.: 1' Minimum

Space between the Wall and the Tube: One Tube Diameter

Burners to tube clearance: 5' Minimum

Permissible temperature difference between leaving streams: 30-40°F

Firebox Temperature Calculation

Surface per tube = 58.5 ft/tube x 1.734 sq. ft/ft = 101.44 sq. ft./tube

Total Surface = 136 tubes x 101.44 sq. ft./tube = 13796 sq.ft. = A_R

Spacing ratio - center-center distance/O.D. = 12/6.625 = 1.8

Effectiveness factor, α = .92 (Fig. 19)

Shield tubes, A_{cps} = exposed length/tube x No. of tubes x center-center distance.

$$= 58.5 \text{ ft/tube} \times 8 \text{ tubes} \times 1 \text{ ft} = 468 \text{ sq. ft.}$$

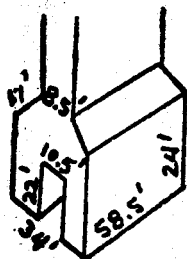
Wall Tubes A_{cpw} = 58.5 ft/tube x 128 tubes x 1 ft = 7488 sq. ft.

Total Wall Tubes = $\alpha A_{cps} + \alpha A_{cpw}$ = Total equivalent cold plane surface

$$(1)(468) + (.92)(7488) = 468 + 6889 = 7357 (\alpha \text{ for shield tubes}$$

is consider unity.)

$$\text{Envelope area} = (2)(24 \times 34) + (2) \left(\frac{34 + 8.5}{2} \right) (11.24) + 2(58.5 \times 17) + 2(58.5 \times 24)$$



$$\begin{aligned} &+ (8.5 \times 58.5) + (58.5 \times 34) + 2(22 \times 58.5) - 2(10.5)(22) \\ &= 1632 + 478 + 1989 + 2808 + 497 + 1989 + 2574 - 462 \\ &= 11505 \text{ sq. ft.} \end{aligned}$$

A_R = Refractory area = Envelope area - Total equivalent cold plane surface

$$A_R = 11505 - 7357 = 4148 \text{ sq. ft.}$$

$$\text{Ratio Refractory area} = \frac{A_R}{\alpha A_{cpt}} = \frac{4148}{7357} = .564$$

Total eq. cold
plane surface

$$\begin{aligned} \text{Furnace Volume} &= 24 \times 34 \times 58.5 + (12 + 8.5) 12 \times 58.5 - 22 \times 10.5 \times 58.5 \\ &= 47736 + 14391 - 13514 = 48613 \text{ cu. ft.} \end{aligned}$$

$$\begin{aligned} L &= \text{Mean beam length} = \frac{2}{3} (\text{furnace volume})^{1/3} \text{ (Table 1, Fig. 22a)} \\ &= \frac{2}{3} (48613)^{1/3} \\ &= \frac{2}{3} (36.365) = 24.24 \text{ ft.} \end{aligned}$$

Partial pressure, $\text{CO}_2 + \text{H}_2\text{O} = .225$ Figure 22 @ 30% excess air

$$P_L = (.225)(24.24) = 5.45 \text{ atm. ft.}$$

Let us assume for the heater operation there is no preheated air or fuel,

$$\text{so } q_a = q_f = 0.$$

$$\text{Total average loss} = q_L/q_n = 0.02$$

Estimated average firebox temperature = 1500°F

Flame emissivity: For $T_G = 1500^\circ\text{F}$ & $P_L = 5.45 \text{ atm. ft.}$

<u>Tube Metal Temp., °F</u>	<u>ϵ_r</u>
600	.555
680	.593
800	.591

By interpolation from Fig. 23a - d

ϕ : Exchange factor = .641 at $P_F = .593$ & $A_R/\alpha A_{cpt} = .564$ (Fig. 24)

$$\alpha A_{cpt} \phi = (7357)(.641) = 4715 \text{ sq. ft.}$$

$$q_n/\alpha A_{cpt} \phi = \frac{313.86 \text{ MM BTU/hr}}{4715 \text{ sq. ft.}} = 66554 \text{ BTU/hr sq. ft.}$$

For this type of furnace, assume that the exit gas temperature is the same as for the average firebox temperature, i.e. like a well-stirred fire box.

Then $t_{g2} = t_g = 1500^\circ\text{F}$

$$q_{g2}/q_n = .43 \quad \text{Fig. 37 @ 30\% E.A.}$$

$$\begin{aligned} \frac{q_R}{\alpha A_{cpt} \phi} &= \left[1 + \frac{q_a}{q_n} + \frac{q_F}{q_n} - \frac{q_{g2}}{q_n} - \frac{q_L}{q_n} \right] \frac{q_n}{\alpha A_{cpt} \phi} \\ &= [1 + 0 + 0 - .43 - 0.02] 66554 \\ &= (.55) (66554) = 36605 \text{ BTU/hr sq. ft.} \end{aligned}$$

If we plot the calculated $q_R/\alpha A_{cpt}$ of 36605 at the assumed firebox temperature of 1500°F , the point lies to the left of the absorption curve (Fig. 25b).

So, let us try another calculation at 1750°F emissivity, by interpolation @ 30% E.A.

$$T_g = 1750^\circ\text{F} \text{ \& } P_L = 5.45 \text{ atm. ft.}$$

Tube metal temp., $^\circ\text{F}$	P_F
600	.575
680	.573
800	.571

Exchange factor, $\phi = .629$ at $P_F = .573$ & $A_R/\alpha A_{cpt} = .564$

$$\alpha A_{cpt} \phi = (.629)(7357) = 4628 \text{ sq. ft.}$$

$$q_n/\alpha A_{cpt} \phi = \frac{313.86 \text{ MM BTU/hr}}{4628 \text{ sq. ft.}} = 67824 \text{ BTU/hr sq. ft.}$$

$$q_{g2}/q_n = .52$$

$$q_R/\alpha A_{cpt} \phi = (1 - 0.02 - .52) 67824 = 31100 \text{ BTU/hr sq. ft.}$$

Exchange factor = .655 at $P_F = .593$ & $A_R/\alpha A_{cpt} = .631$ (Fig. 24)

$$\alpha A_{cpt} \phi = (7357)(.655) = 4819 \text{ sq.ft.}$$

$$q_n/\alpha A_{cpt} \phi = \frac{313.86 \text{ MM Btu/Hr}}{4819 \text{ sq. ft.}} = 65132 \text{ Btu/Hr. sq.ft.}$$

For this type of furnace, assume that the exit gas temperature is the same as for the average firebox temperature, i.e. like a well-stirred fire box.

Then $t_{g2} = t_g = 1500^\circ\text{F}$

$$q_{g2}/q_n = .43 \quad \text{Fig. 37 @ 30\% E.A.}$$

$$\begin{aligned} \frac{q_R}{\alpha A_{cpt} \phi} &= \left[1 + \frac{q_a}{q_n} + \frac{q_F}{q_n} - \frac{q_{g2}}{q_n} - \frac{q_L}{q_n} \right] \frac{q_n}{\alpha A_{cpt} \phi} \\ &= [1 + 0 + 0 - .43 - 0.02] 65132 \\ &= (.55) (65132) = \underline{35823} \text{ Btu/Hr sq.ft.} \end{aligned}$$

If we plot the calculated $q_R/\alpha A_{cpt}$ of 35823 at the assumed firebox temperature of 1500°F , the point lies to the left of the absorption curve (Fig. 25).

So, let us try another calculation at 1750°F emissivity, by interpolation @ 30% E.A.

$$T_g = 1750^\circ\text{F} \text{ \& } P_L = 5.45 \text{ atm. ft.}$$

Tube metal temp., °F	P_F
600	.575
680	.573
800	.571

Exchange factor, $\phi = .635$ at $P_F = .573$ & $A_R/\alpha A_{cpt} = .631$

$$\alpha A_{cpt} \phi = (.635)(7357) = 4672 \text{ sq.ft.}$$

$$q_n/\alpha A_{cpt} \phi = \frac{313.86 \text{ MM Btu/Hr}}{4672 \text{ sq.ft.}} = 67183 \text{ Btu/Hr. sq.ft.}$$

$$q_{g2}/q_n = .52$$

$$q_R/\alpha A_{cpt} \phi = (1 - 0.02 - .52) 67183 = \underline{30908} \text{ Btu/Hr. sq.ft.}$$

Plotting this point on Fig. 25b and joining the two calculated points by a straight line, it crosses the absorption curve at 1588°F. This is the fire-box temperature.

Now, again from Fig. 37.

$$q_{g2}/q_n = .46$$

$$q_R = [1 - 0.02 - .46]313.86 = \underline{163.2 \text{ MM Btu/Hr}}$$

$$q_r/A_{Rt} = \frac{163.2 \text{ MM Btu/Hr}}{13796 \text{ sq.ft.}} \approx 11830 \text{ Btu/Hr. Sq. Ft.}$$

The obtained heat flux is just below the allowable value of 12,000 -- now check the crossover temperature of the oil.

Enthalpy change - $163.2 \times 10^6 \text{ Btu/Hr} - 132 \text{ Btu/\#}$

$$H_{co} = H_o - \Delta H_{co} = 438 - 132 = 306 \text{ Btu/\#}$$

Crossover temperature = 558°F vs. 554°F (assumed).

$$\text{Average radiant fluid temperature} = \frac{558 + 685}{2} = 622^\circ\text{F vs } 624^\circ\text{F}$$

$$\text{Average Tube Wall} = 622 + 60.5 = 682.5^\circ\text{F vs } 680^\circ\text{F}$$

Calculated values are so close to assumptions no further correction is necessary.

Convection Section Design

Heat Absorbed

Convection section duty = heater duty - radiant section duty

$$\text{Convection section duty} = 239.32 - 163.2 = 76.12 \times 10^6 \text{ Btu/Hr}$$

$$\begin{aligned} \frac{\text{Stack Heat Content}}{\text{Release}} &= \frac{q_s}{q_n} = 1 + \frac{q_a}{q_n} + \frac{q_f}{q_n} - \left(\frac{q_R + q_c}{q_n} \right) - \left(\frac{q_L}{q_n} \right) \\ &= 1 + 0 + 0 - 0.763 - 0.02 = 0.217 \end{aligned}$$

@ $\frac{q_s}{q_n} = 0.217$ and 30% excess air,

Stack Temperature = 830°F

(Figure 37)

Radiation Losses Through the Shield Tubes

$$(1 - \alpha) A_{cp} F = (1 - 0.76)(468)(0.648) = 72.8 \text{ ft}^2$$

where $F = 0.648$ is determined by interpolation at our firebox temperature at 1573°F

From Fig. 58 and $TW_{\text{Test}} = 500 + 60 = 560$

$$\frac{q_{SL}}{(1 - \alpha) A_{cp} F} = 28500 \frac{\text{Btu}}{\text{ft}^2 \text{ Hr}}$$

$$q_{SL} = 28500 (72.8) = 2.07 \times 10^6 \text{ Btu/Hr}$$

Determining flue gas temperature to the convection section

$$\frac{\text{Heat Content of Flue Gas}}{\text{Net Heat Release}} @ 1573^\circ\text{F} = 0.46 \quad (\text{Figure 37})$$

$$\begin{aligned} \frac{\text{Heat Content of Flue Gas to Conv. Section}}{\text{Net Heat Release}} &= 0.46 - \frac{2.07 \times 10^6}{313.86 \times 10^6} \\ &= 0.45 \end{aligned}$$

From Figure 37 $T_{\text{fl. gas to conv.}} = 1560^\circ\text{F}$

Temperatures

Temperature Differences:

$$\Delta T_{\text{Hot}} = T_g - T_{co} = 1560 - 558 = 1002^\circ\text{F}$$

$$\Delta T_{\text{Cold}} = T_g - T_i = 830 - 490 = 340^\circ\text{F}$$

$$\begin{aligned}\text{Log Mean Temp} &= \frac{\Delta T_{\text{hot}} - \Delta T_{\text{cold}}}{\ln \frac{\Delta T_{\text{hot}}}{\Delta T_{\text{cold}}}} \\ &= \frac{1002 - 340}{\ln \frac{1002}{340}} \\ &= 640\end{aligned}$$

Average Temperatures:

$$\text{Oil Average Intube Fluid} = \frac{T_{hi} + T_{co}}{2} = \frac{490 + 558}{2} = \frac{1048}{2} = 524^{\circ}\text{F}$$

$$\text{Tube Wall} = \text{Oil Av. Int. Fluid} + 100 = 524 + 100 = 624^{\circ}\text{F}$$

$$\text{Gas} = \text{Oil Av. Int. Fluid} + \text{Log Mean Temp.} = 524 + 640 = 1164^{\circ}\text{F}$$

$$\text{Gas Film} = \text{Oil Av. Int. Fluid} + 1/2 \text{ Log Mean Temp.} = 524 + 1/2 (640) = 844^{\circ}\text{F}$$

Convection Section Lay-out

Let us select a convection section with eight 6.625 in. O.D. tubes per row, 12 inches spacing, staggered.

$$\begin{aligned}\text{Gross width} &= (\text{No. tubes/row} + 1/2 \text{ tube/row}) \times \text{tube spacing, in.} \\ &= (8 + 1/2) \times 12 = 102 \text{ inches}\end{aligned}$$

$$\begin{aligned}\text{Free Width} &= \text{Gross width} - \text{tube wall plane} \\ &= 102 - 8 \times 6 = 54 \text{ inches} = 4.5 \text{ ft.}\end{aligned}$$

$$\begin{aligned}\text{Free width area} &= \text{Free width} \times \text{exposed tube length} \\ &= 4.5 \times 58.5 \\ &= 263.25 \text{ sq. ft.}\end{aligned}$$

$$\begin{aligned}\text{Flue Gas Mass Velocity} = G &= \frac{\text{Total Flue Gas Mass}}{\text{Free Width Area}} \\ &= \frac{94.2 \text{ lb/sec}}{263.25 \text{ sq. ft.}} = .356 \text{ lb/sq.ft.sec.}\end{aligned}$$

(Permissible flue gas mass velocity - .5 lb/sq.ft.sec. maximum)

Film Transfer Correlation

For $G = .356$; 6-5/8" O.D. & Gas Film Temp. = 844°F From Fig. 29

$$\text{or } h_{cc} = \frac{2.14 (T_f)^{0.28} (G_{\max})^{0.6}}{D_i^{0.4}} = \frac{2.14 (844+460)^{0.28} (.356)^{0.6}}{6.625^{0.4}} \\ = 4.03 \text{ Btu/Hr Ft}^2\text{°F}$$

For Av. Gas Temp. = 1164°F & Tube Wall Temp. = 624°F From Fig. 30

$$h_{cr} = 2.40 \text{ Btu/Hr. sq.ft. °F}$$

For Ave. Tube Wall Temp. = 624°F

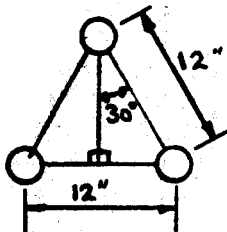
From Fig. 31

$$\text{or } h_{cw} = 9.46 \left[\frac{T_t}{1000} \right]^3 = 9.46 \left[\frac{624 + 460}{1000} \right]^2 = 12.05$$

$$\text{then } h_{cc} + h_{cr} + h_{cw} = 4.03 + 2.40 + 12.05 = 18.48 \text{ Btu/Hr sq.ft. °F}$$

Row-to-Row Tube Spacing

For an equilateral convection section lay out we have:



$$a = \text{Row-to-Row tube spacing} = d \cos 30^\circ \\ = \frac{12}{12} \times .866 \\ = .866 \text{ ft.}$$

$$A_{cw} = \text{tube area per row} = \text{two walls} \times a \times L_e \text{ (exposed length)} \\ = 2 \times .866 \times 58.5 = 101.322 \text{ sq. ft.}$$

$$A_{ct} = \text{tube area per row} = \text{No. tubes/row} \times \text{area/tube} \\ = 8 \times 58.5 \times 1.734 = 811.512 \text{ sq.ft.}$$

Convection Section
Convection Factor

$$= f = \frac{h_{cw}}{h_{cc} + h_{cr} + h_{cw}} \left(\frac{A_{cw}}{A_{ct}} \right) \\ = \frac{12.05}{18.48} \left(\frac{101.322}{811.512} \right) = 0.0817$$

Overall-All Transfer Coefficient

$$\begin{aligned}\text{Total Apparent Gas Film Coefficient} &= h_c = (1 + f) (h_{cc} + h_{cr}) \\ h_c &= (1 + .0814) (4.03 + 2.40) \\ h_c &= 6.95 \text{ Btu/Hr. sq.ft. } ^\circ\text{F}\end{aligned}$$

$$\text{Intube Film Coefficient} = h_c = \frac{K}{D} \left(\frac{CZ}{K} \right)^{1/3} \left(\frac{Z}{Z_w} \right)^{-1.4} \phi \left(\frac{DG}{Z} \right)$$

Crude oil properties will be taken at an average temperature of 474°F

Crude Oil Properties:

API = 34.8	Charact. factor = 11.9
K = .0695	G = 214.42
D = 6 in.	C = .68 x 1.005 = 6834
(Z/Z _w) .14 @ 1.0	Corrected by UOP K
SG. @ 474°F = 0.70 from UOP PD-194	factor according to
Z = .46 Centistokes @ 474°F	UOP Chart F-1

$$Z = Z_{cp}/S.G. \quad \therefore Z_{cp} = Z \times S.G. = (.46)(.7) = .322 \text{ Centipoise}$$

$$\text{Modified Reynolds Number} = DG/Z_{cp} = \frac{(6)(214.42)}{(.322)} \approx 4000$$

$$\phi \left(\frac{DG}{Z} \right) = 14800$$

From Equation (1)

$$h_{ci} = \frac{(.0695)}{(6)} \left[\frac{(.6834)(.322)}{(.0695)} \right]^{1/3} (1)^{-1.4} (14800) = 171.42 [3.1662]^{1/3}$$

$$h_{ci} = (171.43)(1.4678) = 251.62 \text{ Btu/Hr.sq.ft. } ^\circ\text{F}$$

Convection Section
Over-All Transfer
Coefficient

$$= U_c = \frac{h_c (h_{ci})}{h_c + h_{ci}}$$

$$U_c = \frac{(6.95)(251.62)}{6.95 + 251.62}$$

$$U_c = 6.77 \text{ Btu/Hr. Sq.Ft. } ^\circ\text{F}$$

For comparison, the approximate over-all coefficient from Fig. 32 is 6.75
For 6-5/8" O.D. & av. Gas Temp. = 1164°F

Convection Section Area

Convection section area is estimated from the usual heat-transfer equation:

$$A_c = \frac{q_c}{U_c (LMTD)}$$

$$A_c = \frac{(72.12 - 2.07) \times 10^6 \text{ Btu/Hr.}}{\left(\frac{6.77 \text{ Btu}}{\text{Hr.Sq.Ft. } ^\circ\text{F}} \right) (640^\circ\text{F})} = 16176 \text{ Sq. Ft.}$$

$$\text{Number of tubes} = \frac{\text{Convection area, ft}^2}{\text{area/tube}} = \frac{16176}{58.5 \times 1.734} \approx 160 \text{ tubes.}$$

$$\text{Number of rows} = \frac{160 \text{ tubes}}{8 \text{ tubes/row}} = 20 \text{ rows}$$

20 rows are required or a total number of 160 tubes.

As we can see, 20 convection section rows are required to meet our remaining duty requirement. However, if we note the flow through our furnace we can see the heater inlet (convection section inlet) and the convection section outlet is on the same side of the furnace. The convection section outlet immediately feeds the radiant section inlet on the same side. Since the radiant section has 17 tubes per pass, the heater outlet (radiant section outlet) will exit the opposite side of the furnace.

When the furnace inlet and outlet are on opposing sides, there is wasted piping more than one furnace length. To remedy this, we can either add another row of tubes or remove a row of tubes to have the terminals on the same side. An additional row will increase the efficiency minutely since the LMTD of the individual row will be relatively small. Similarly, the removal of one row will decrease the efficiency for the same reason. This increase in efficiency will cause the furnace to fire slightly higher, increasing our average radiant flux minutely.

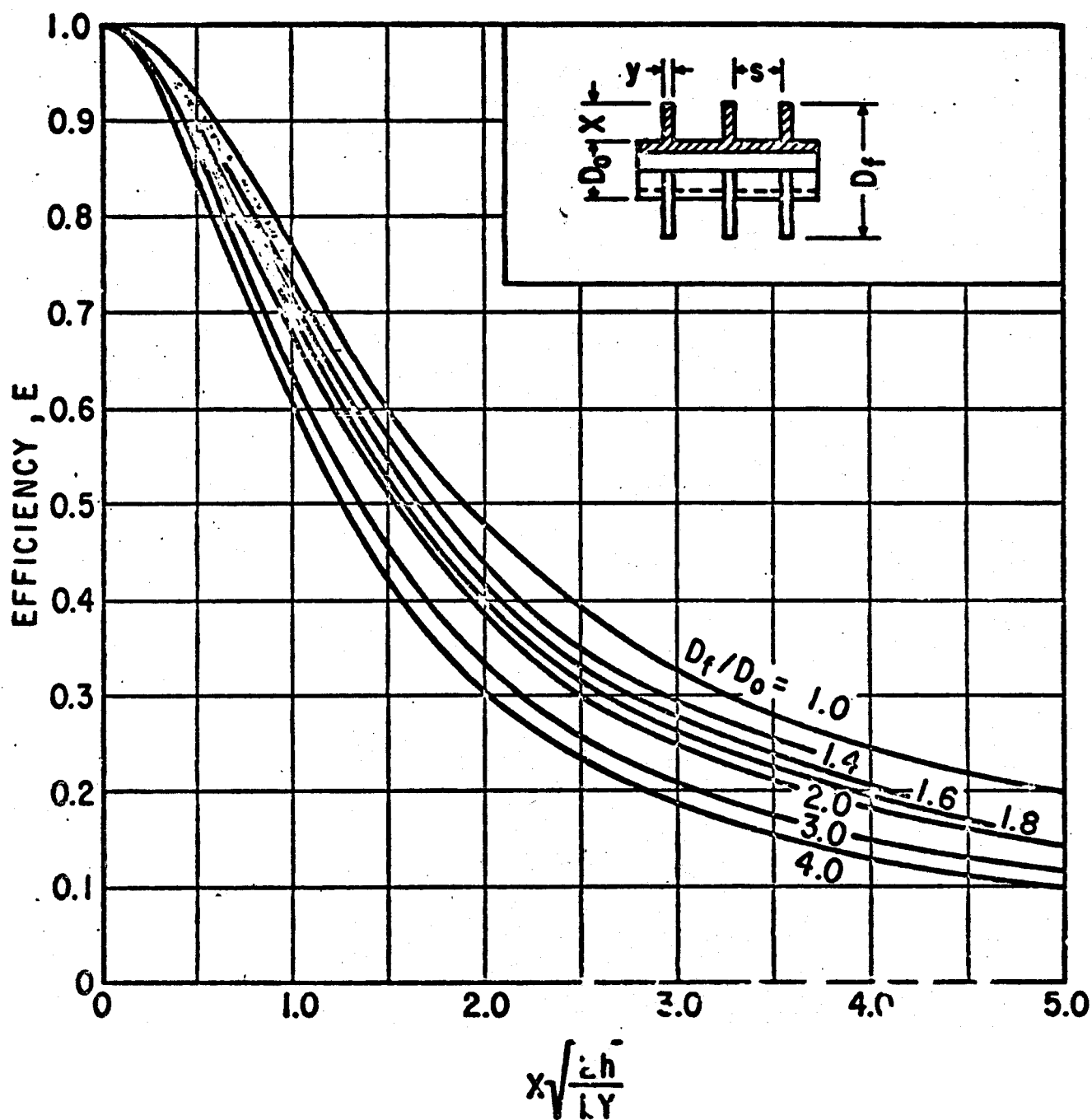


FIGURE 2—Efficiency of annular fins of constant thickness.⁶

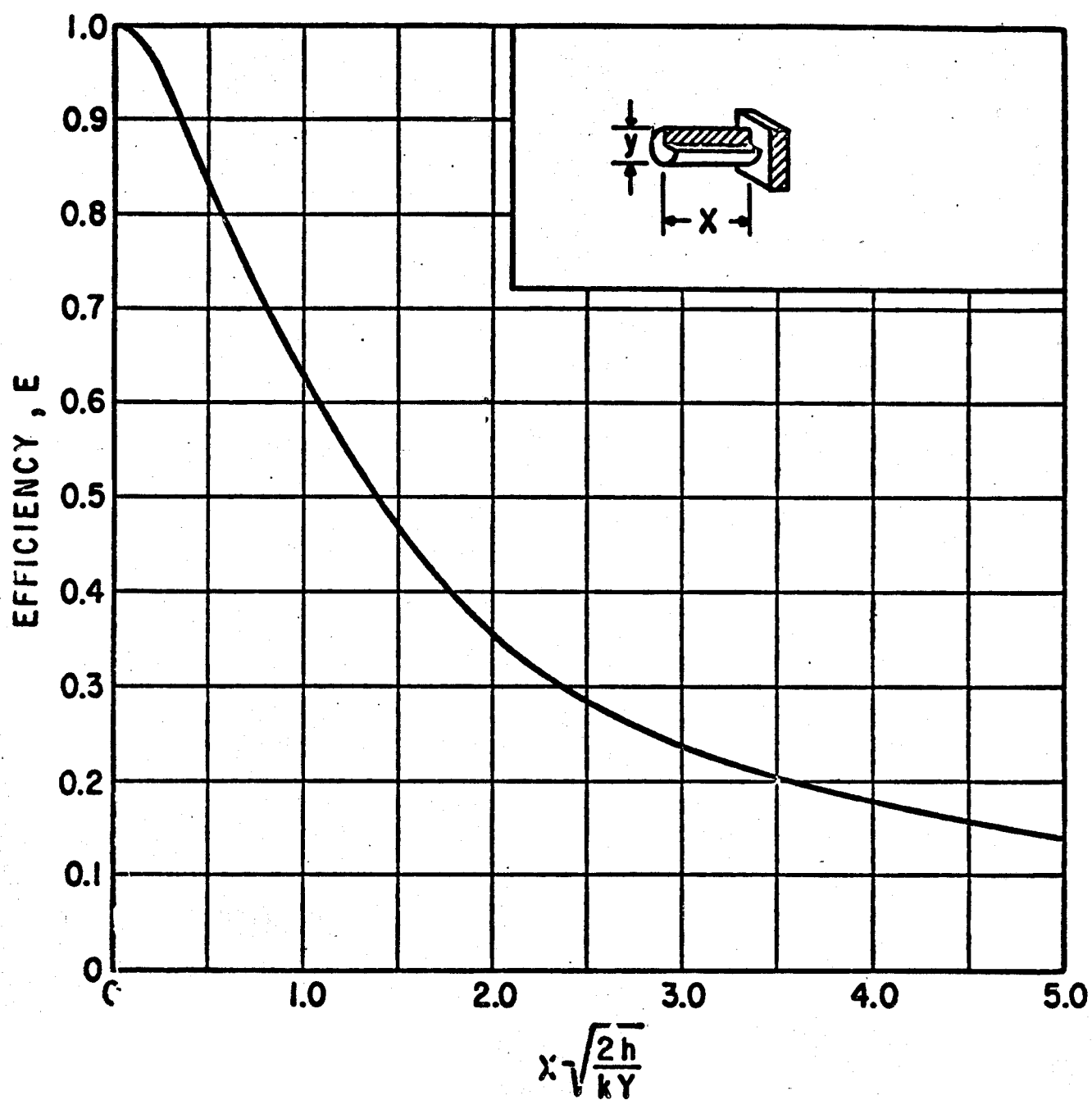


FIGURE 3—Efficiency of spines of constant diameter.⁶

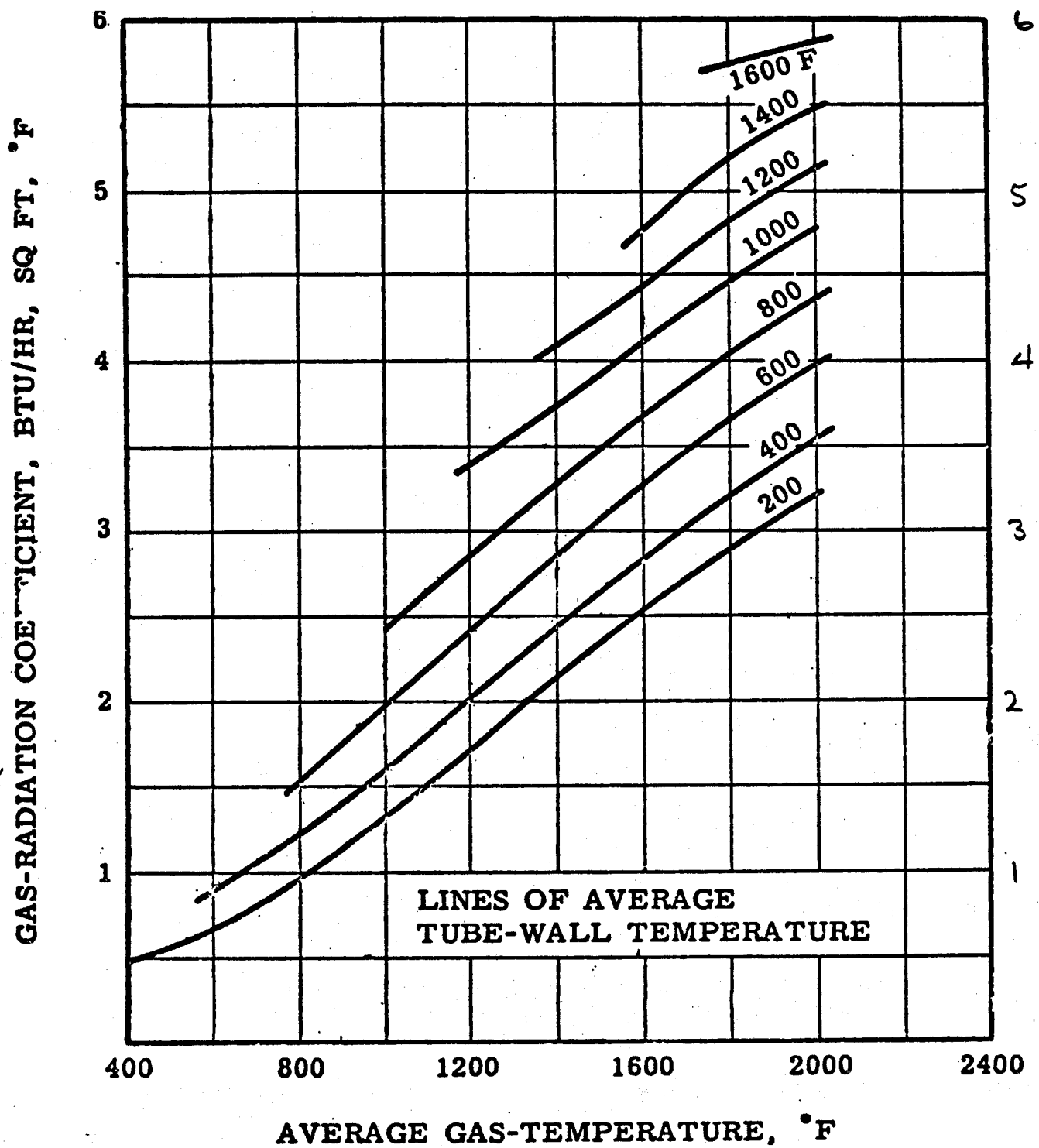


FIGURE 4—Gas radiation coefficients.¹

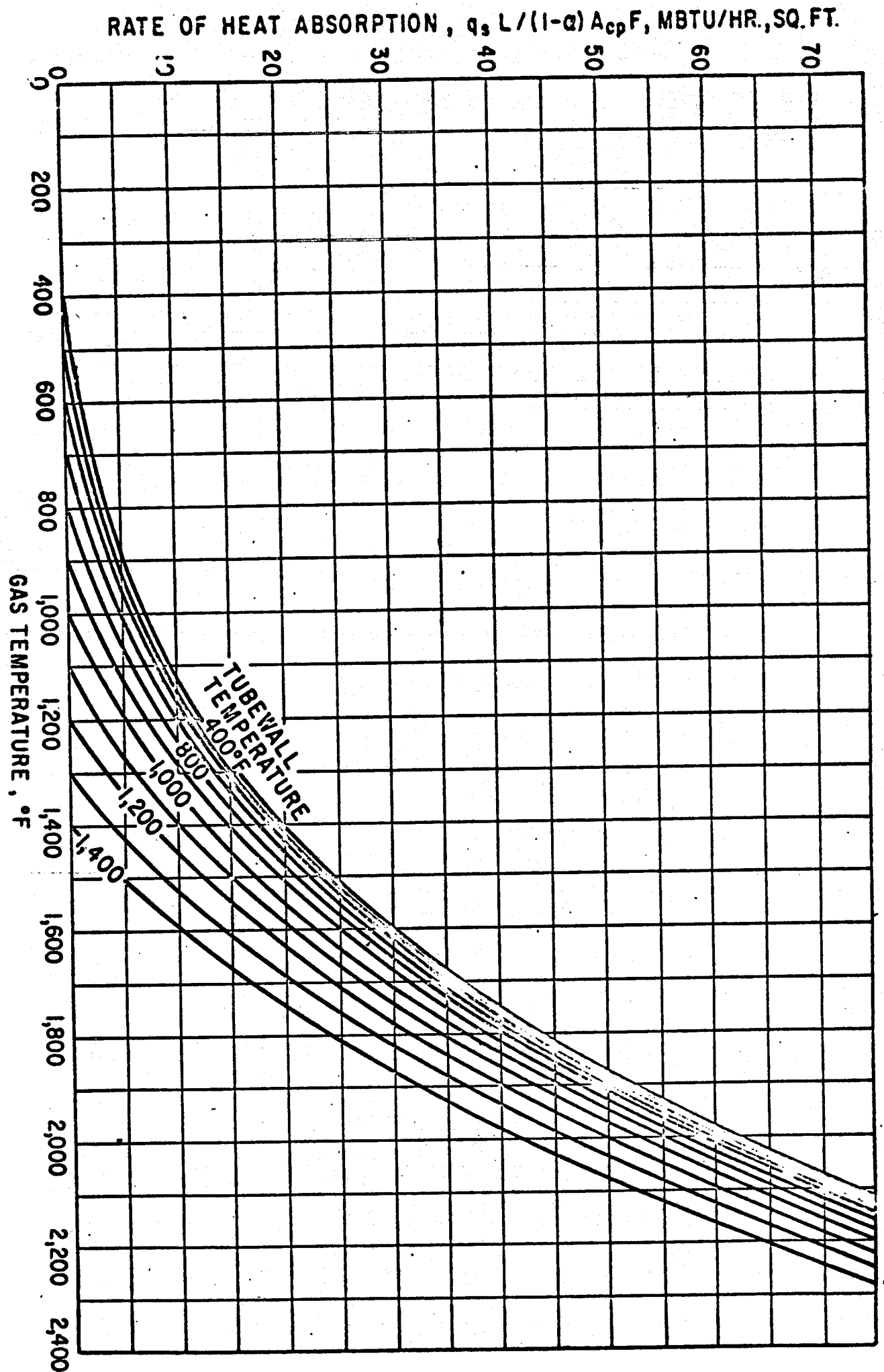


FIGURE 6—Radiation leakage through shield coil.

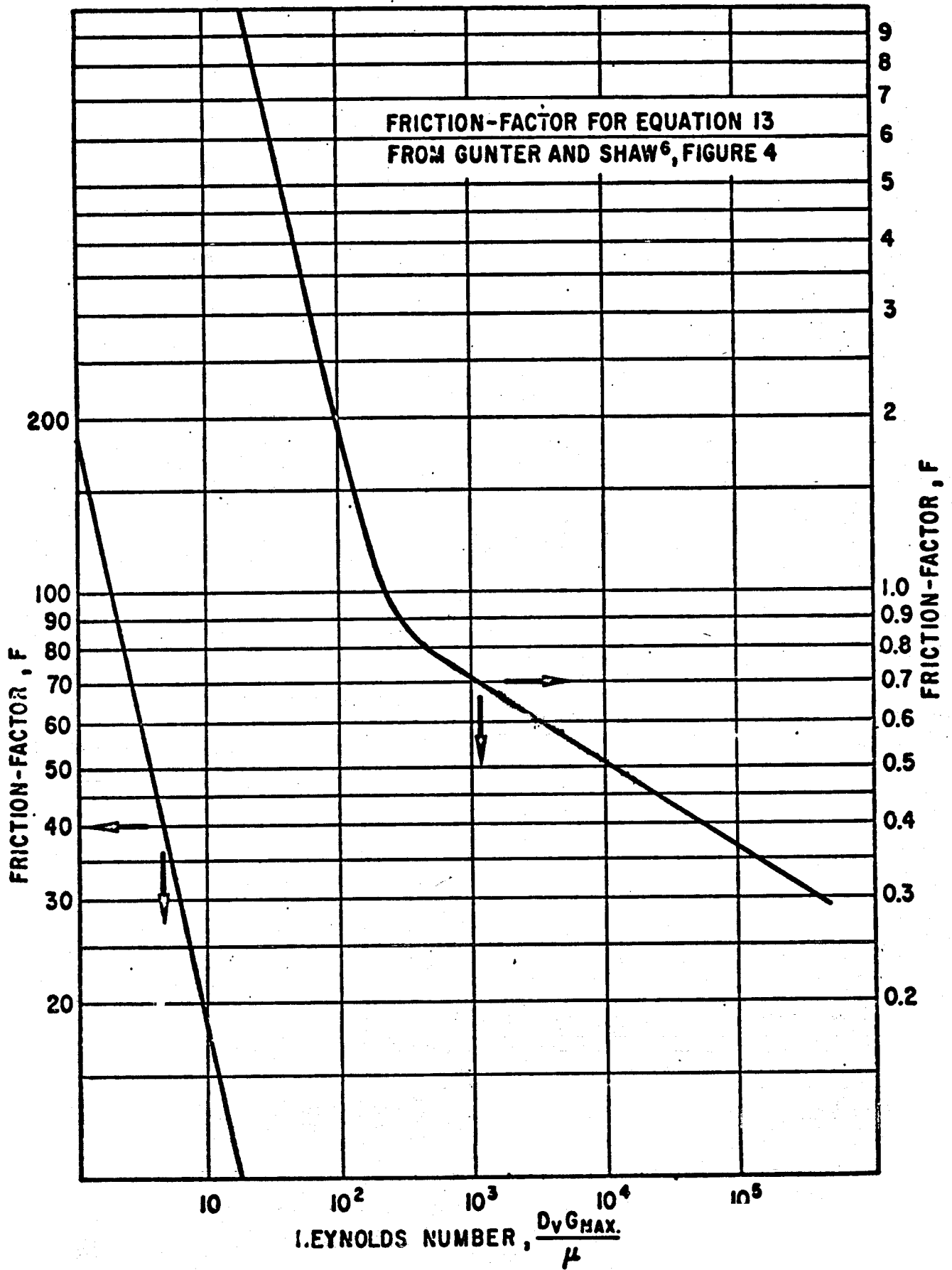


FIGURE 7—Cross flow friction factor.

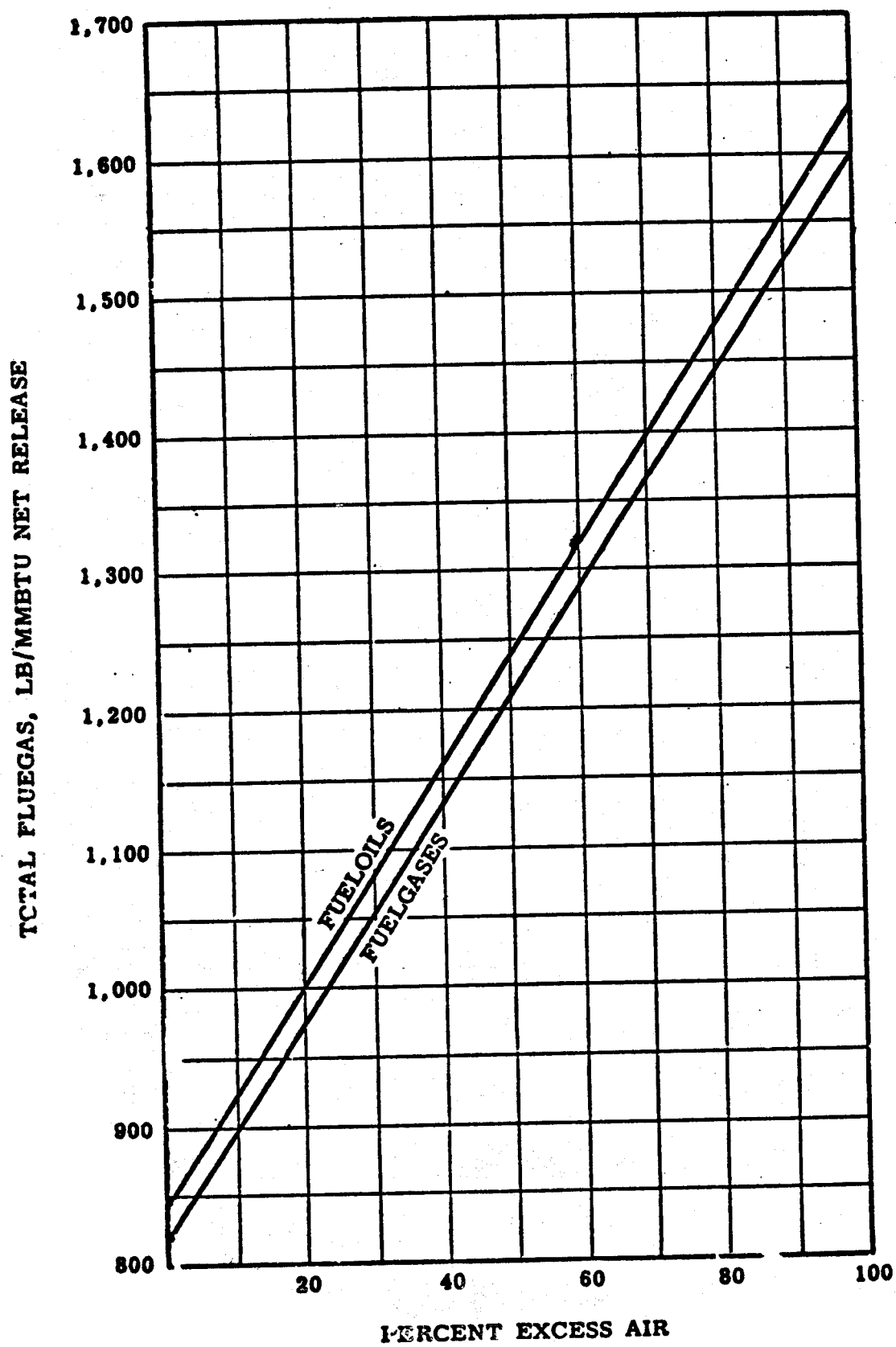


FIGURE 8—Flue gas rates.¹

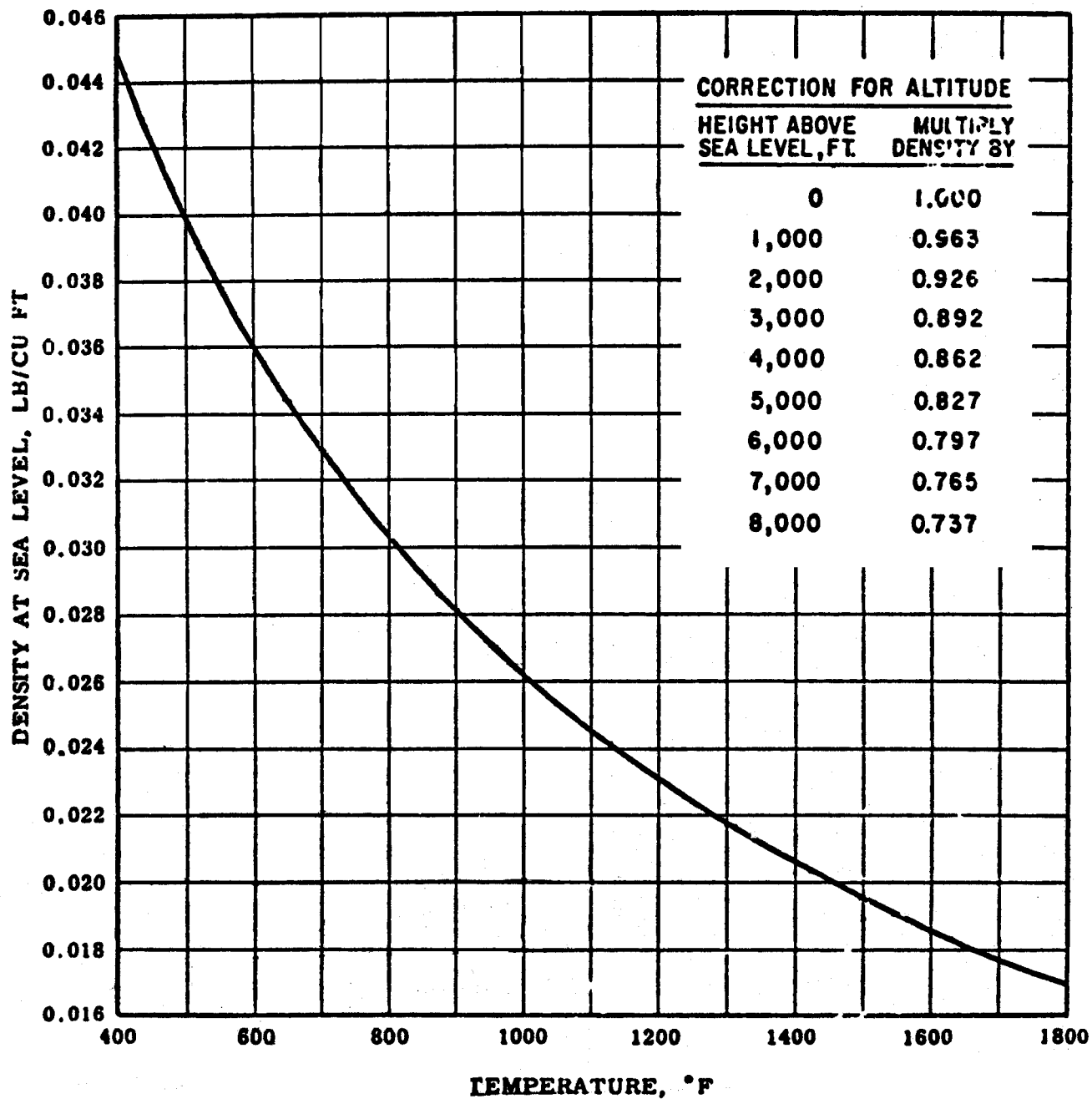


FIGURE 9 - Density of flue gas.¹

**FIGURE 8A
VERTICAL HEATER
BOTTOM SUPPORTED TUBES**

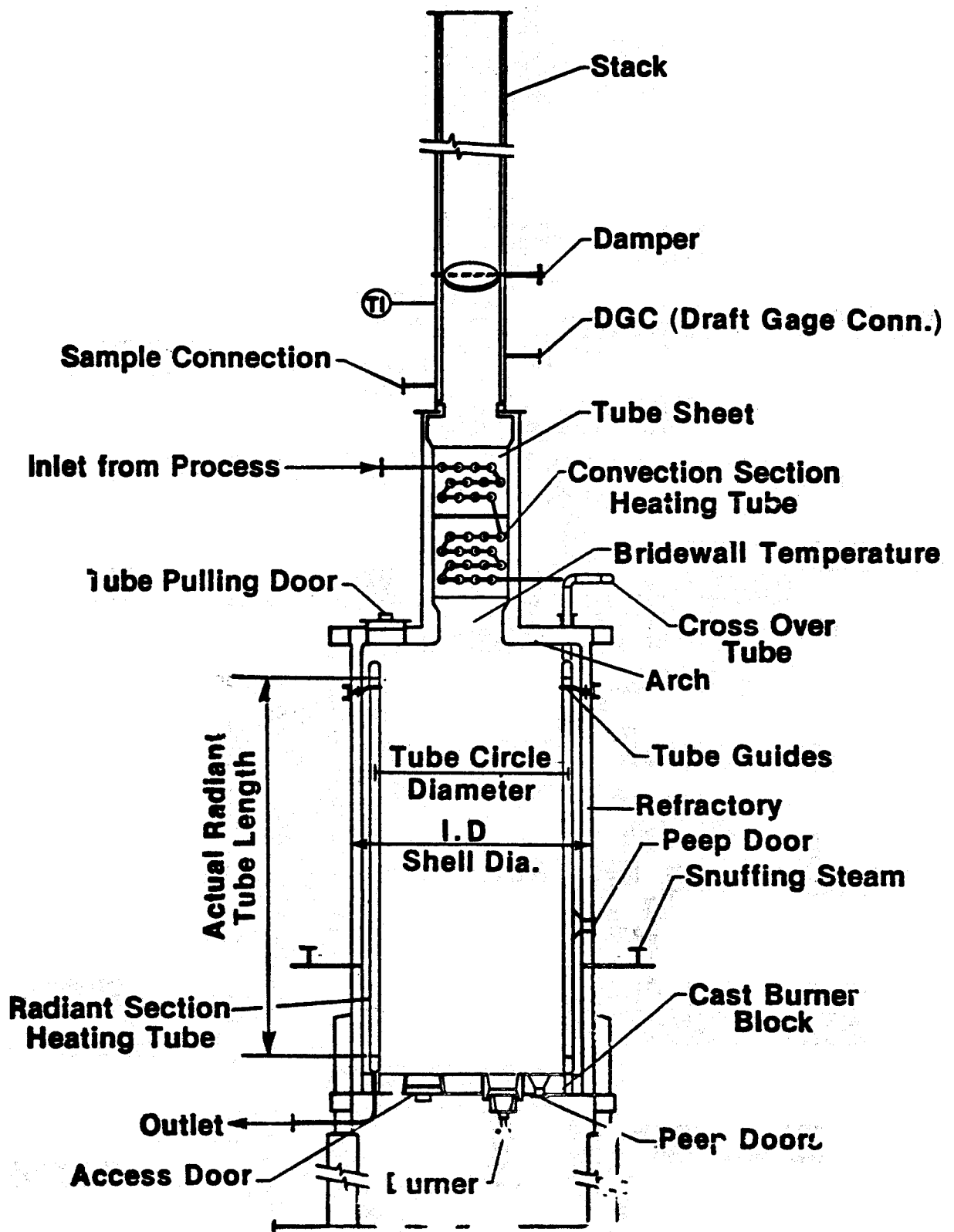
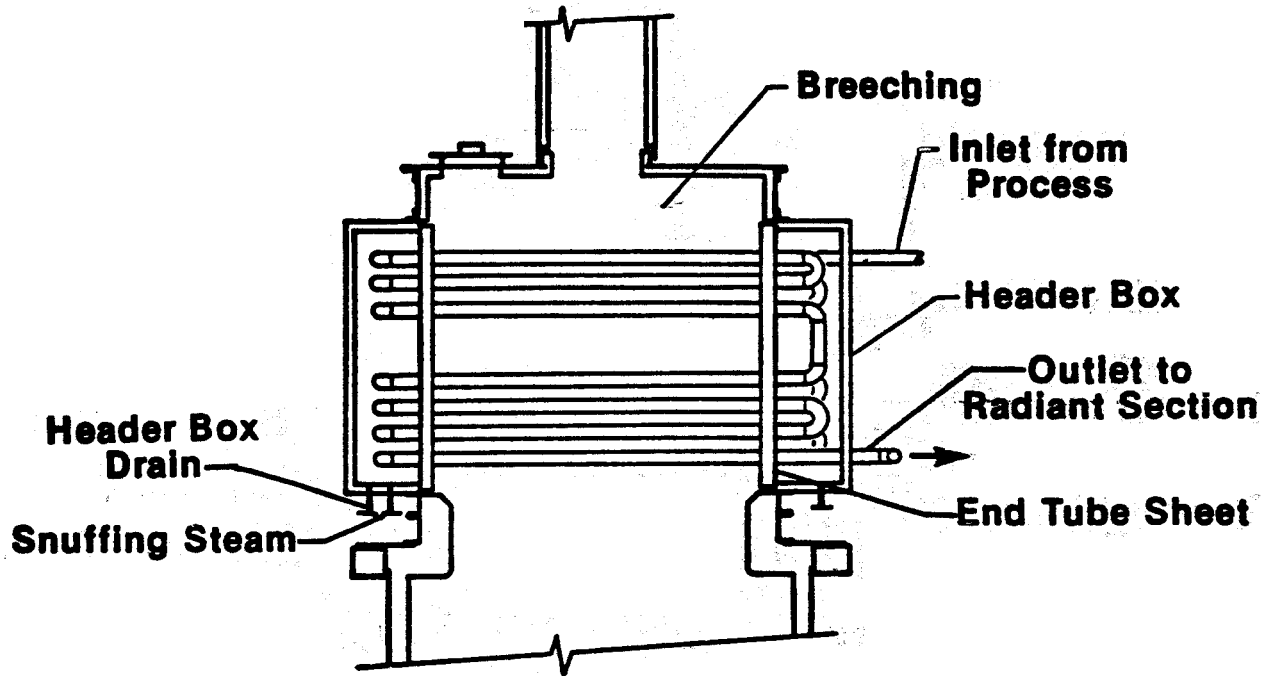
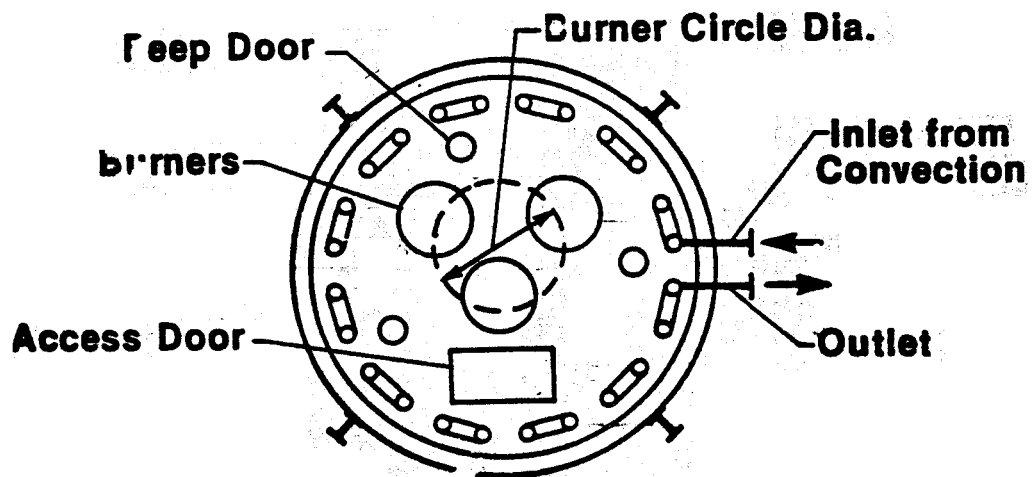


FIGURE 8B



**CONVECTION SECTION
FOR VERTICAL HEATER**



RADIANT SECTION

FIGURE 9A
SIDE FIRED CENTERWALL HEATER

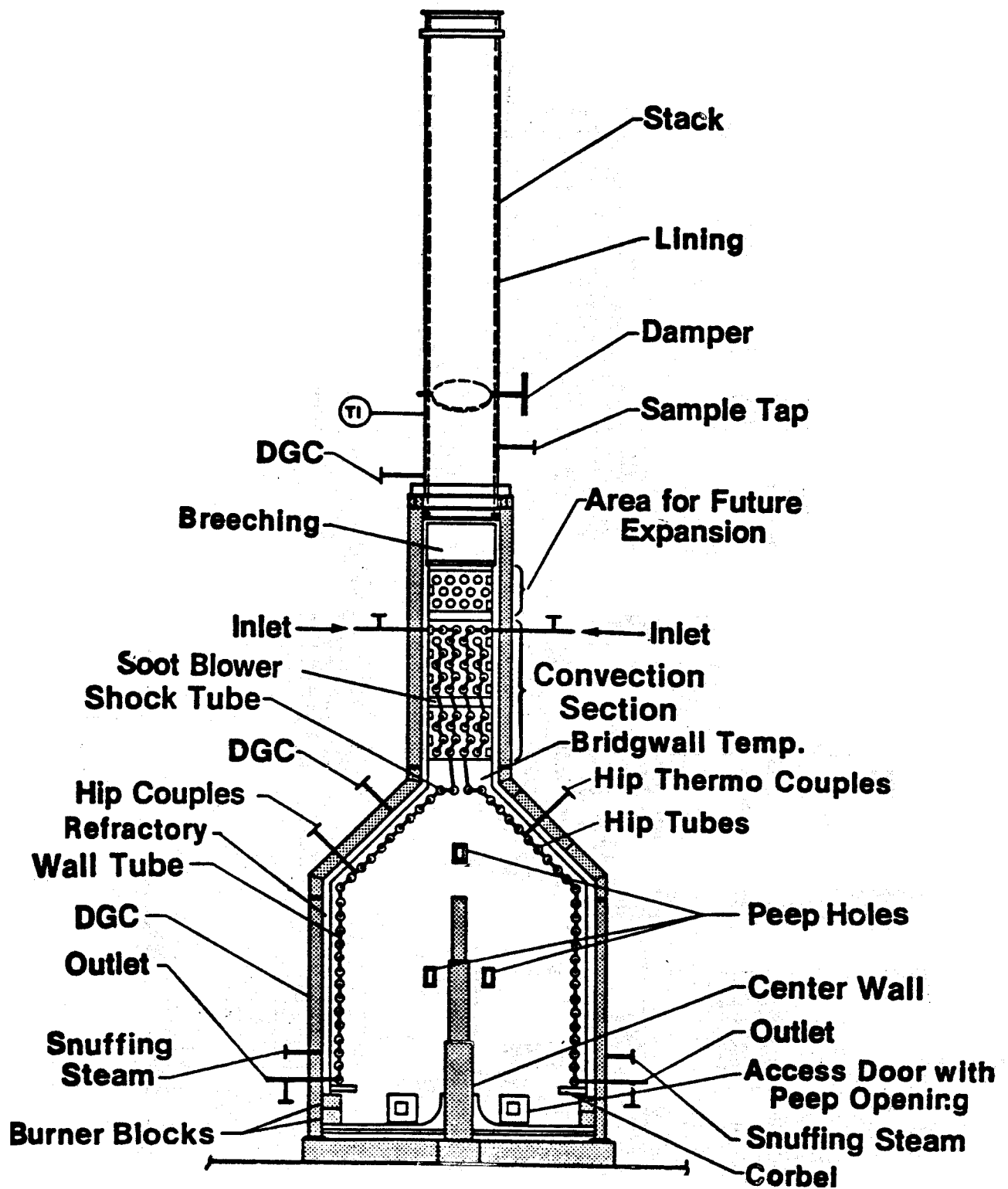


FIGURE 9B
SIDE VIEW CENTERWALL

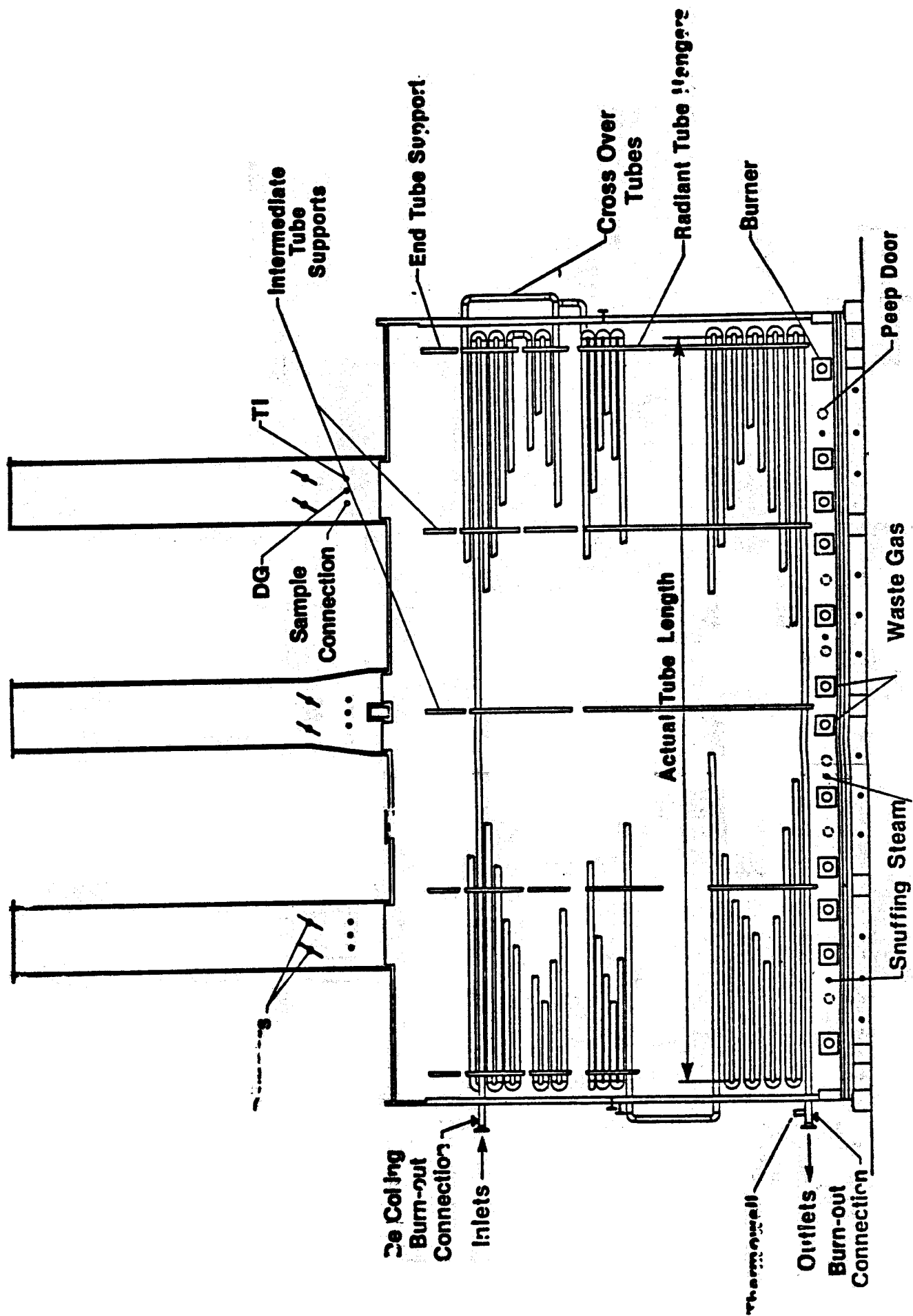


FIGURE 10A

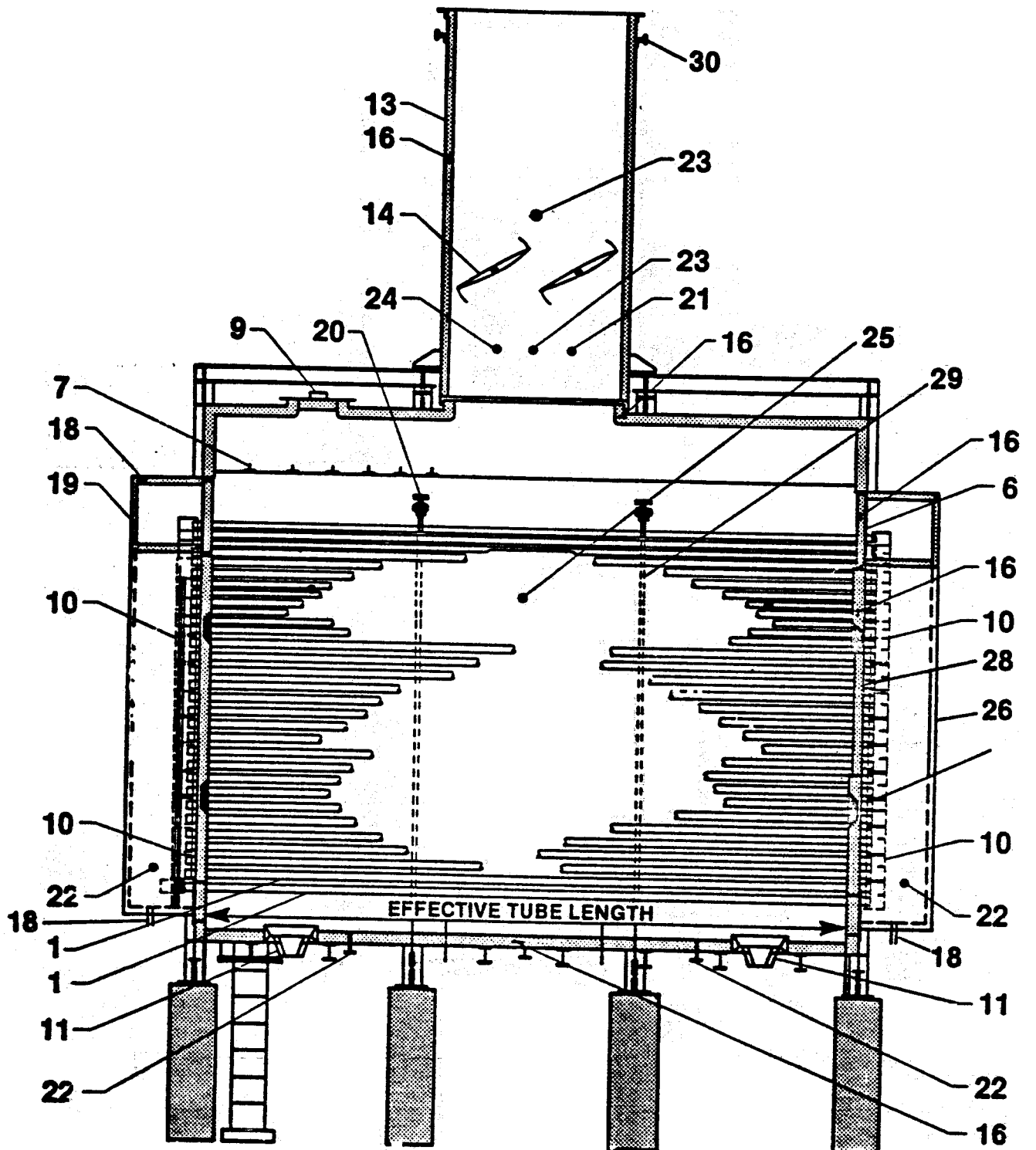


FIGURE 10B

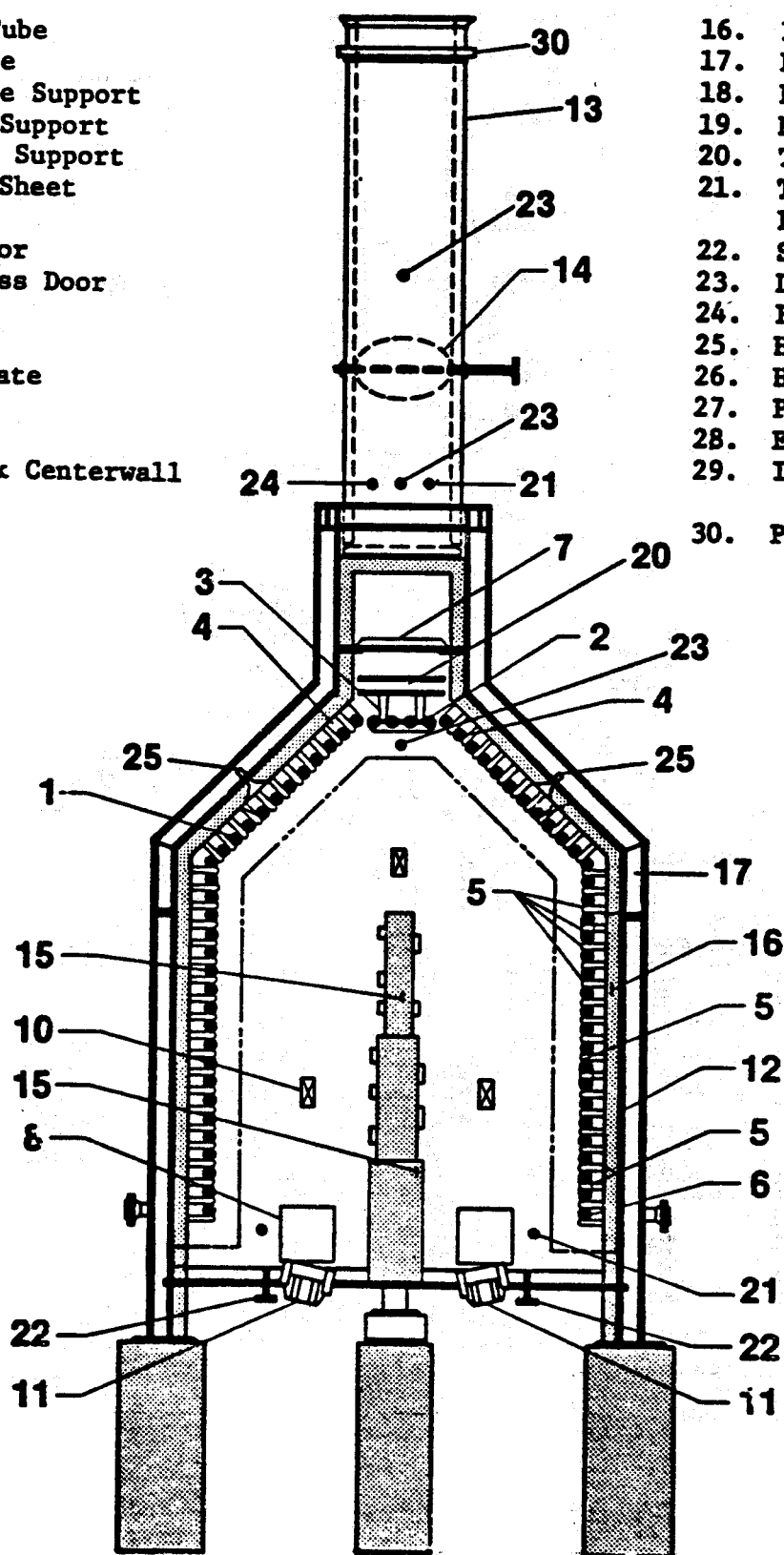
ALL RADIANT BOTTOM FIRED CENTERWALL

No. Nomenclature

1. Radiant Tube
2. Shock Tube
3. Shock Tube Support
4. Hip Tube Support
5. Wall Tube Support
6. End Tube Sheet
7. Baffle
8. Access Door
9. Roof Access Door
10. Peep Door
11. Burner
12. Casing Plate
13. Stack
14. Damper
15. Fire Brick Centerwall

No. Nomenclature

16. Insulation Concrete
17. Buckstay
18. Drain
19. Mineral Wool Blanket
20. Tube Support Beam
21. Thermocouple Conn. for Flue Gas Temperature
22. Snuffing Steam Conn.
23. Draft Gage Conn.
24. Flue Gas Sampling Conn.
25. Hip Thermocouple Conn.
26. Header Box
27. Plug Type Fittings
28. End Tube Sheet
29. Intermediate Tube Supports
30. Painters Trolley



**FIGURE 11A
DOUBLE FIRED HEATER**

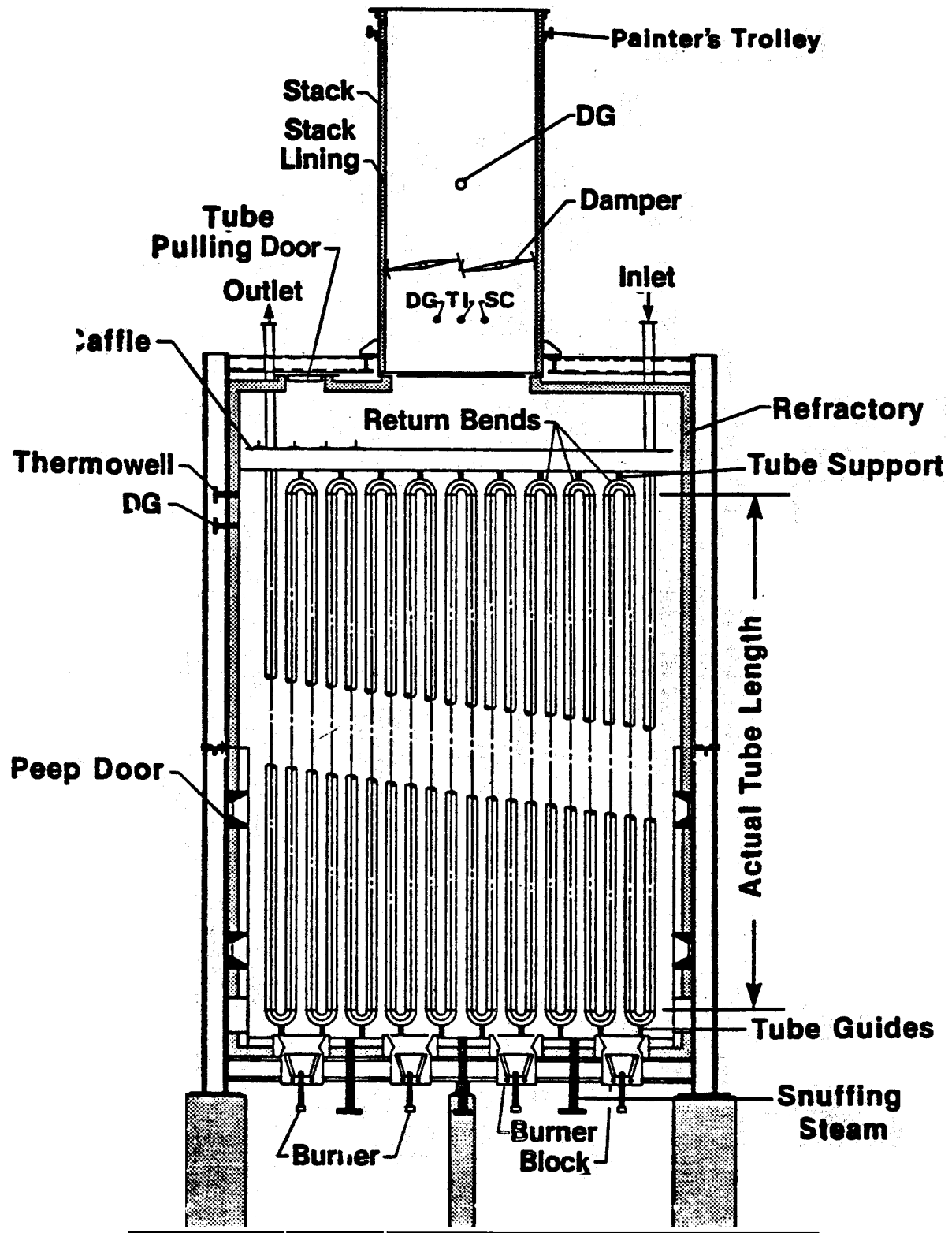


FIGURE 11B

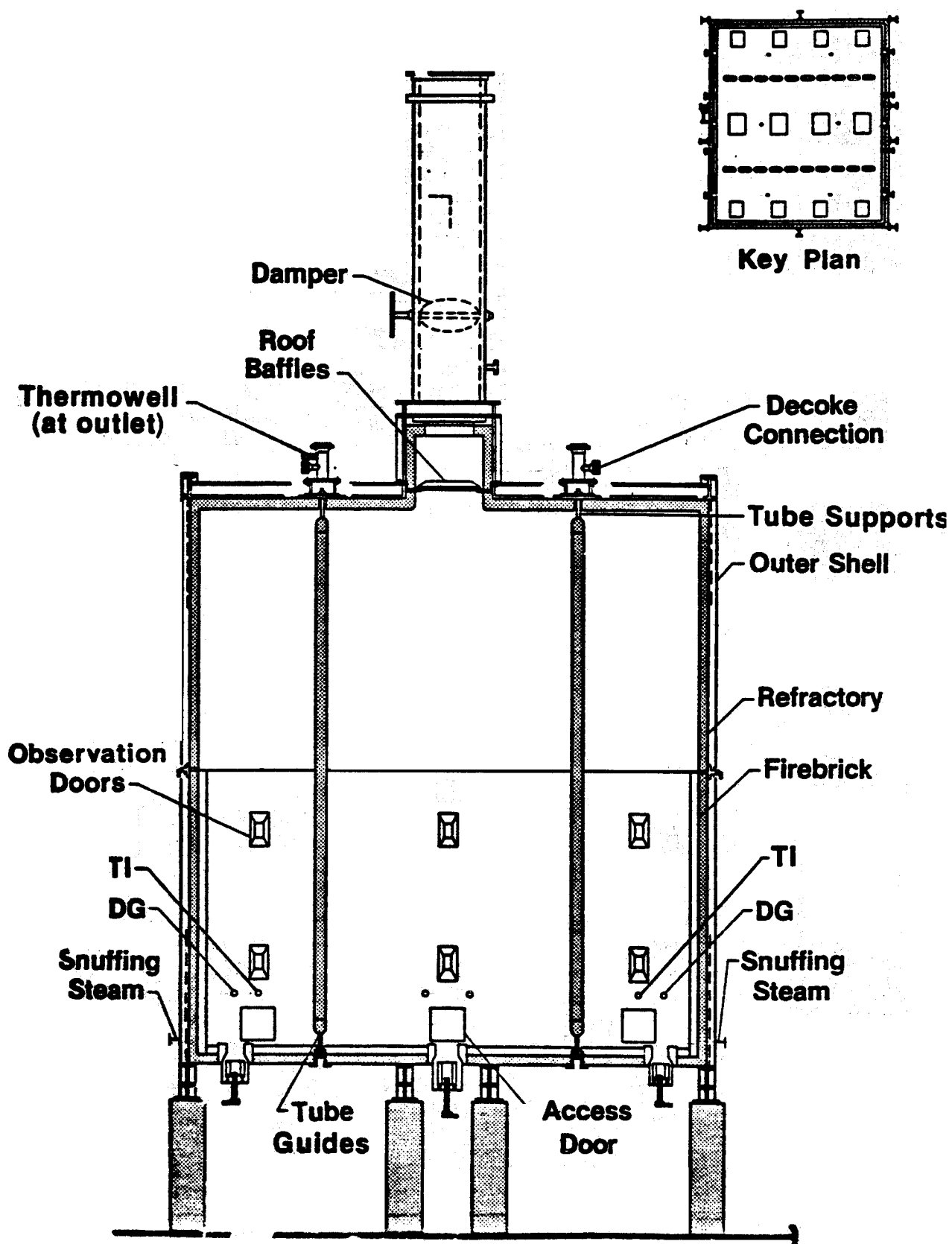


FIGURE 12 HELICAL COIL HEATER

No. Nomenclature

1. Heating Tube
2. Tube Support
3. Top Support Guide
4. Shell
5. Stack
6. Painters Trolley
7. Damper
8. Platform (1st)
9. Platform (2nd)
10. Ladder with Cage
11. Ladder
12. Access Door

No. Nomenclature

13. Peep Door
14. Burner
15. Insulation
16. Refractory
17. Floor Peep Door
18. Draft Gage
19. Sample Point
20. Stack Ti Thermowell
21. Fire Box Ti Thermowell
22. Snuffing Steam
23. Damper Position Indicator

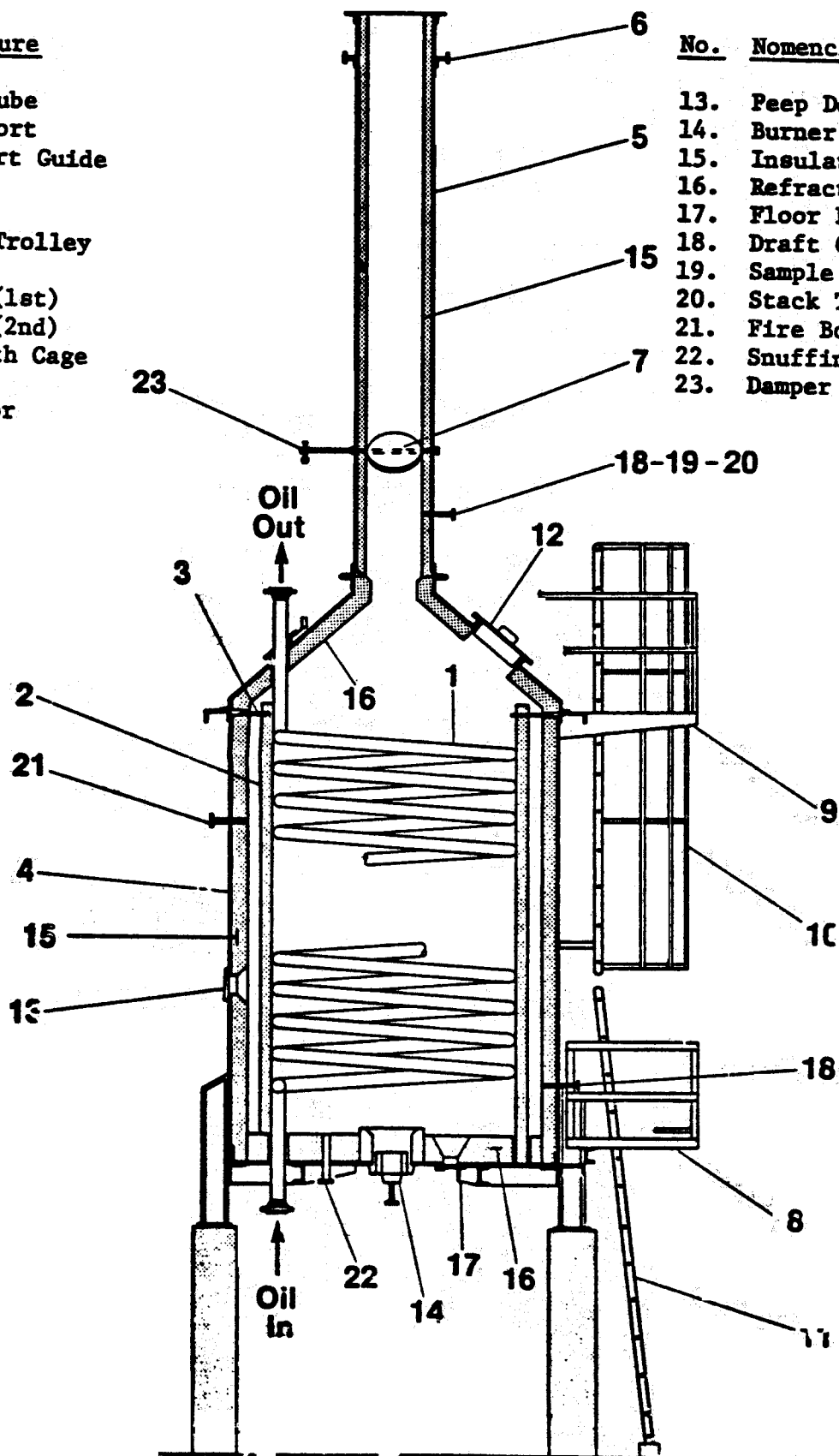


FIGURE 13

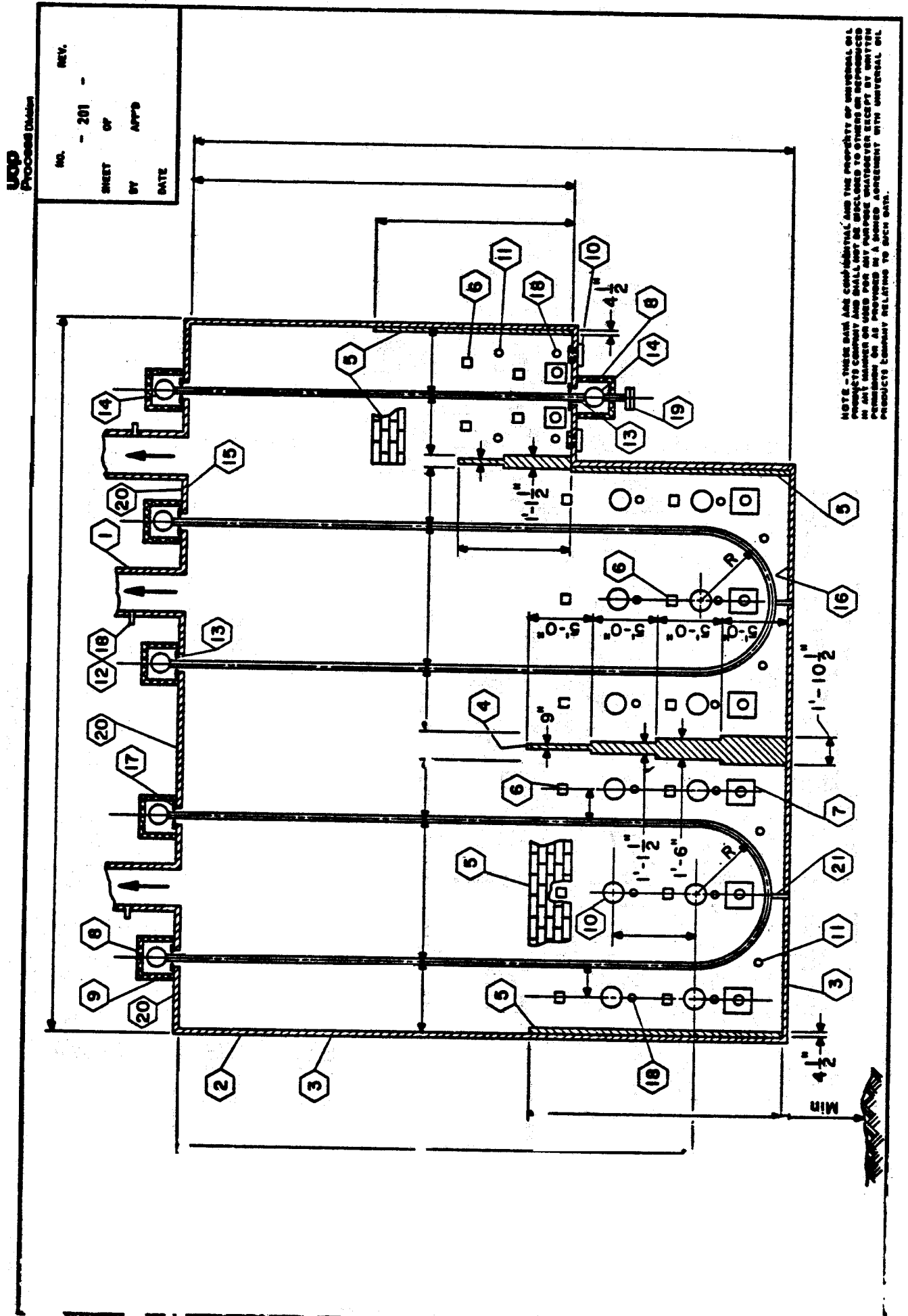
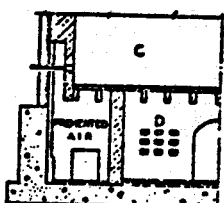
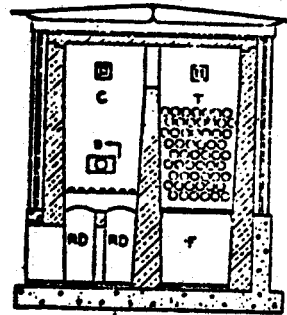
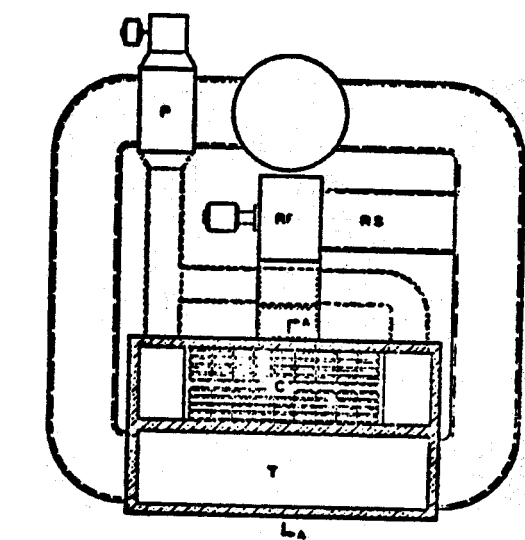


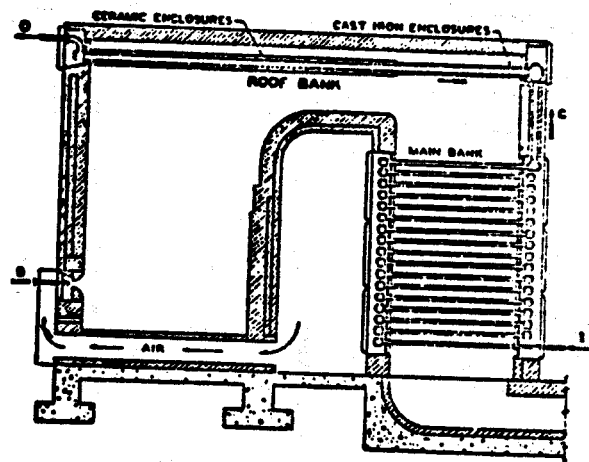
FIGURE 14a



SECTION A-A

HALF SECTION B-B

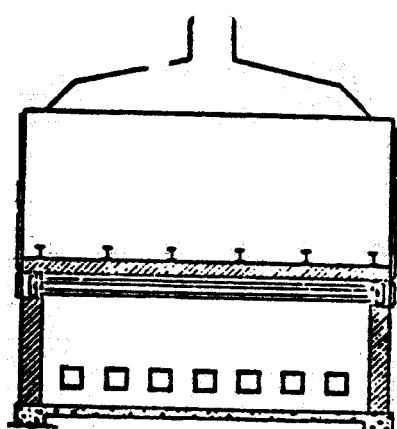
Flue Gas Recirculating Furnace with Air preheater
 C—Combustion chamber; T—Tube or heating chamber; P—Flue to stack;
 RF—Recirculating fan; RD—Recirculating gas distributing duct; RS—
 Recirculating fan suction duct; T—Air, preheater



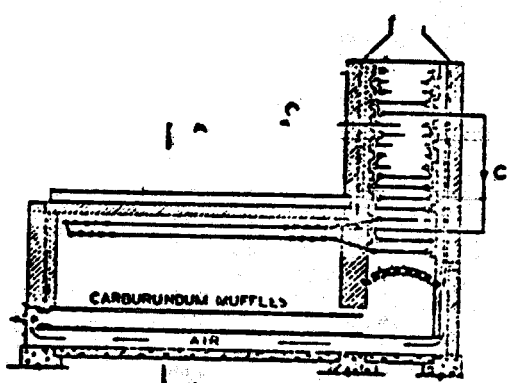
Early shickled tube radiant heat oil heater
 I—Oil inlet; O—Oil outlet; C—Outside cross-over; B—Burners

I

Early shickled flame radiant heat oil heater
 I—Oil inlet; O—Oil outlet; C—
 Outside cross-over



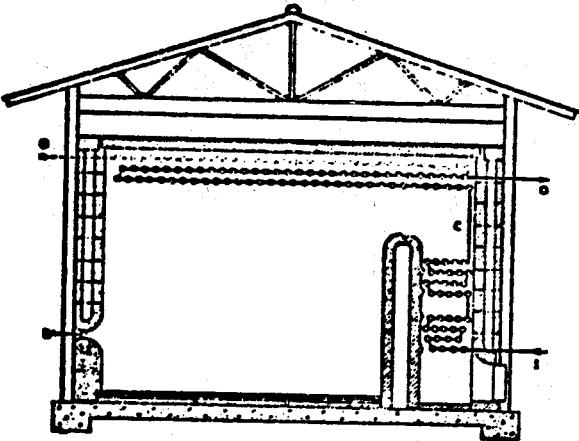
SECTION A-A



L.A.

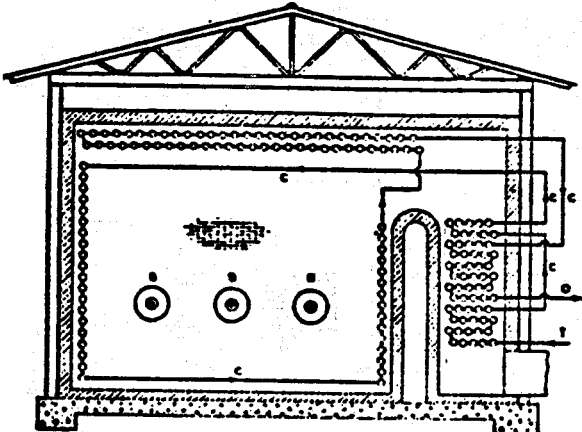
C

FIGURE 14b



Early open-flame heater with roof tubes only
I—Oil inlet; O—Oil outlet; B—Burners

D



Open-flame heater with roof and wall tubes
I—Oil inlet; O—Oil outlet; C—Outside cross-overs; B—Burners

E

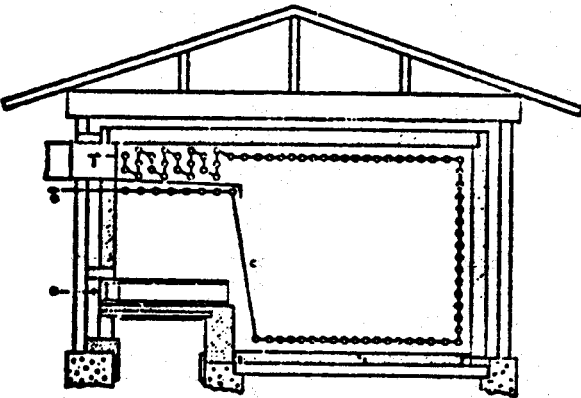
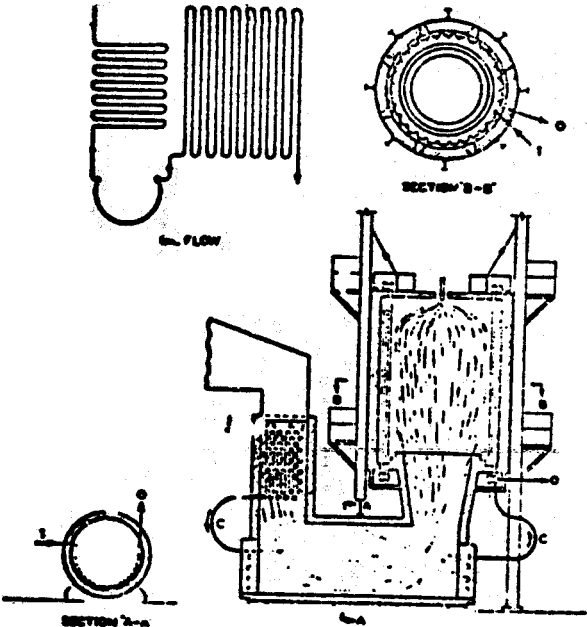


Fig. 19. Partially shielded flame heater
I—Oil inlet; O—Oil outlet; C—Outside cross-over; B—Burners

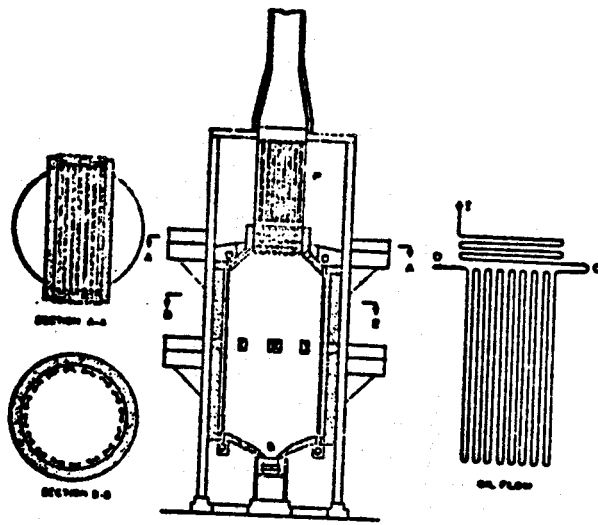
F



Early down-draft cylindrical heater
I—Oil inlet; O—Oil outlet; C—Outside cross-overs

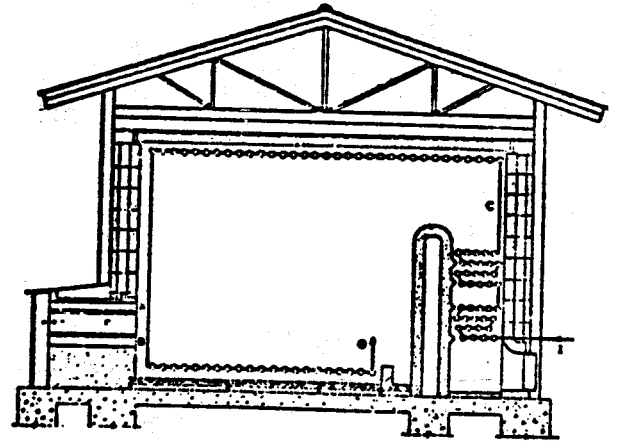
G

FIGURE 14c



Updraft cylindrical heater

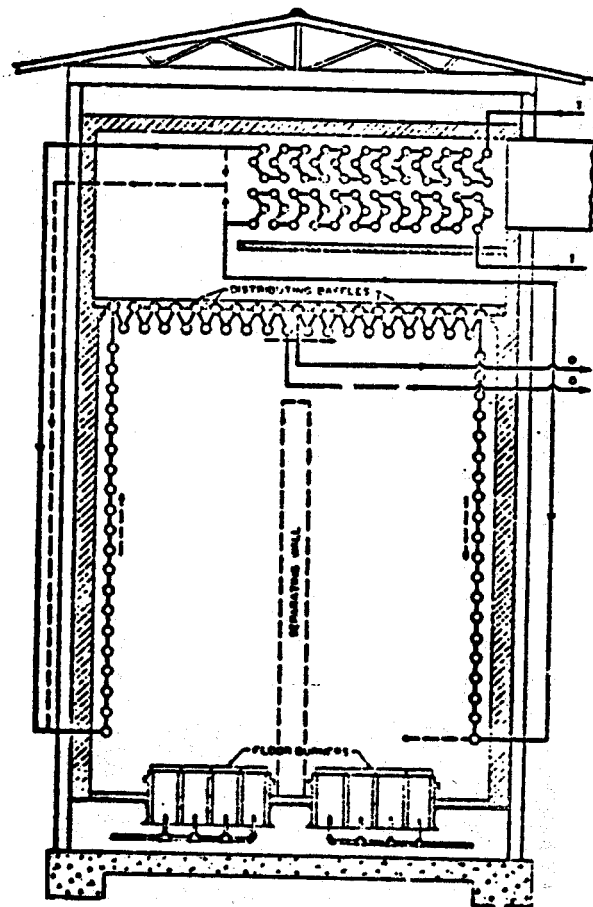
I—Oil inlet; O—Oil outlet; B—Burner; P—Air preheater



Roof and floor tube heater

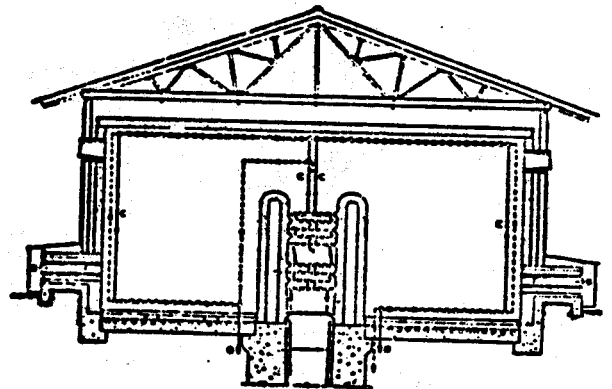
I—Oil inlet; O—Oil outlet; C—Outside cross-over

H



Updraft rectangular heater

I

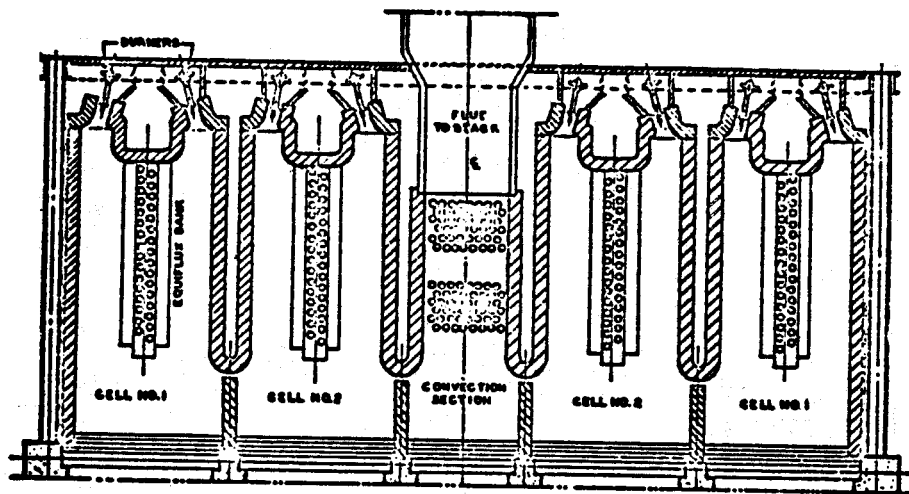


... air apartment heater

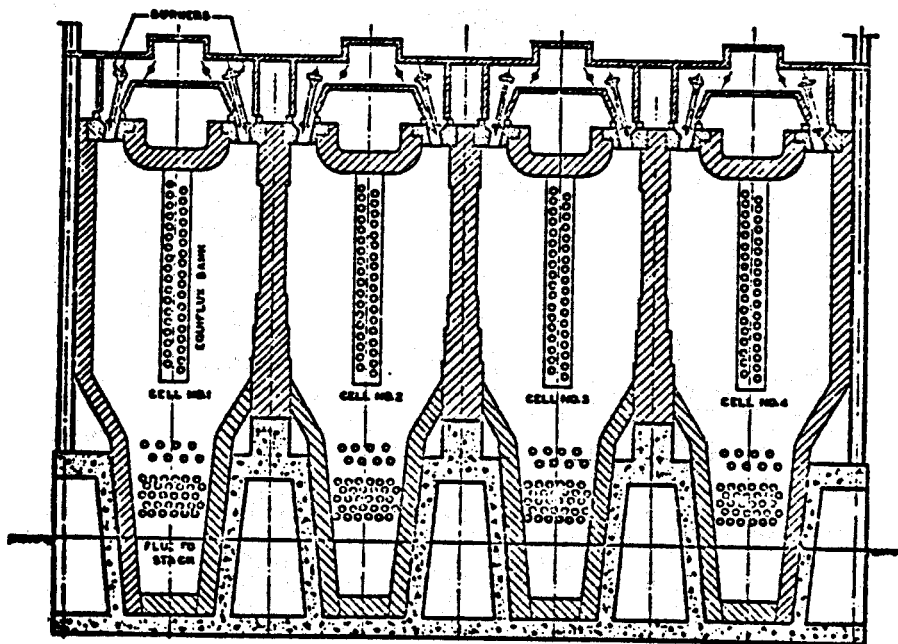
I

J

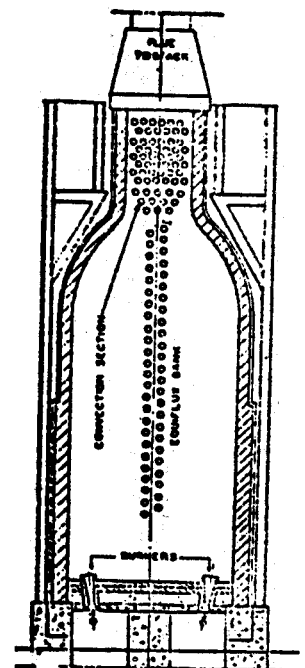
FIGURE 15



A



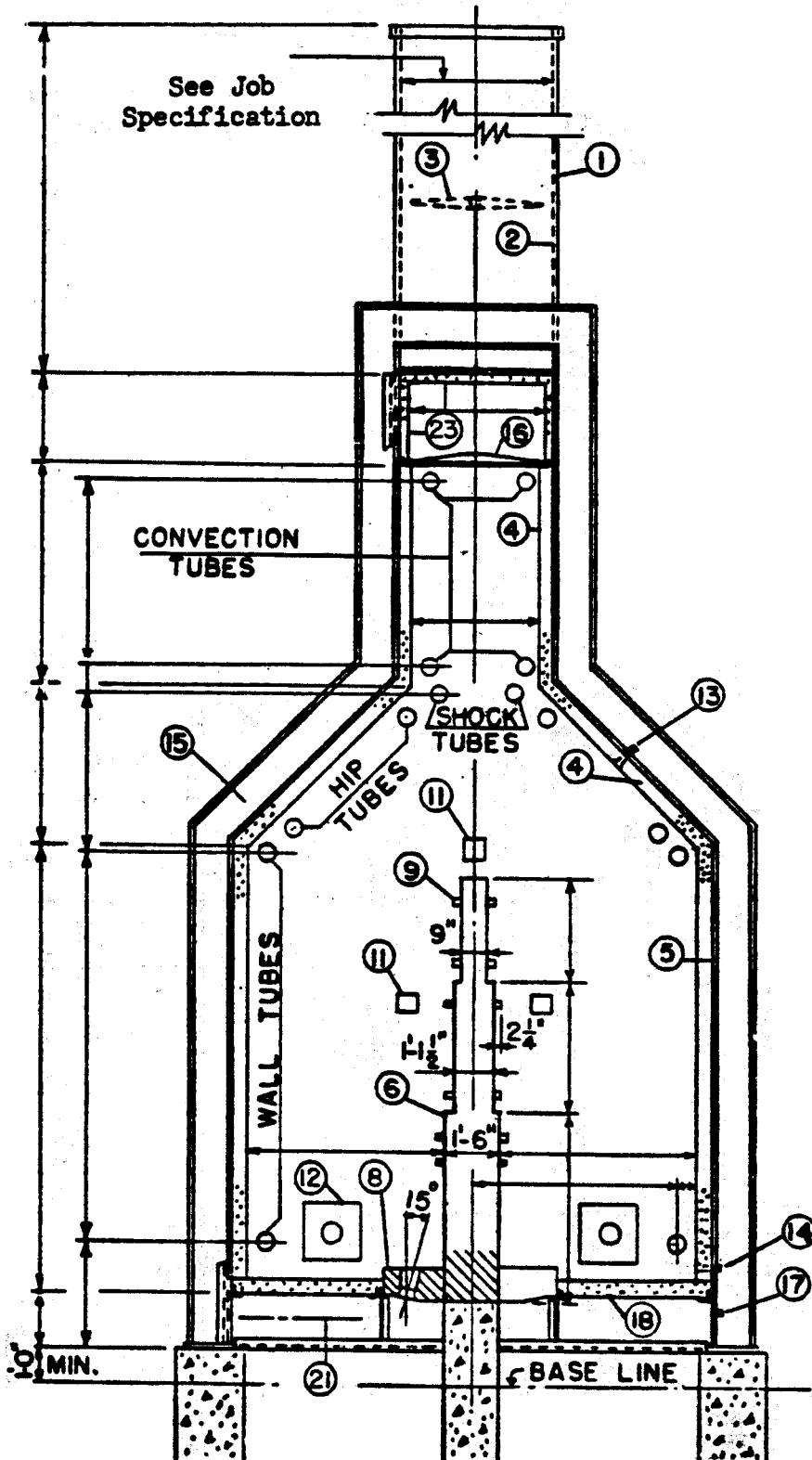
B



C

FIGURE 16

CENTER-WALL UPDRAFT
CROSS SECTION

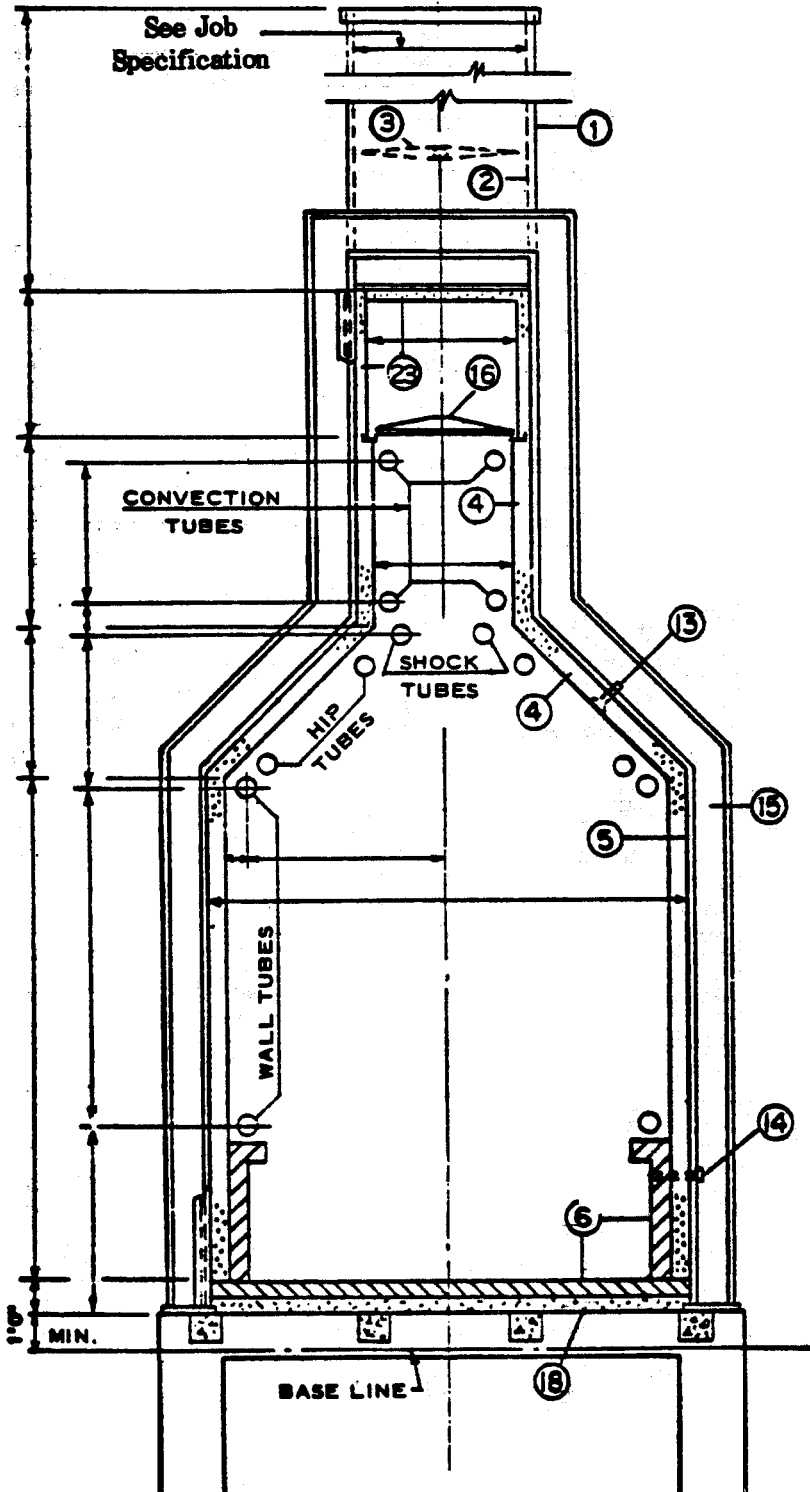


1. Stack $\frac{1}{4}$ " steel plate minimum thickness.
2. $1\frac{1}{2}$ " insulation UOP Specification No. 292.
3. 18-8 stainless steel damper and damper parts with positive manual 2-way control from grade.
4. 5" insulating concrete UOP Specification No. 294 or No. 296.
5. $\frac{3}{16}$ " steel casing plate
6. High duty regular type firebrick ASTM C-27.
8. High duty regular type fireclay precast burner blocks ASTM C-27.
9. Brick projections.
11. Peep openings (both ends) See Job Specification
12. Access openings (both ends) See Job Specifications.
13. $1\frac{1}{2}$ " thermocouple connections (plugged) See Job Specifications
14. 1" steam smothering connections. See Job Specification.
15. Heater frame (buckstay).
16. Meehanite process HR-1 or alloy plate baffles.
17. $\frac{3}{4}$ " steam smothering connections. See Job Specification
18. 5" reinforced heat resisting concrete UOP Specification No. 293.
21. Gas burners-See Job Specification
23. 3" insulation UOP Specification No. 294 or No. 296.

FIGURE 17

END OR BOTTOM FIRED UPDRAFT

CROSS SECTION



1. Stack $\frac{1}{4}$ " steel plate minimum thickness.
2. $1\frac{1}{2}$ " insulation UOP Specification No. 292.
3. 18-8 stainless steel damper parts with positive manual 2-way control from grade.
4. 5" insulating concrete UOP Specification No. 294, or No. 296.
5. $\frac{3}{16}$ " steel casing plate.
6. High duty regular type firebrick ASTM C-27.
11. Peep openings (both ends) See Job Specification.
12. Access openings (both ends) See Job Specification.
13. $1\frac{1}{2}$ " thermocouple connections (plugged) See Job Specification.
14. 1" steam smothering connections. See Job Specification.
15. Heater frame (buckstay)
16. Meehanite process HR-1 or alloy plate baffles.
18. 5" reinforced heat resisting concrete UOP Specification No. 293.
23. 3" insulation UOP Specification No. 294 or No. 296.
24. Burners - See Job Specification

FIGURE 18
BASIC TYPES OF PROCESS HEATERS

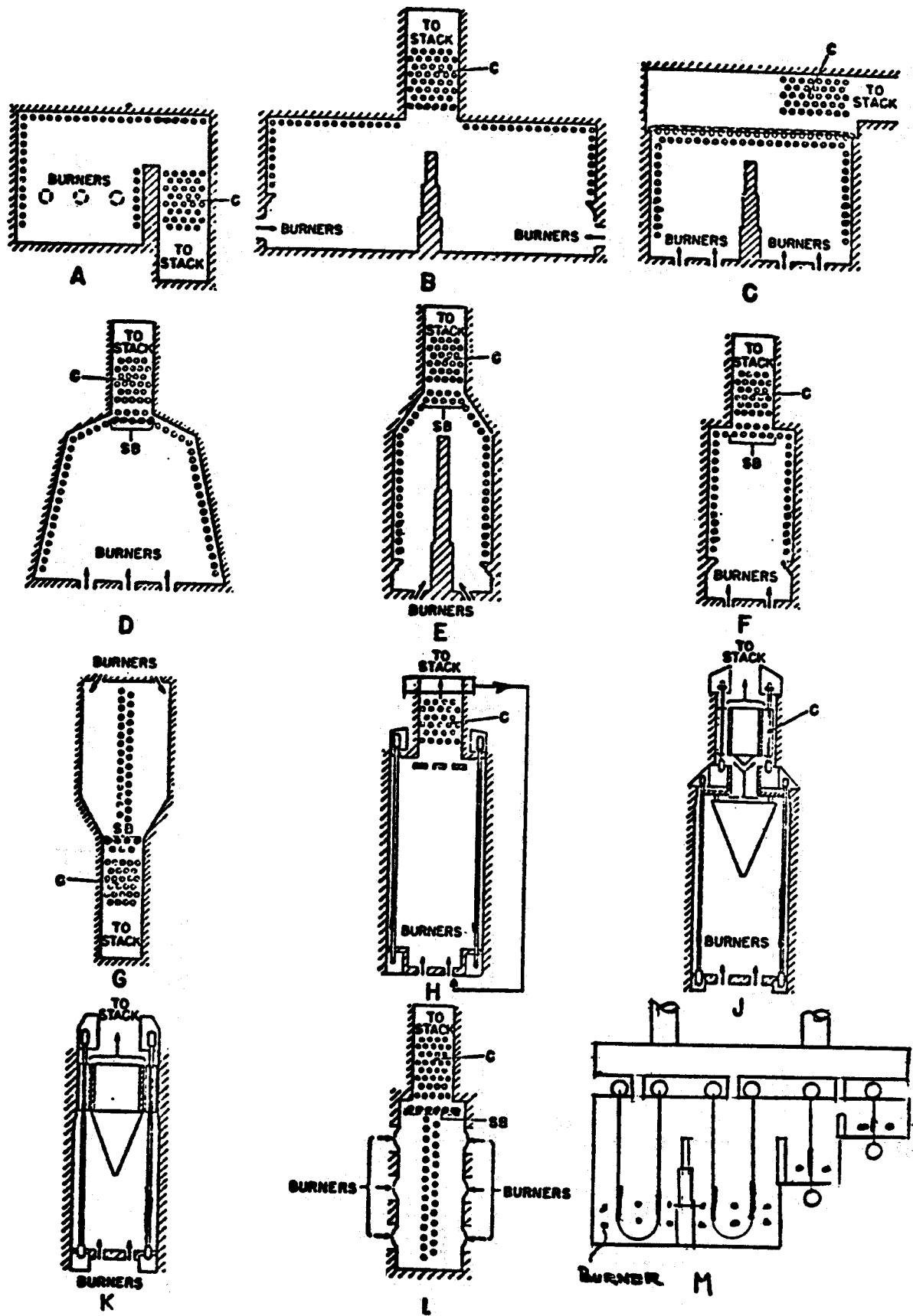


FIGURE 19

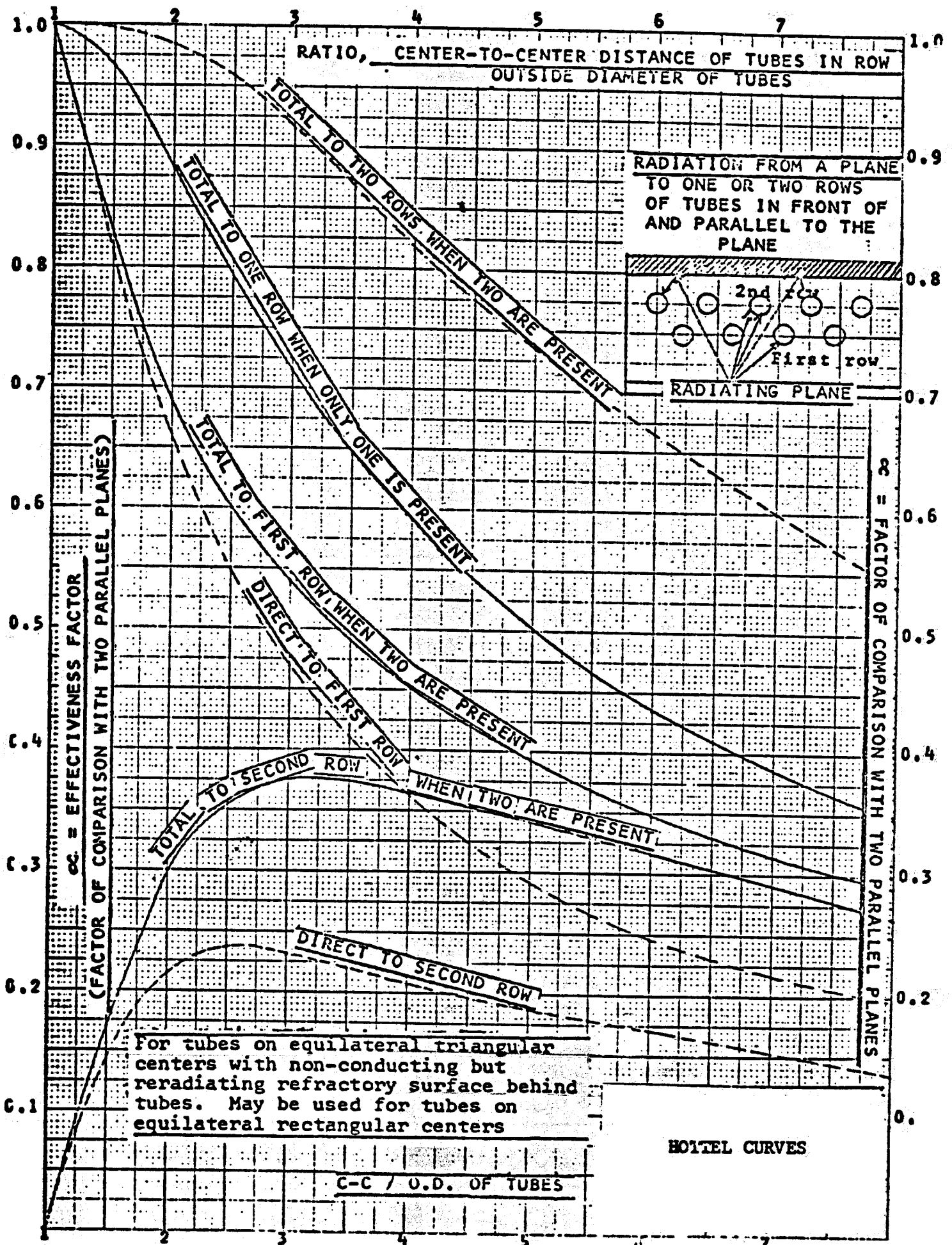


FIGURE 20

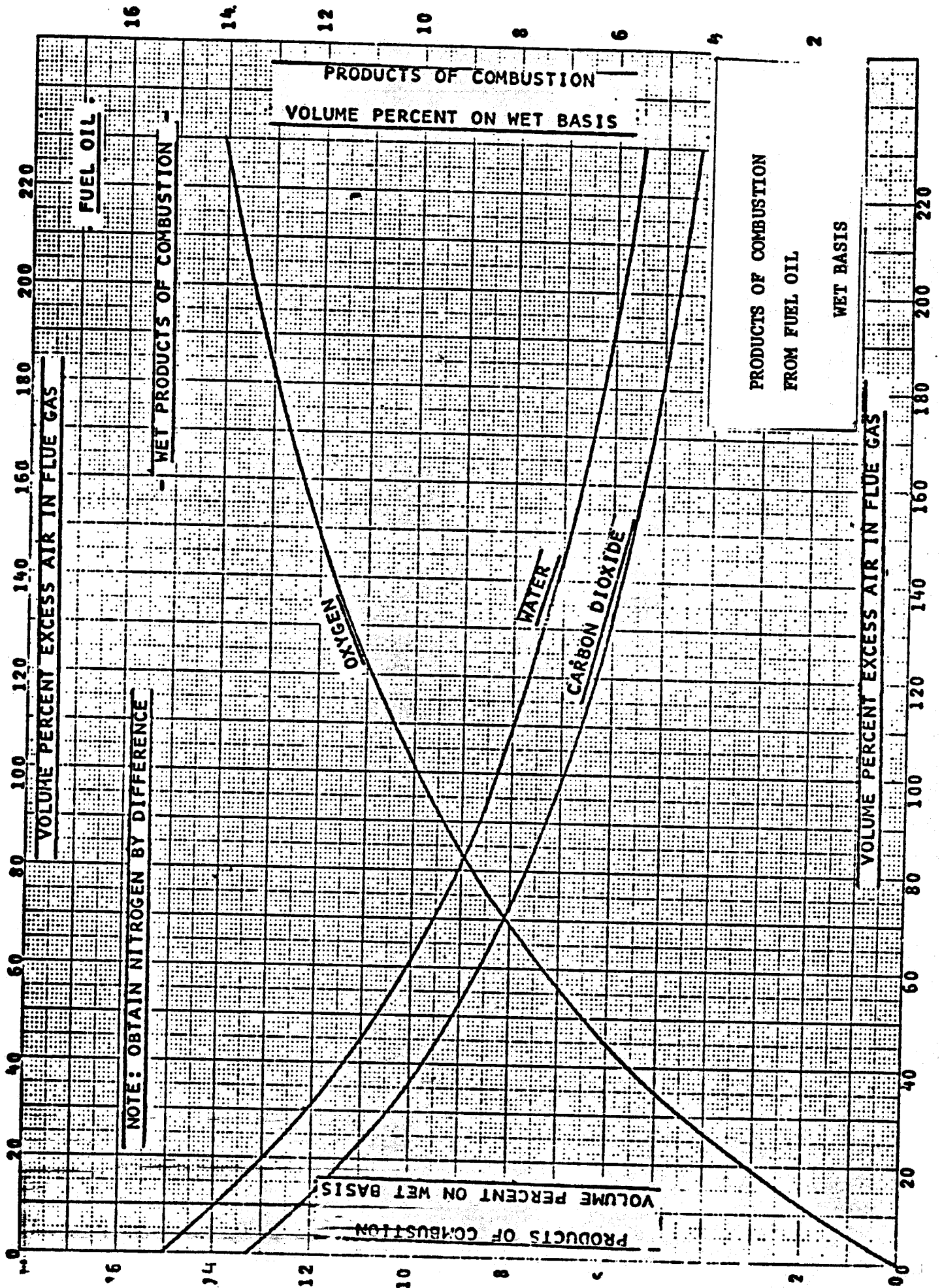
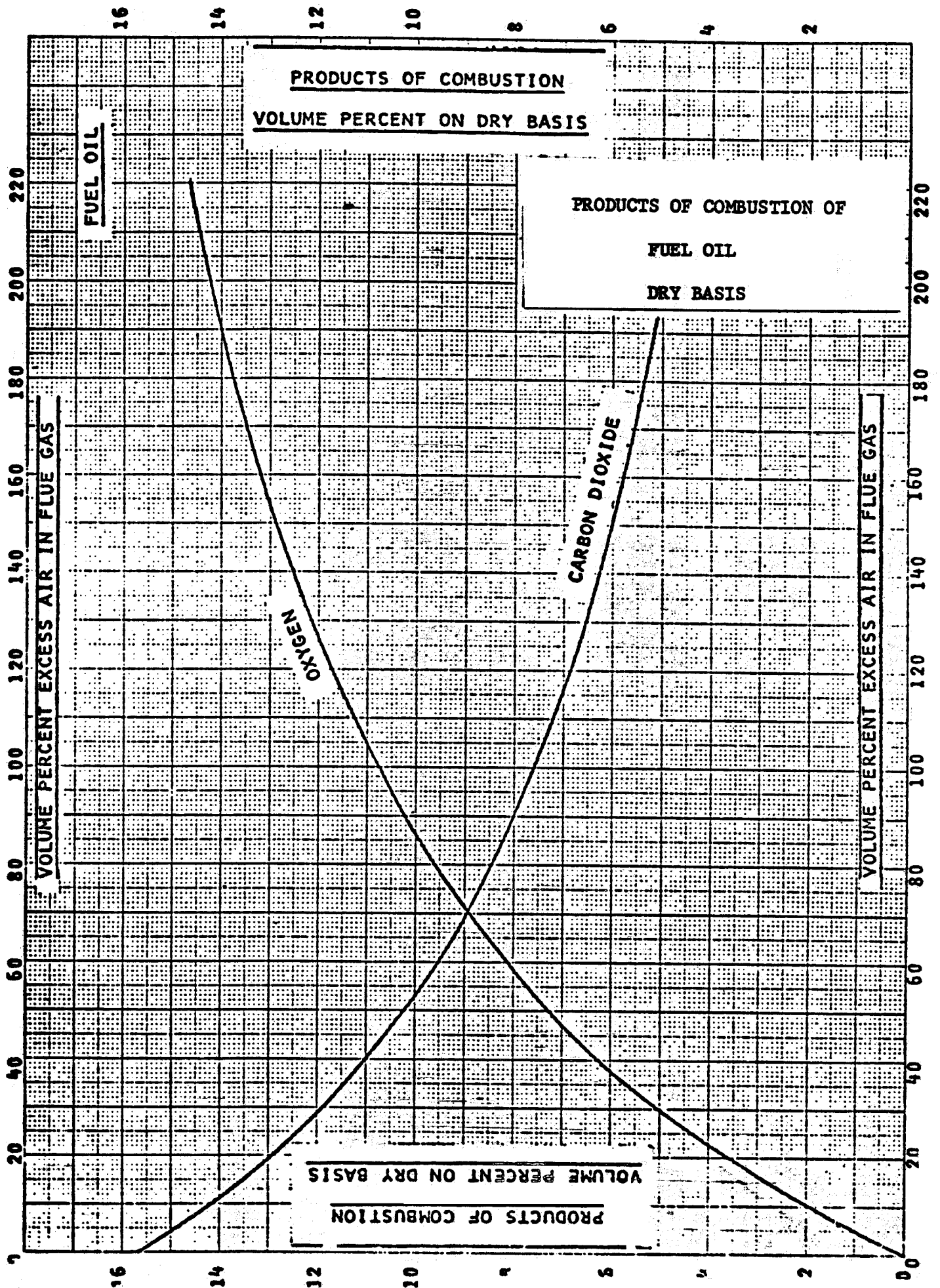
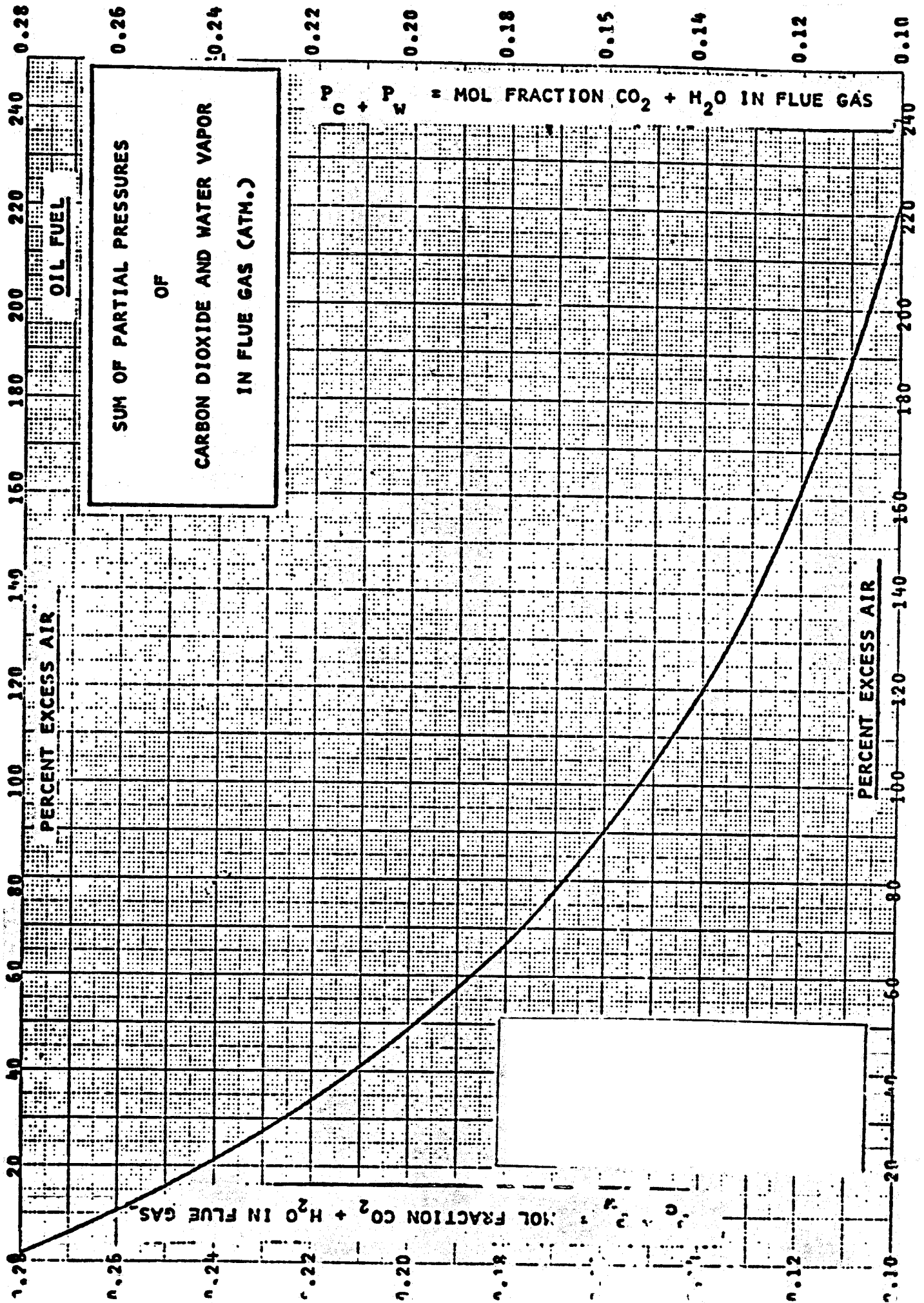


FIGURE 21





The following table gives the factor by which the mean beam length for gas radiation for various standard furnace shapes.

Shape	Characterizing Dimension	Factor by which X is multiplied to obtain mean beam length When $P_{GL} = 0$	For average values of P_{GL}
Cylinders			
Infinite			
Radiation to convex surface	diameter		0.95
Radiation to center of base	diameter	1.00	0.90
Infinite of half-circular cross section, radiating to center of base	radius		1.26
Semi-infinite, radiating to center of base	diameter		0.90
Height = diameter radiating to center of base	diameter		0.77
radiating to whole surface	diameter	0.67	0.60

TABLE 1

	P_{GL}	P_{GL}	P_{GL}
Parallelepipeds			
1:1:1 (cube)	0.01	0.10	1.00
1:1:4	0.61	0.58	0.54
radiating to 1 x 4 face			
radiating to 1 x 1 face	0.82	0.77	0.73
radiating to all faces	0.78	0.74	0.70
1:2:6	0.81	0.77	0.72
radiating to 2 x 6 face			
radiating to 1 x 6 face	1.07	1.02	0.95
radiating to 1 x 2 face	1.13	1.07	1.00
radiating to all faces	1.07	1.02	0.95
1:∞:∞ (infinite parallel planes)	1.09	1.03	0.97
distance between planes	1.82	1.72	1.62

For rectangular parallelepipeds, radiating to the base, values of L for low values of P_{GL} may be obtained from Curve No. B-333. For average values multiply by 0.85. For right circular cylinders of varying height to diameter ratio, radiating to the cylindrical surface L may be obtained from Curve No. B-339.

TABLE I
FIGURE 22a

MEAN LENGTH OF RADIANT BEAMS IN VARIOUS GAS (FURNACE) SHAPES

Dimensional Ratios
(length, width, height in
any order or diameter (d), feet)

Length of Beam, Feet

Rectangular Furnaces:

1:1:1 to 1:1:3

$\frac{2}{3} \sqrt[3]{\text{furnace volume}}$

1:2:1 to 1:2:4

$\frac{2}{3} \sqrt[3]{\text{furnace volume}}$

1:1:4 to 1:1: ∞

1.0 x (smallest dimension)

1:2:5 to 1:2:8

1.3 x (smallest dimension)

1:3:3 to 1: ∞ : ∞

1.8 x (smallest dimension)

Cylindrical Furnaces:

d x d

$\frac{2}{3}$ x (diameter)

d x 2d to d x ∞ d

1.0 x (diameter)

FLAME EMISSIVITY, P_f

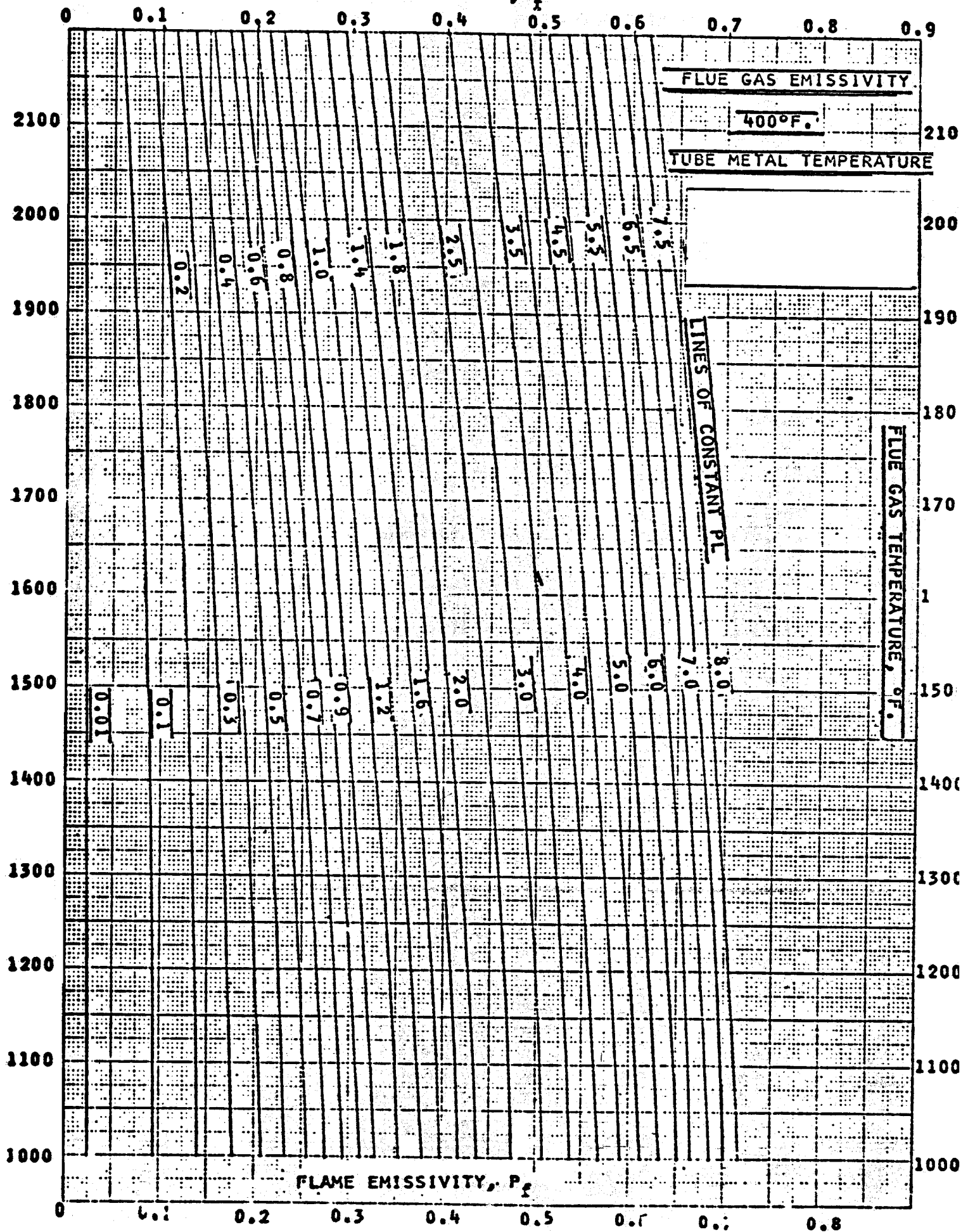
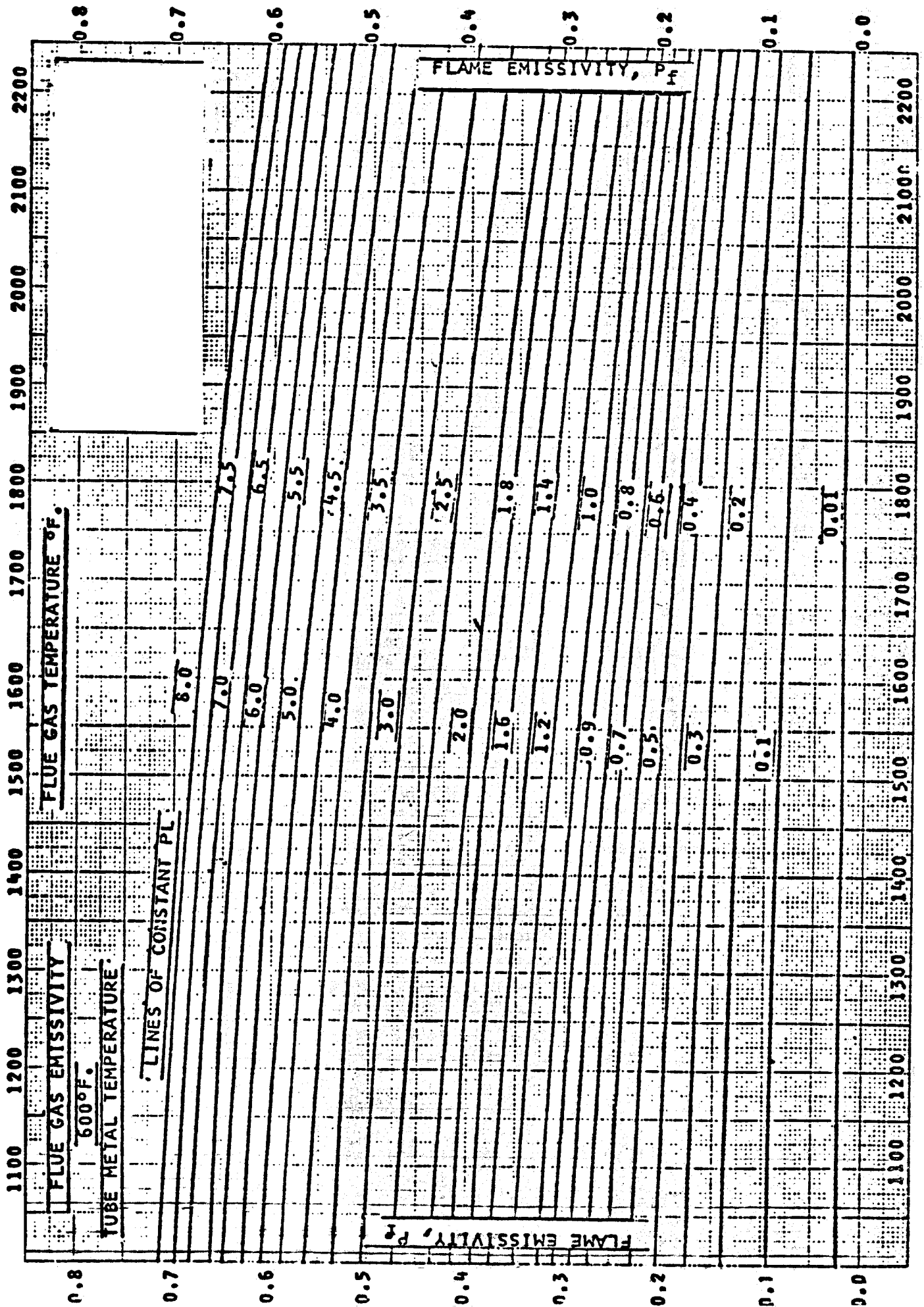


FIGURE 23b



FLUE GAS TEMPERATURE OF.

FIGURE 23c

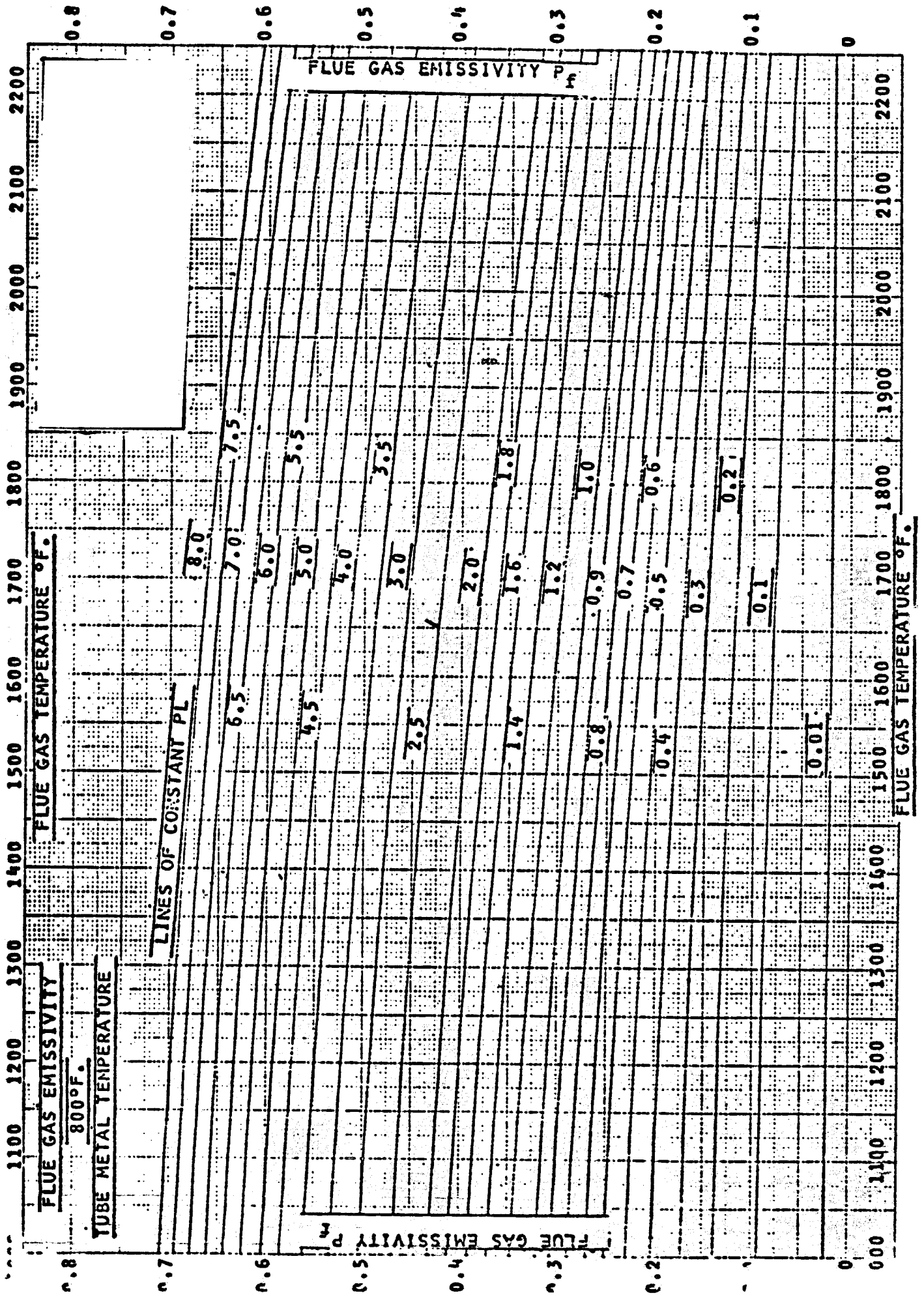


FIGURE 24

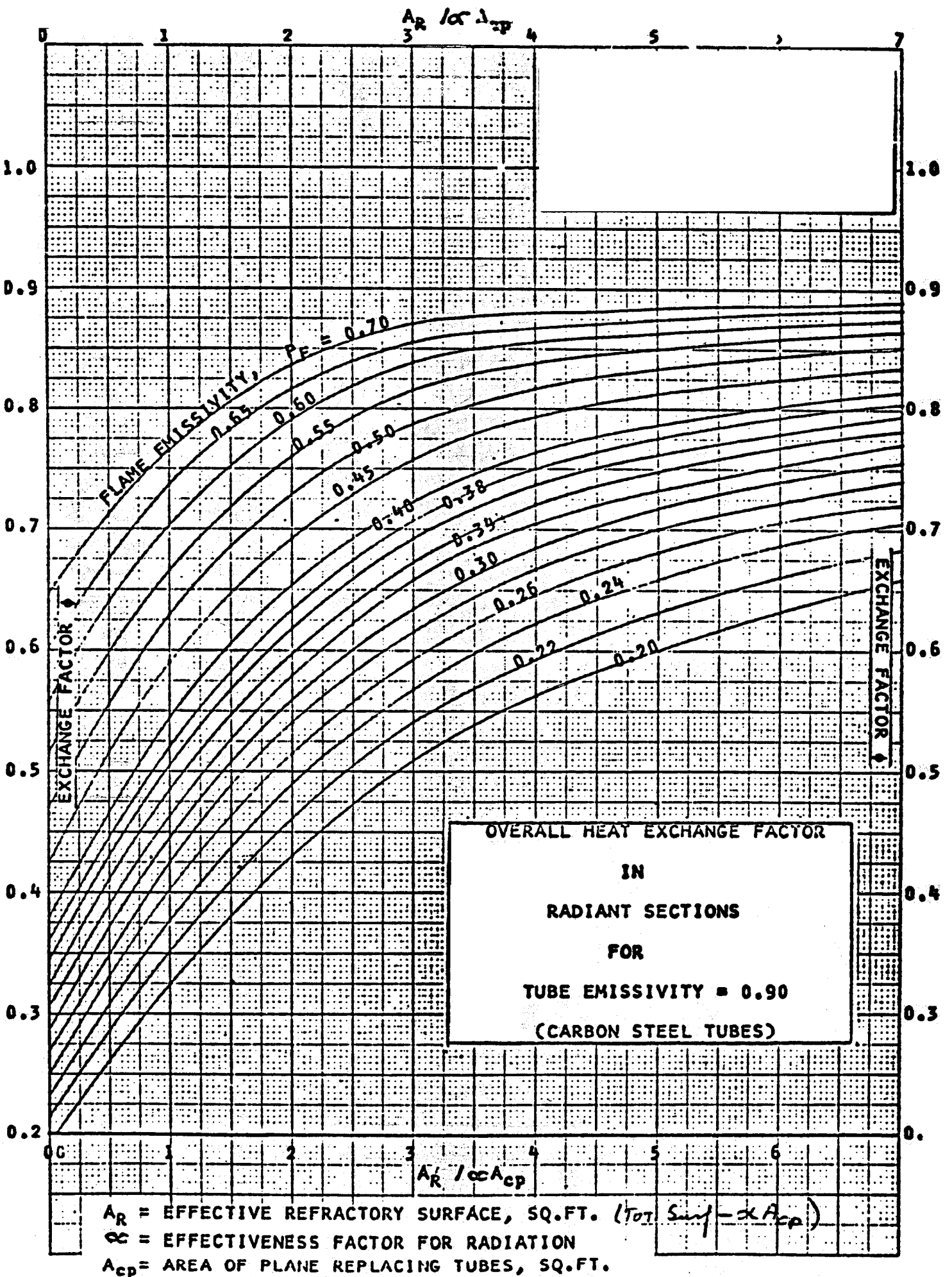


FIGURE 211

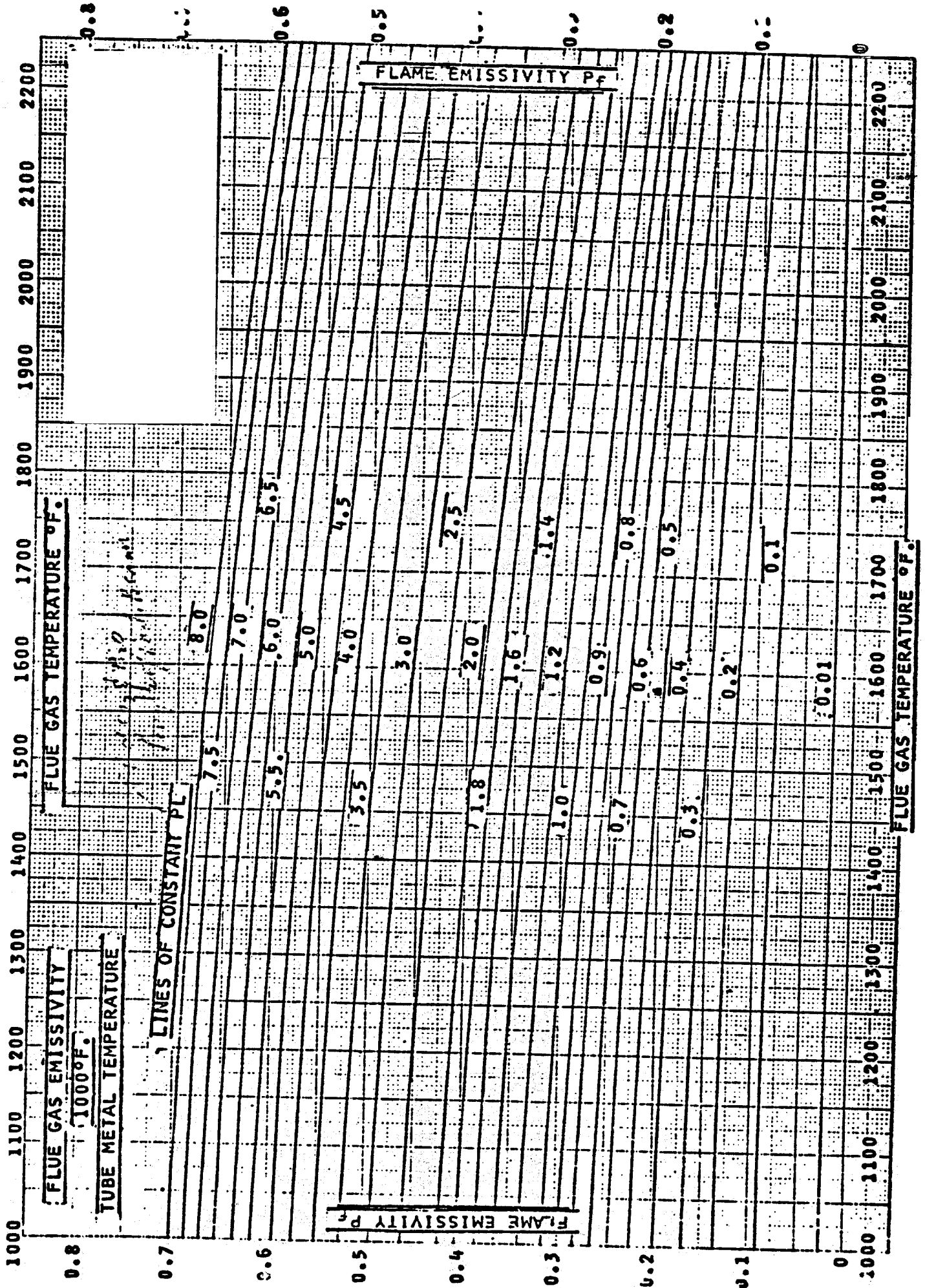


FIGURE 25a

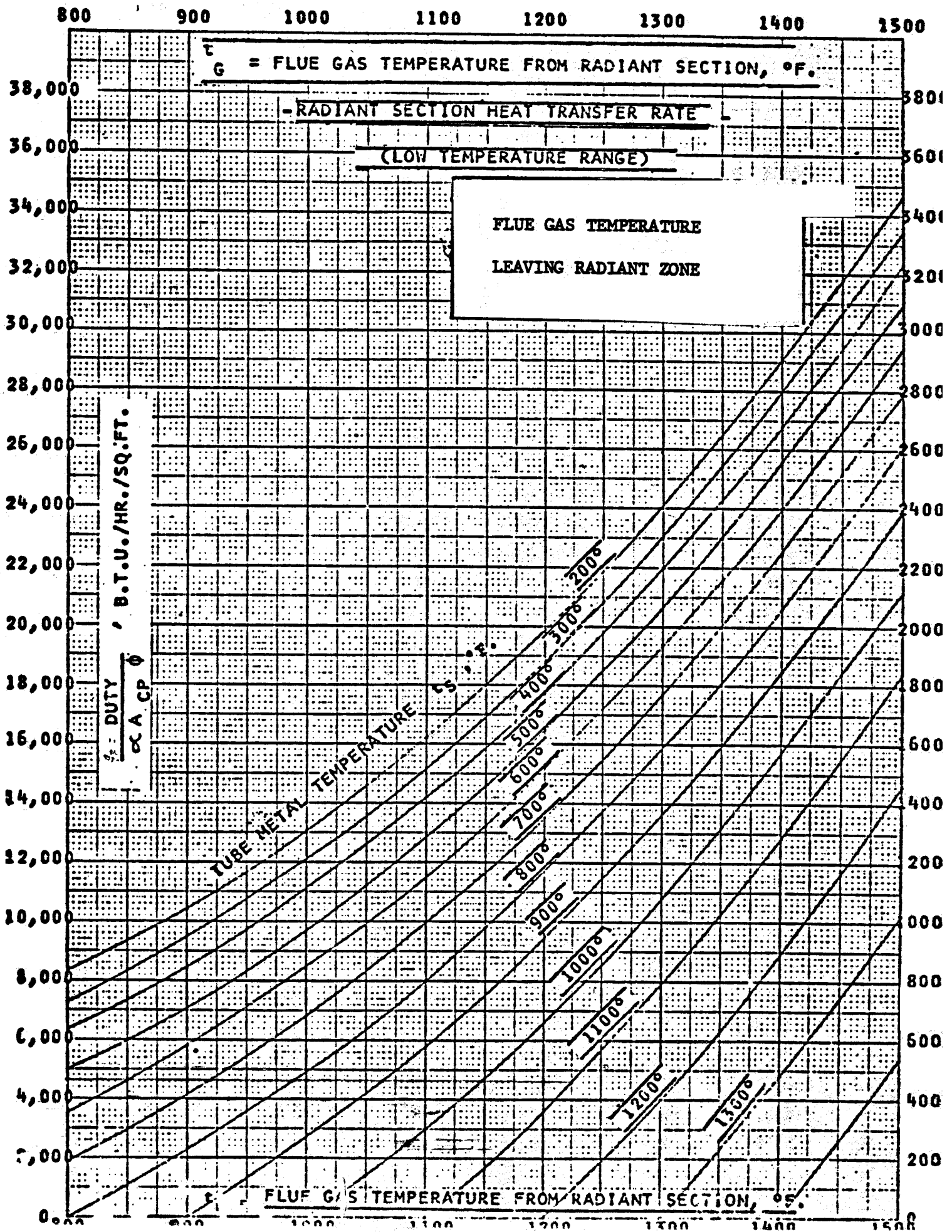


FIGURE 25b

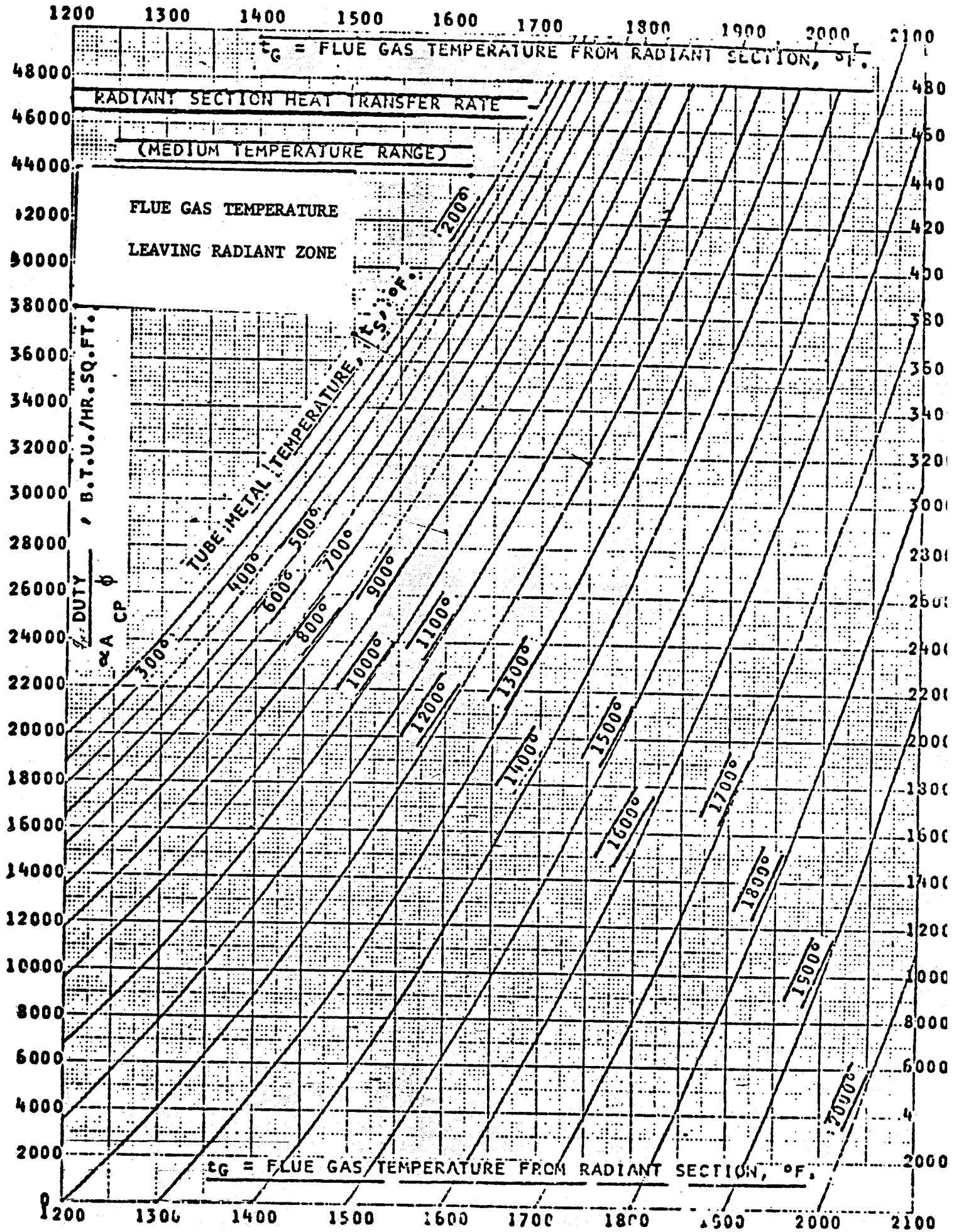


FIGURE 25c

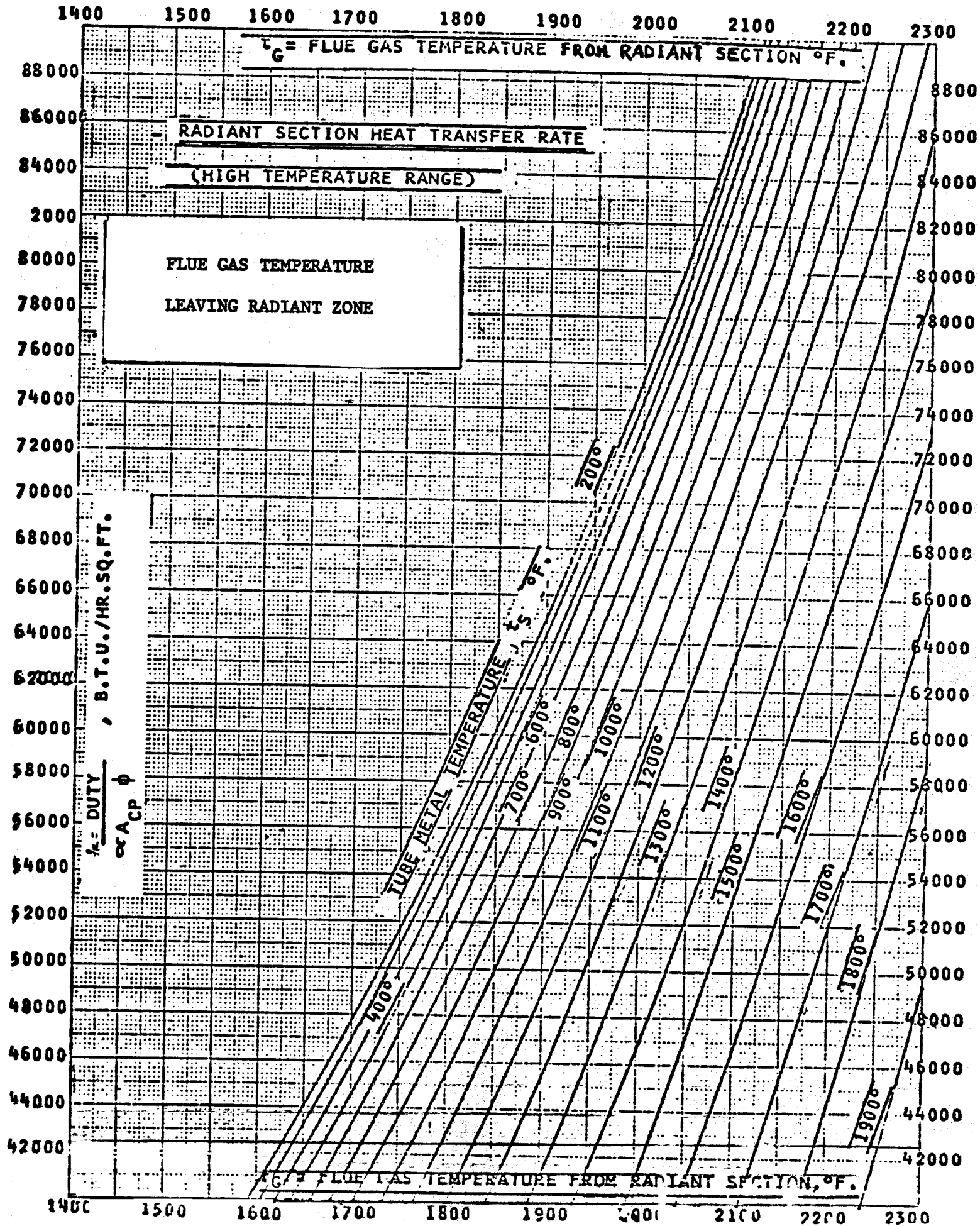
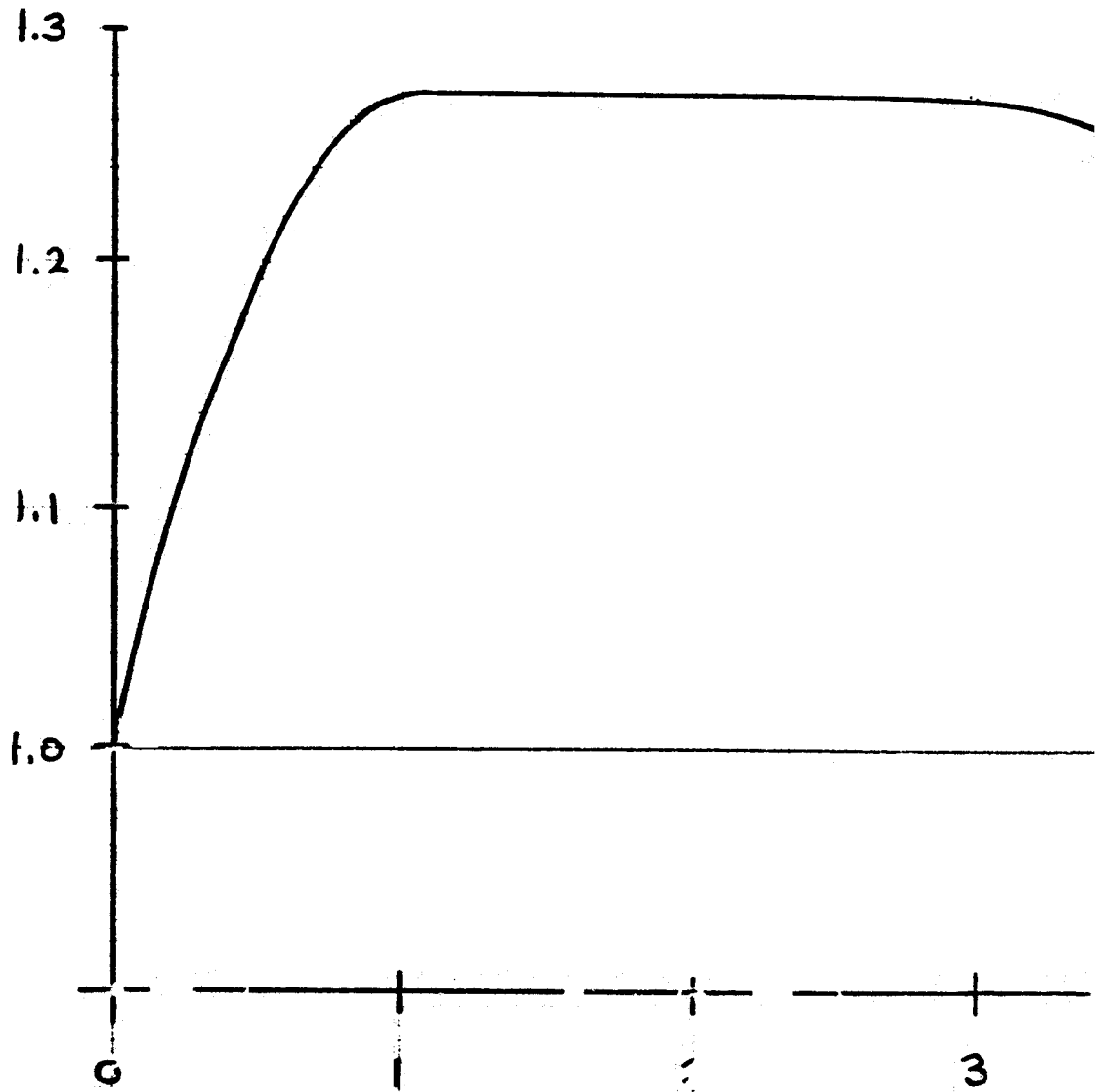


Figure 25d

Ratio of Heat Transfer Rates
in Radiant Section
between a tube X distance from
a wall and one against a wall



Distance From Wall
(tube Diameters)

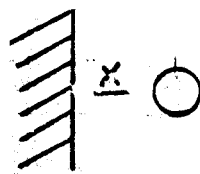
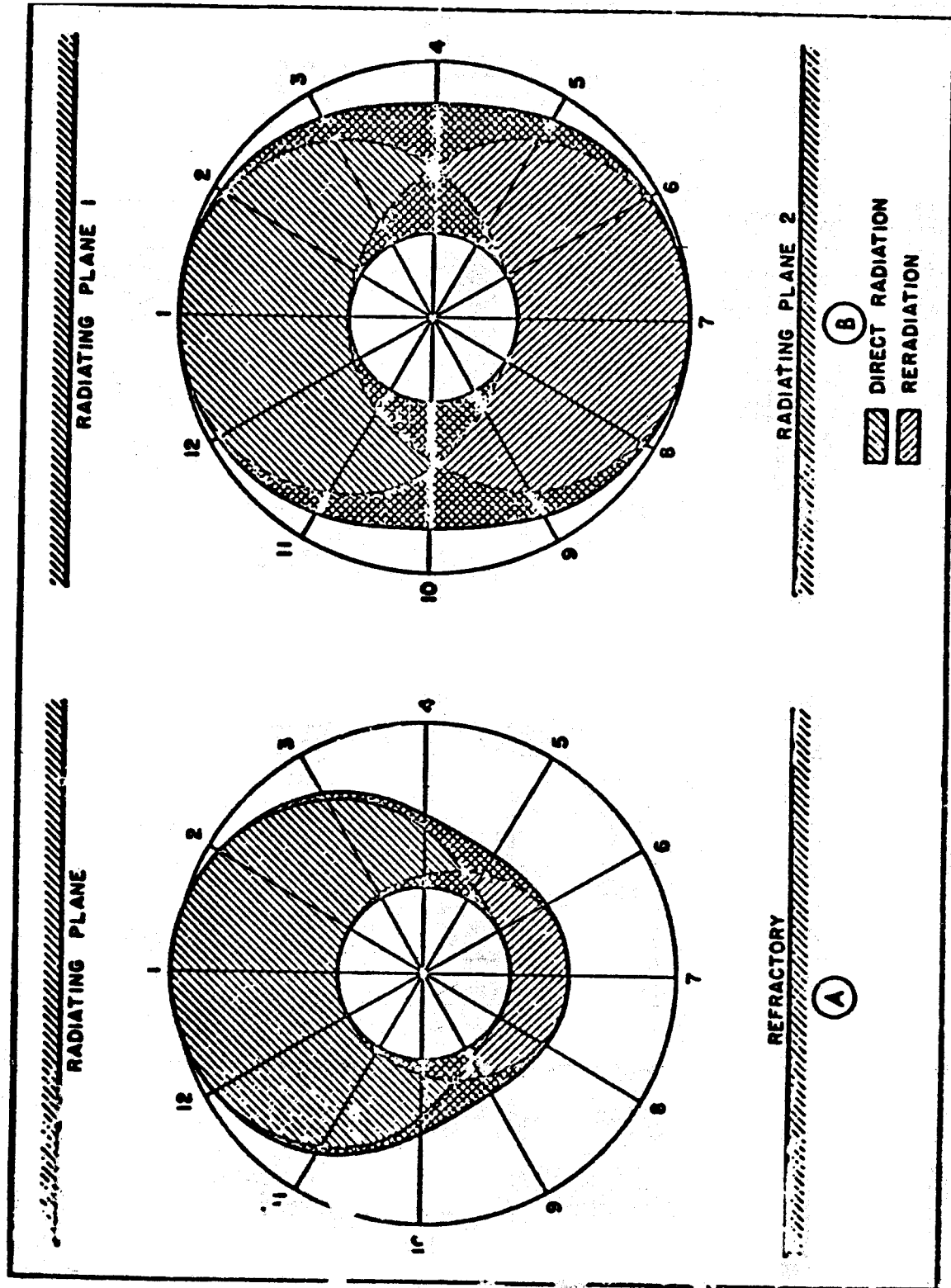
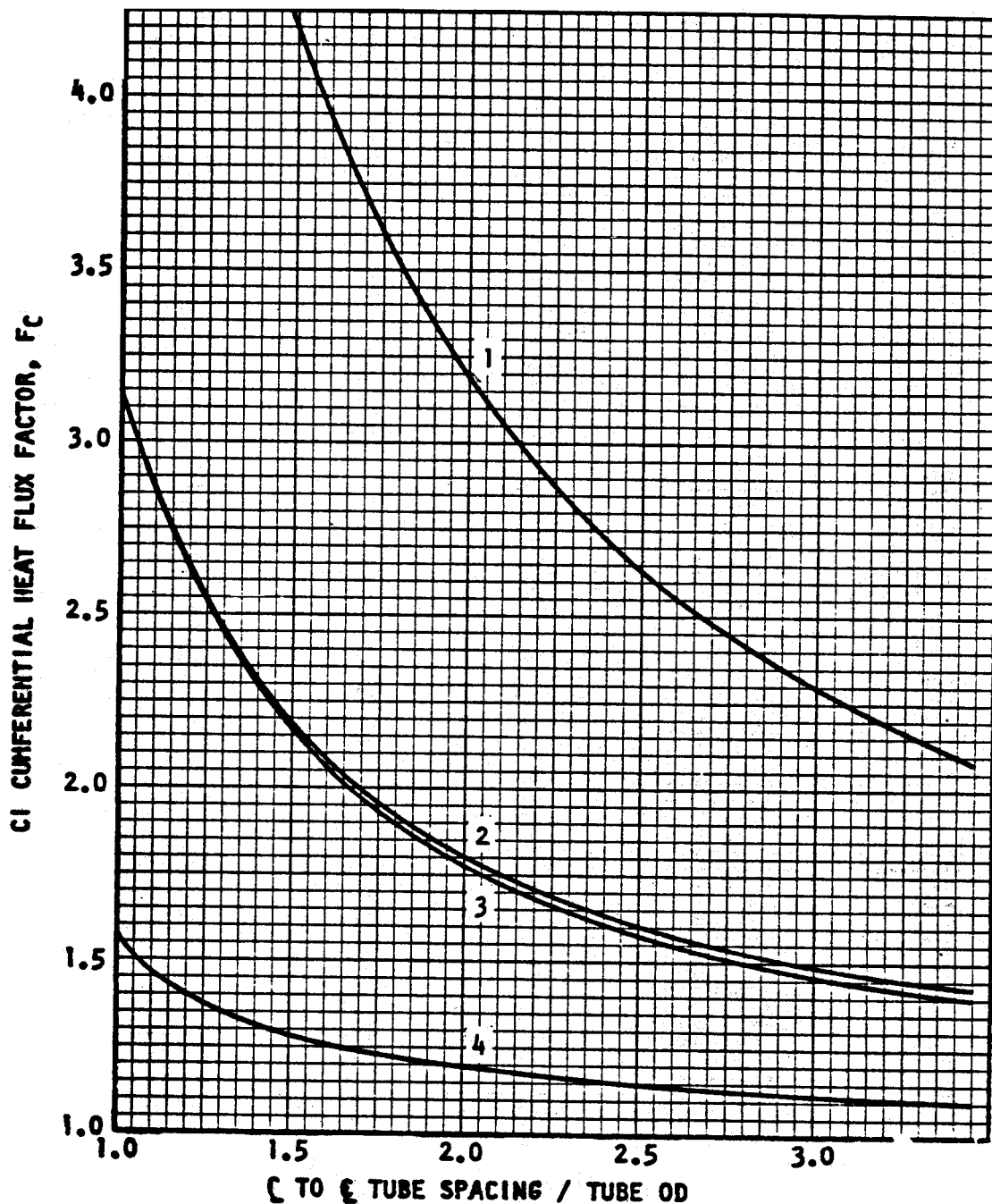


Figure 25e



Flux Distribution around Circumference of Tubes in Single Row Banks. A—Primary Radiating Plane from One Side,—Refractory from the Other. B—Primary Radiation Both Sides

Figure 25f



Curves

1. Double row against wall triangular spacing.
2. Double row with equal radiation from both sides. Two diameters between rows (tubes are spaced equilaterally).
3. Single row against wall.
4. Single row with equal radiation from both sides.

Explanatory Notes

1. These curves are valid when used with a tube center to refractory wall spacing of one and one-half times the nominal

tube diameter involved. Any appreciable variation from this spacing must be given special consideration.

2. These curves do not consider convection heat transfer to the tubes, circumferential heat transfer by conduction in the tube wall, or variations in heat flux in different zones of the radiant section.

3. These curves are based on the work of H. C. Hottel.

Ratio of Maximum Local to Average Heat Flux

FIGURE 26a

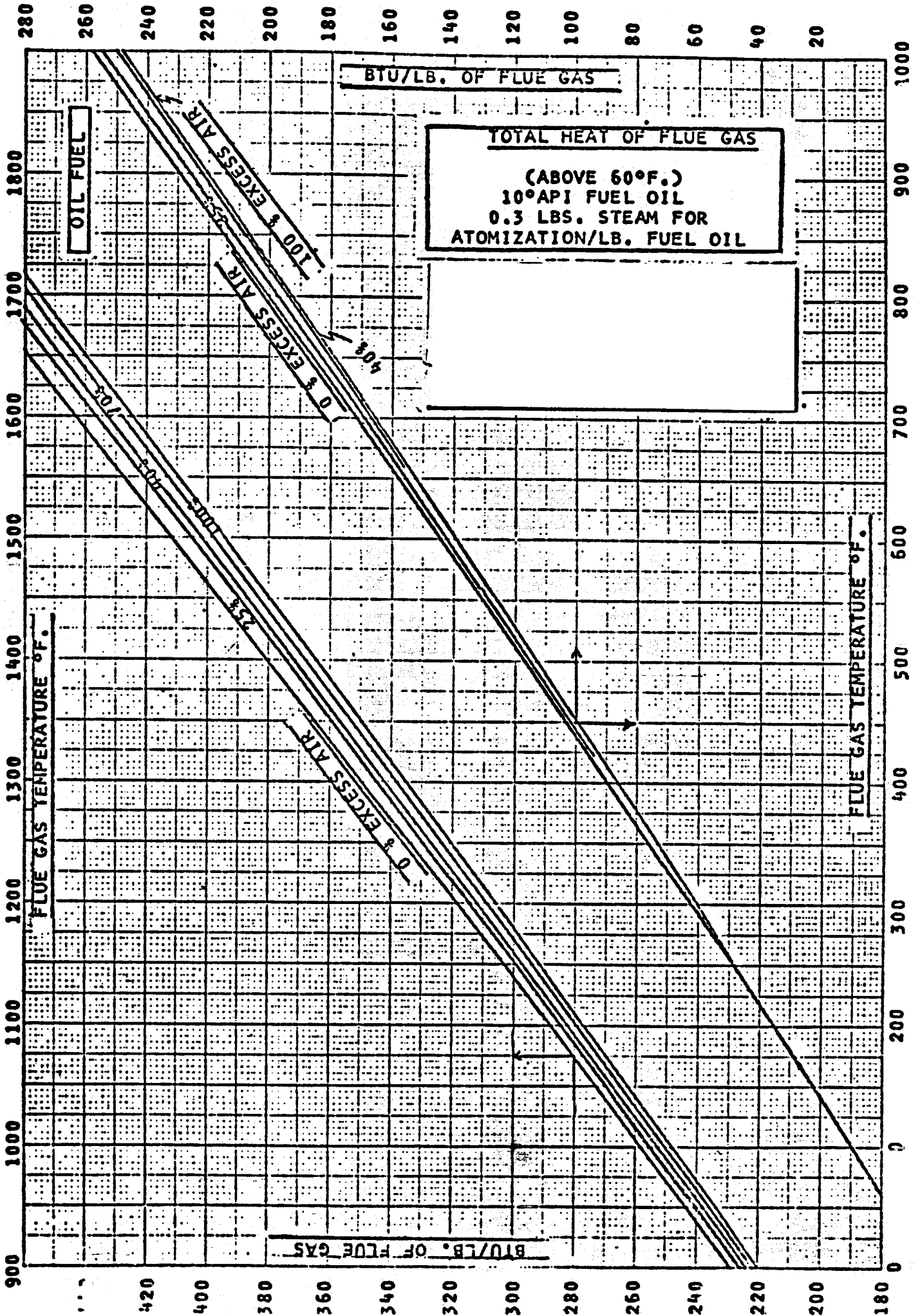


FIGURE 26b

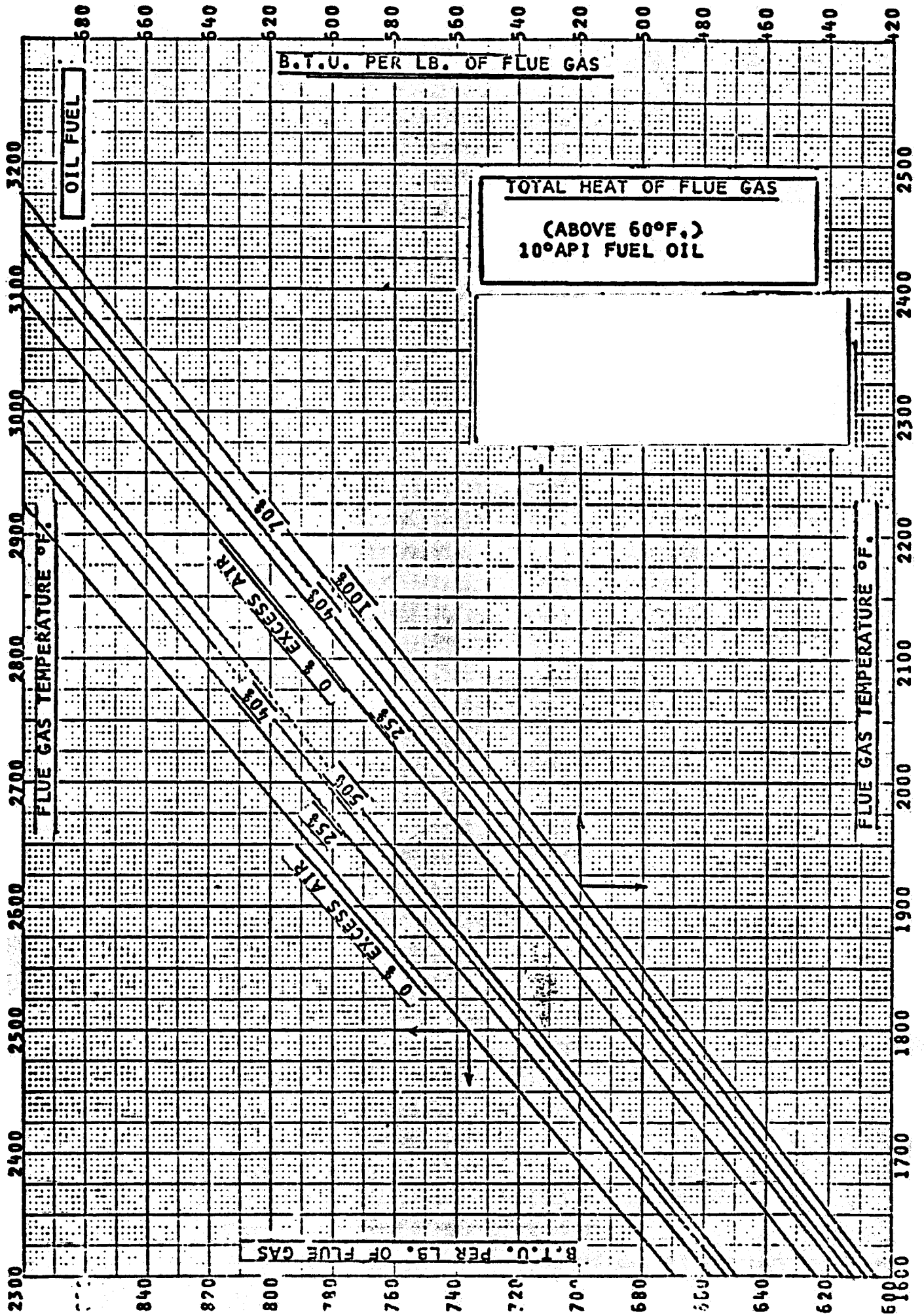


FIGURE 26c

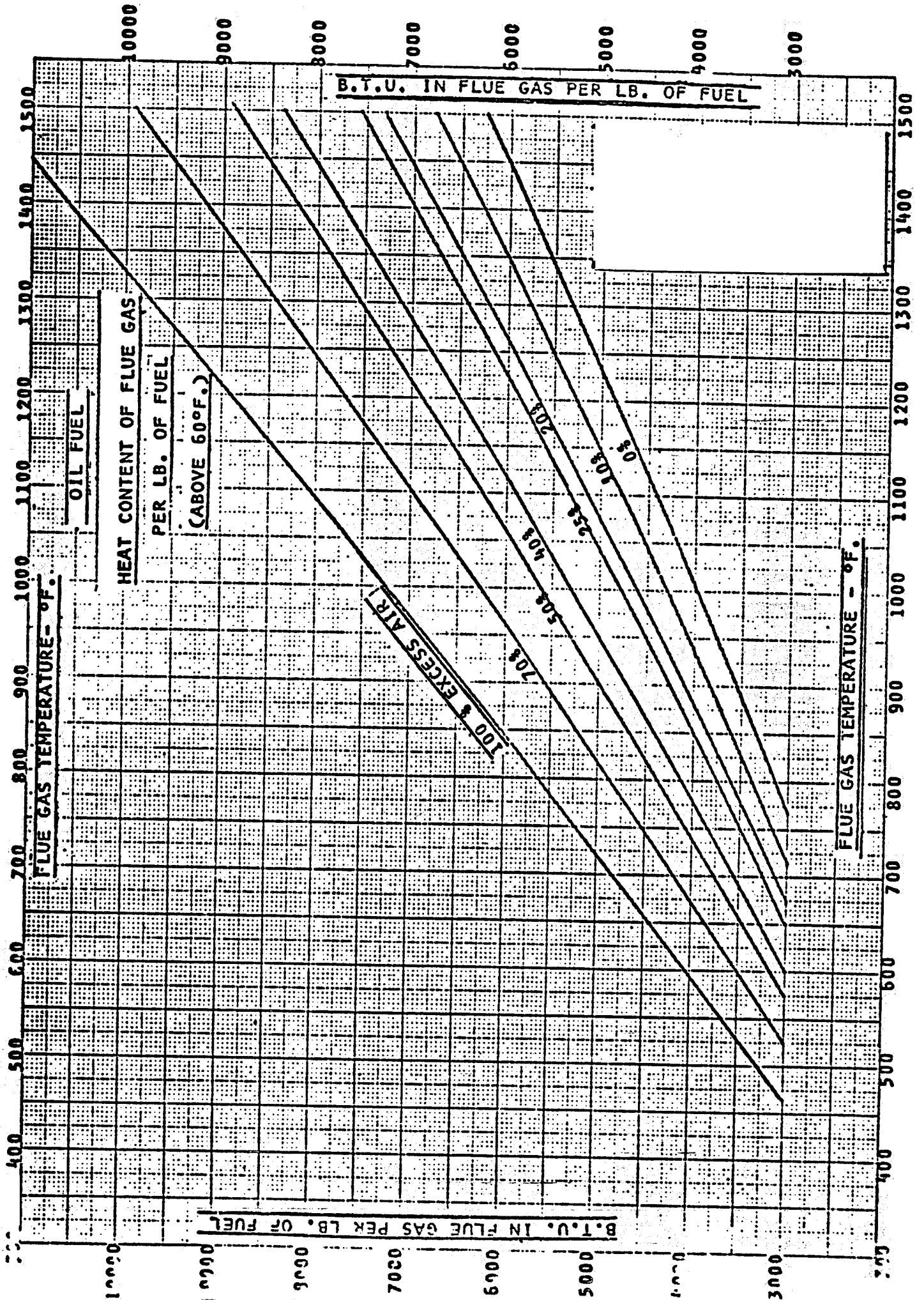


FIGURE 26d

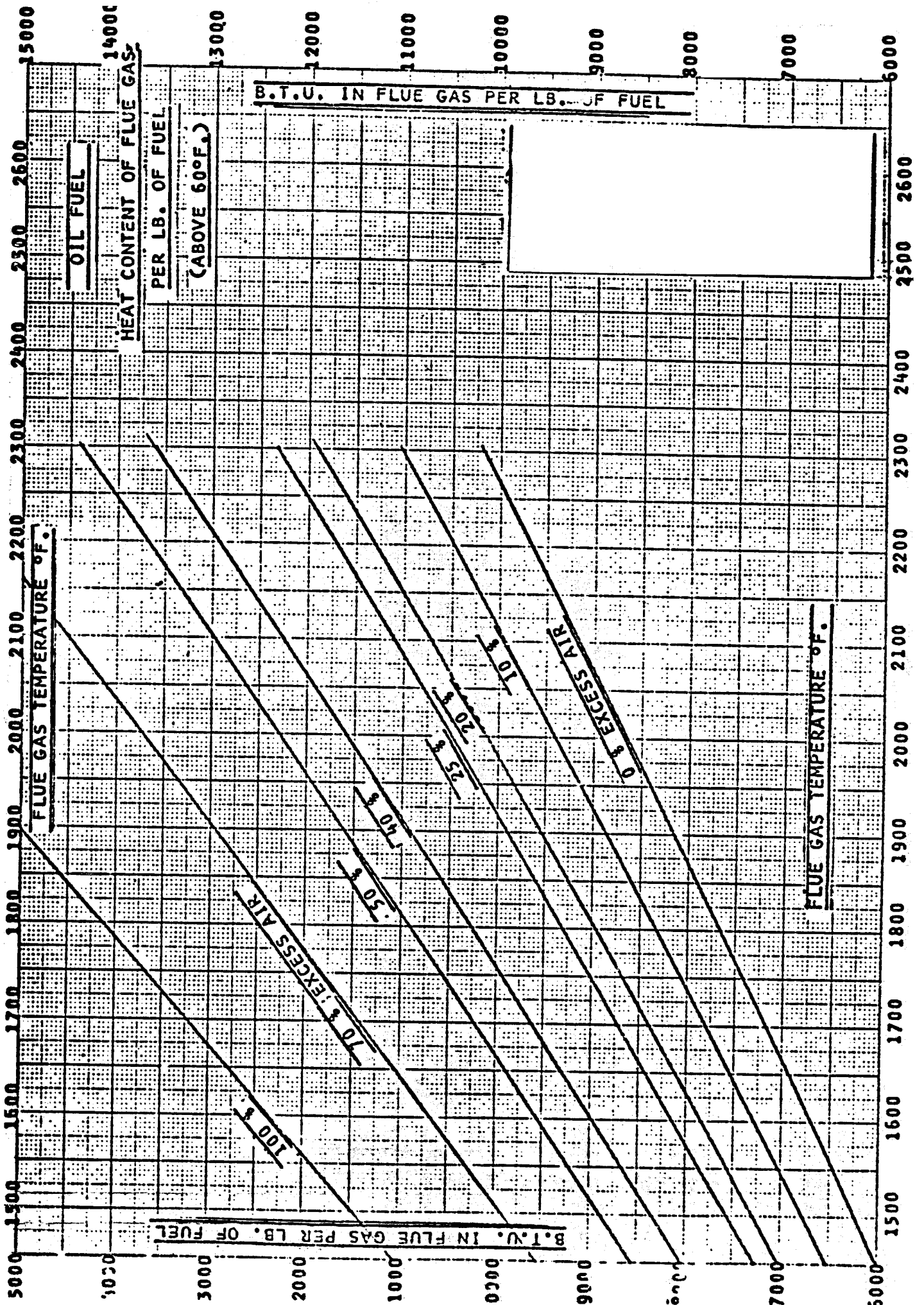


FIGURE 27

TOTAL FLUEGAS, LB/MMBTU NET RELEASE

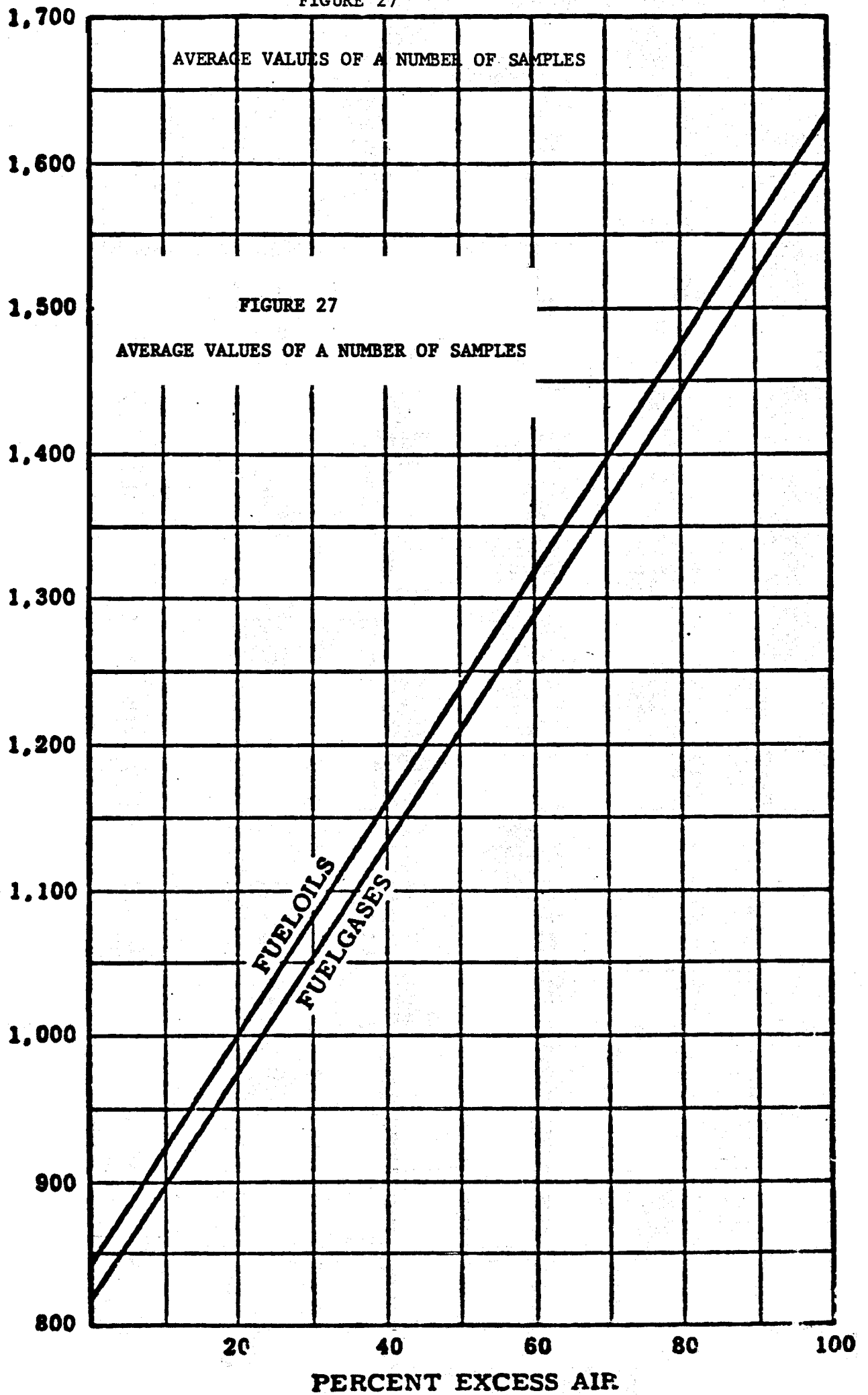
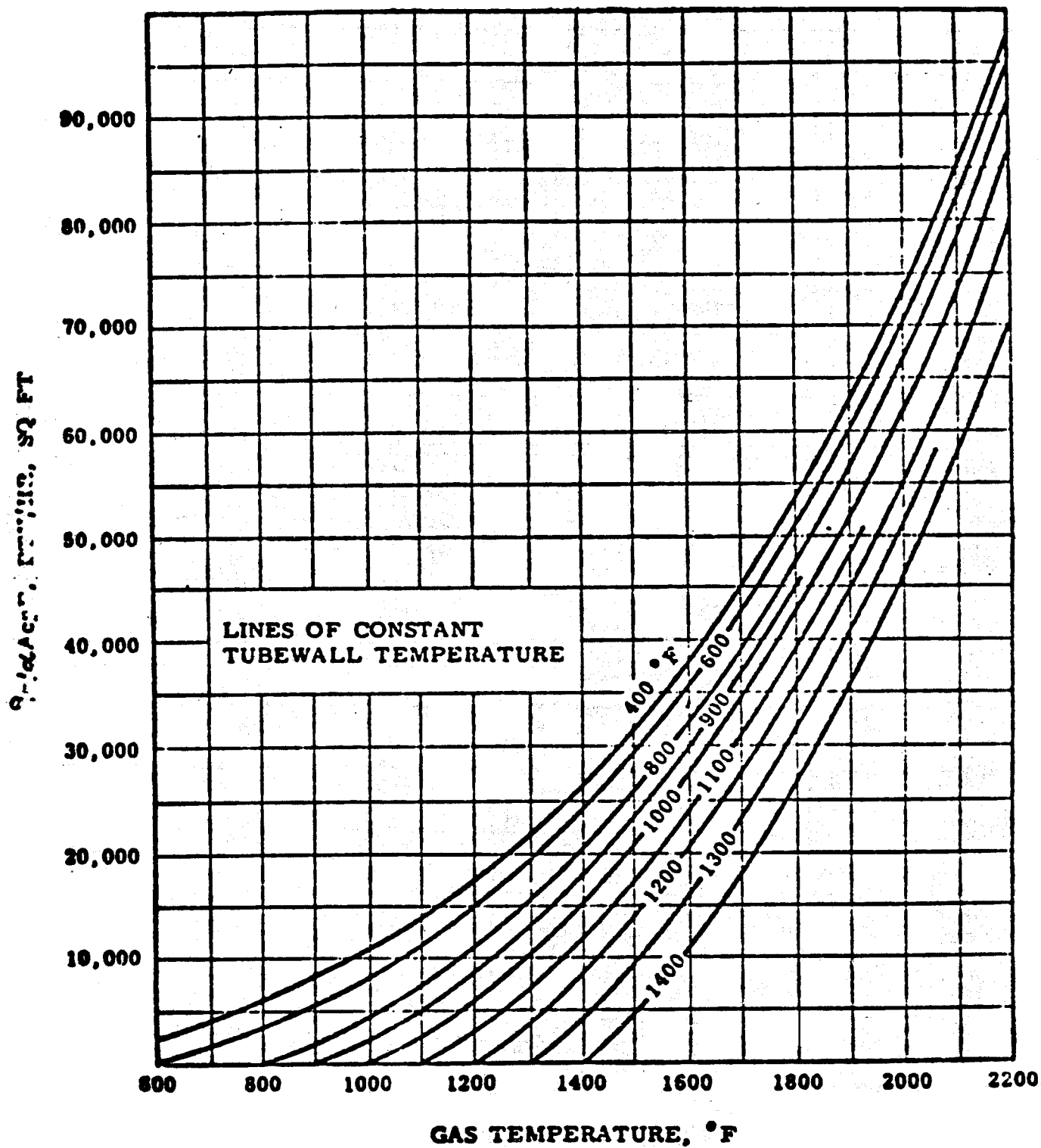


FIGURE 28



Total heat-absorption in the radiant section.

FIGURE 29

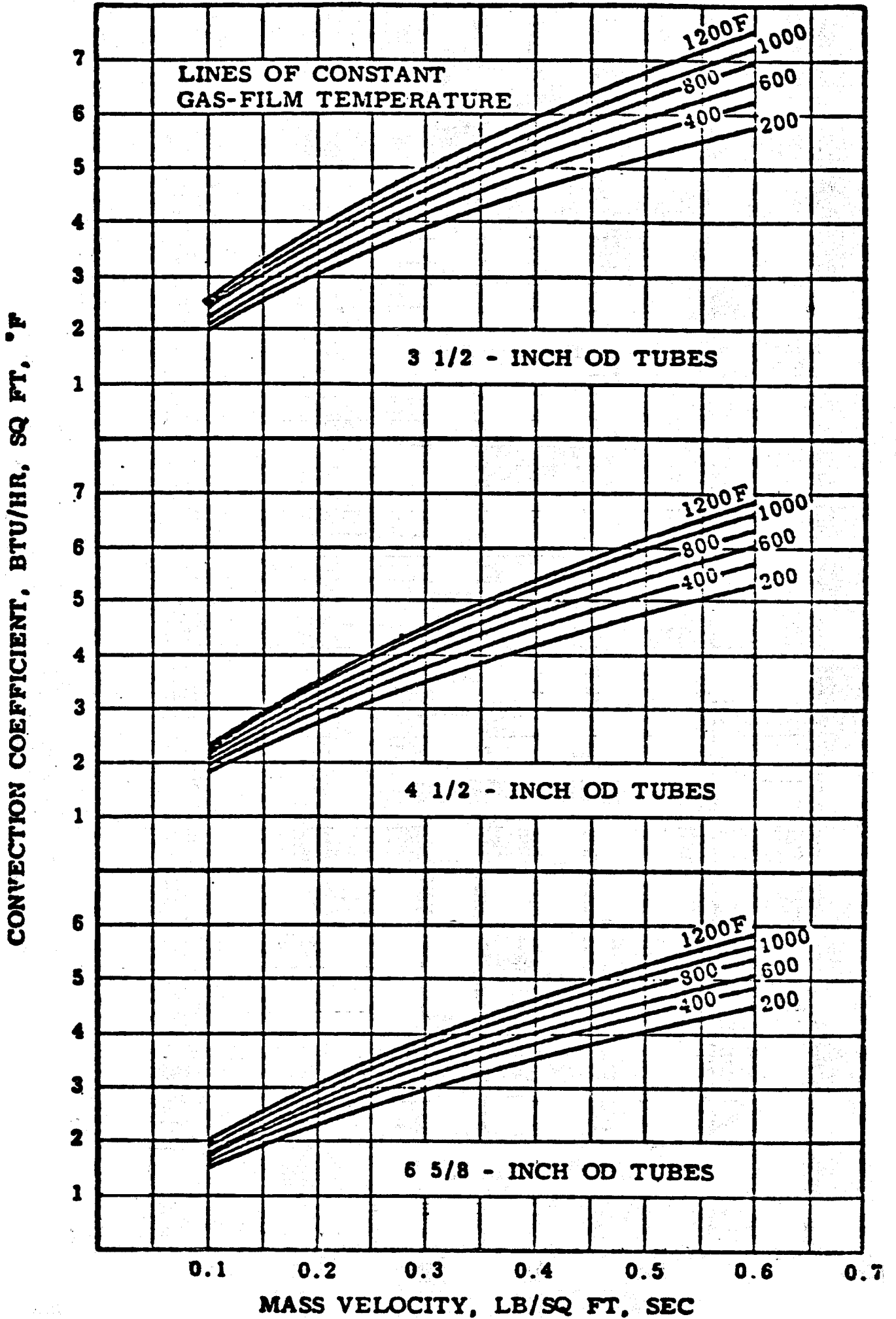


FIGURE 30

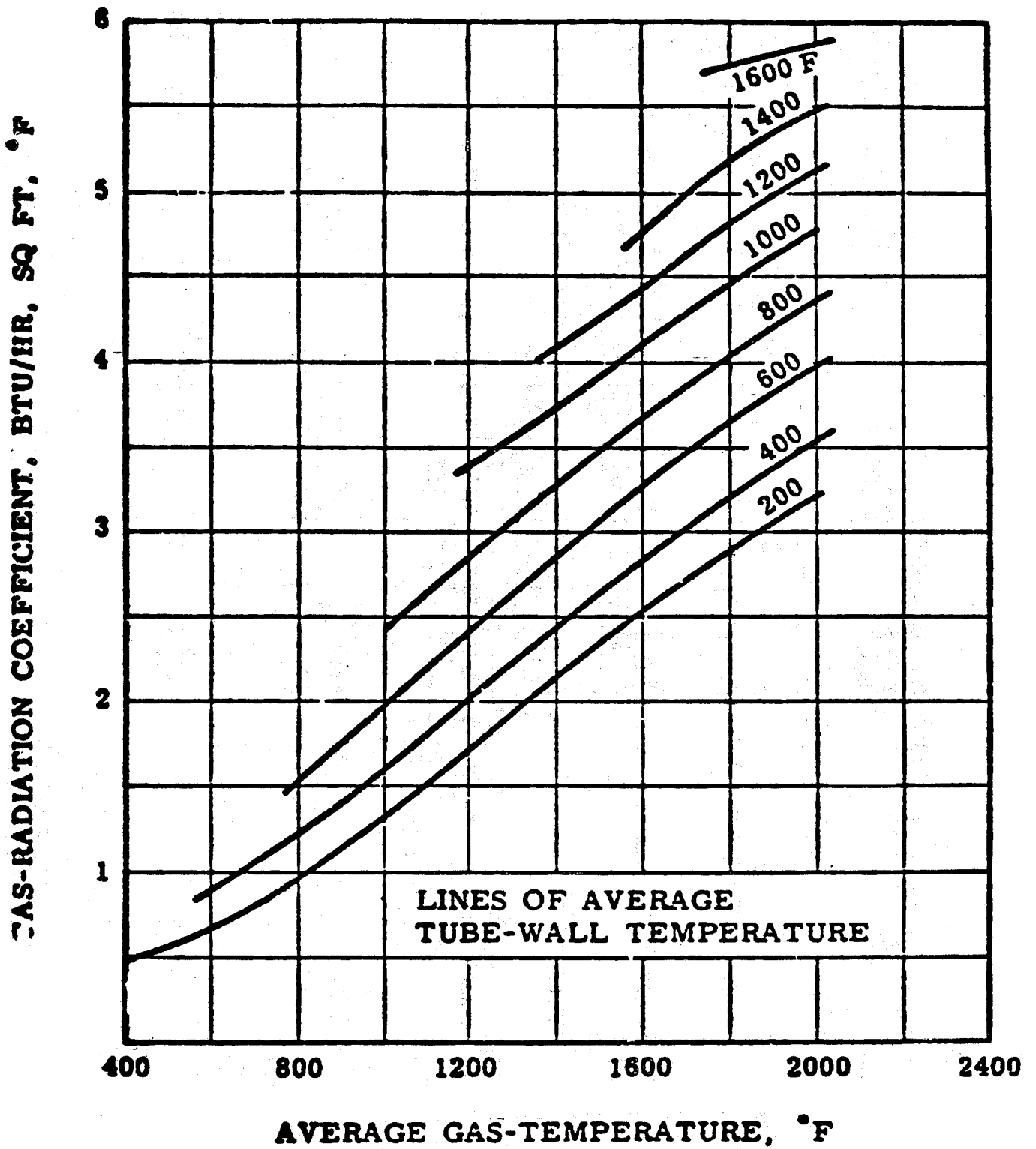


FIGURE 31

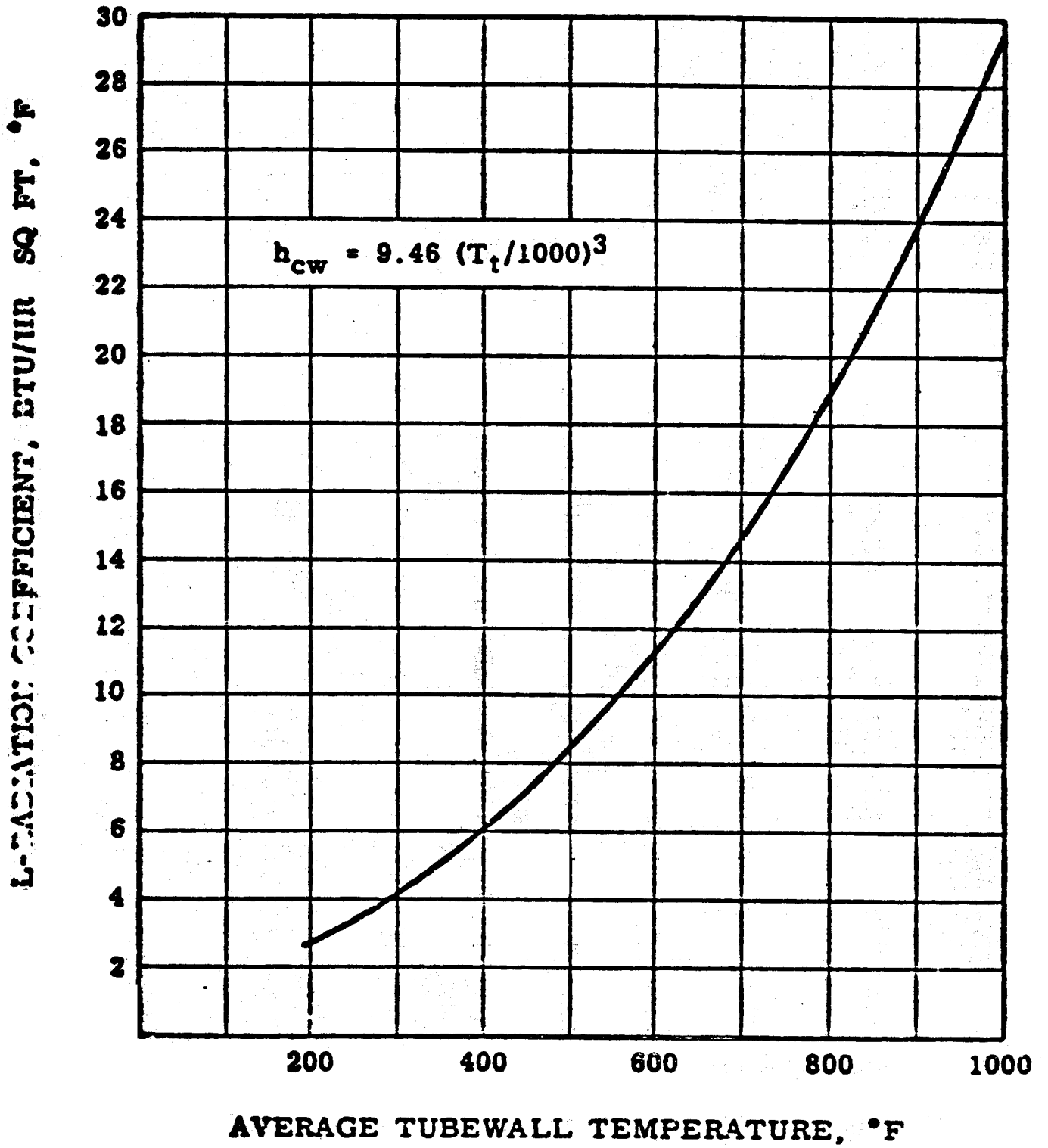


FIGURE 32

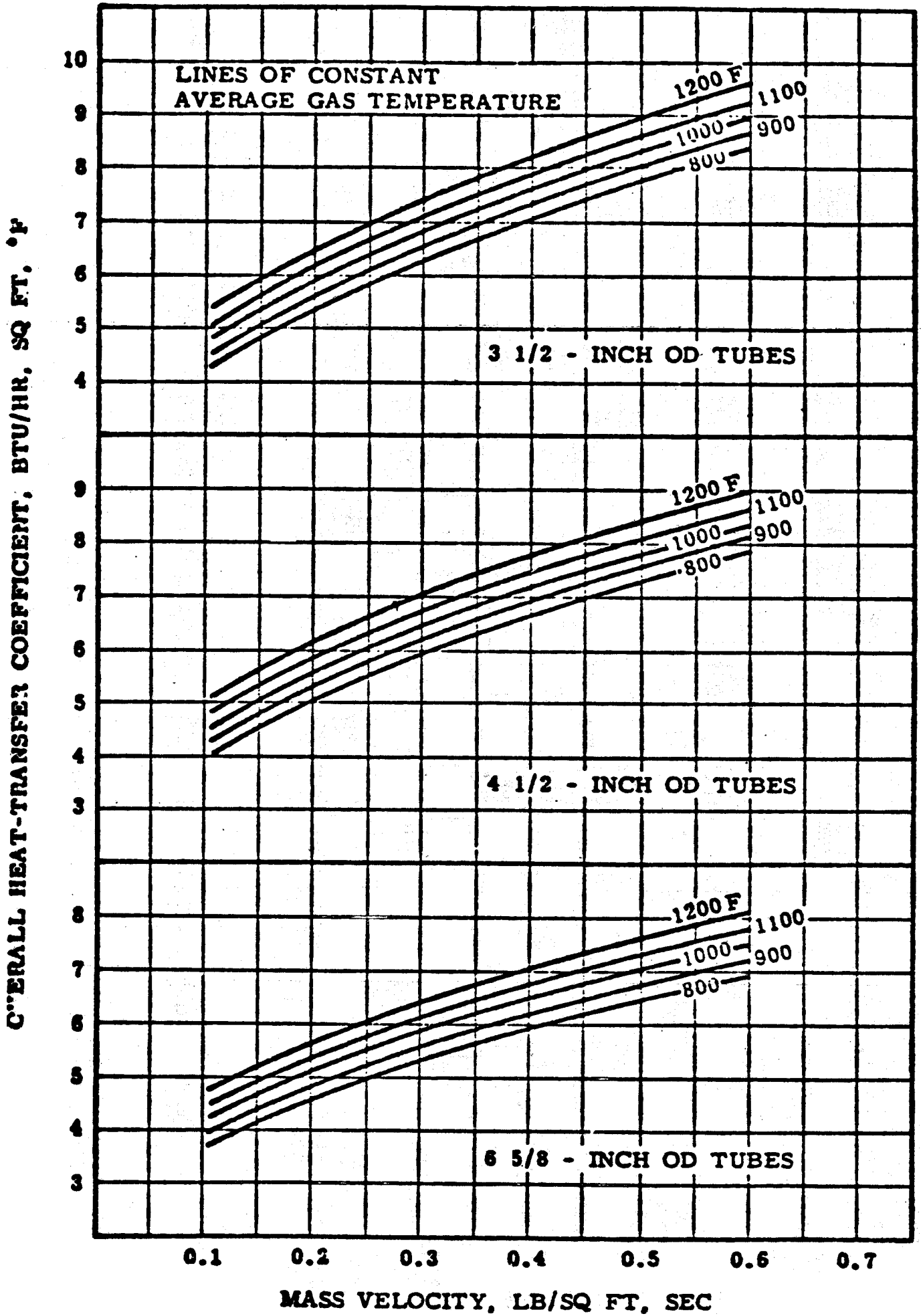


FIGURE 33

DENSITY OF FLUE GAS

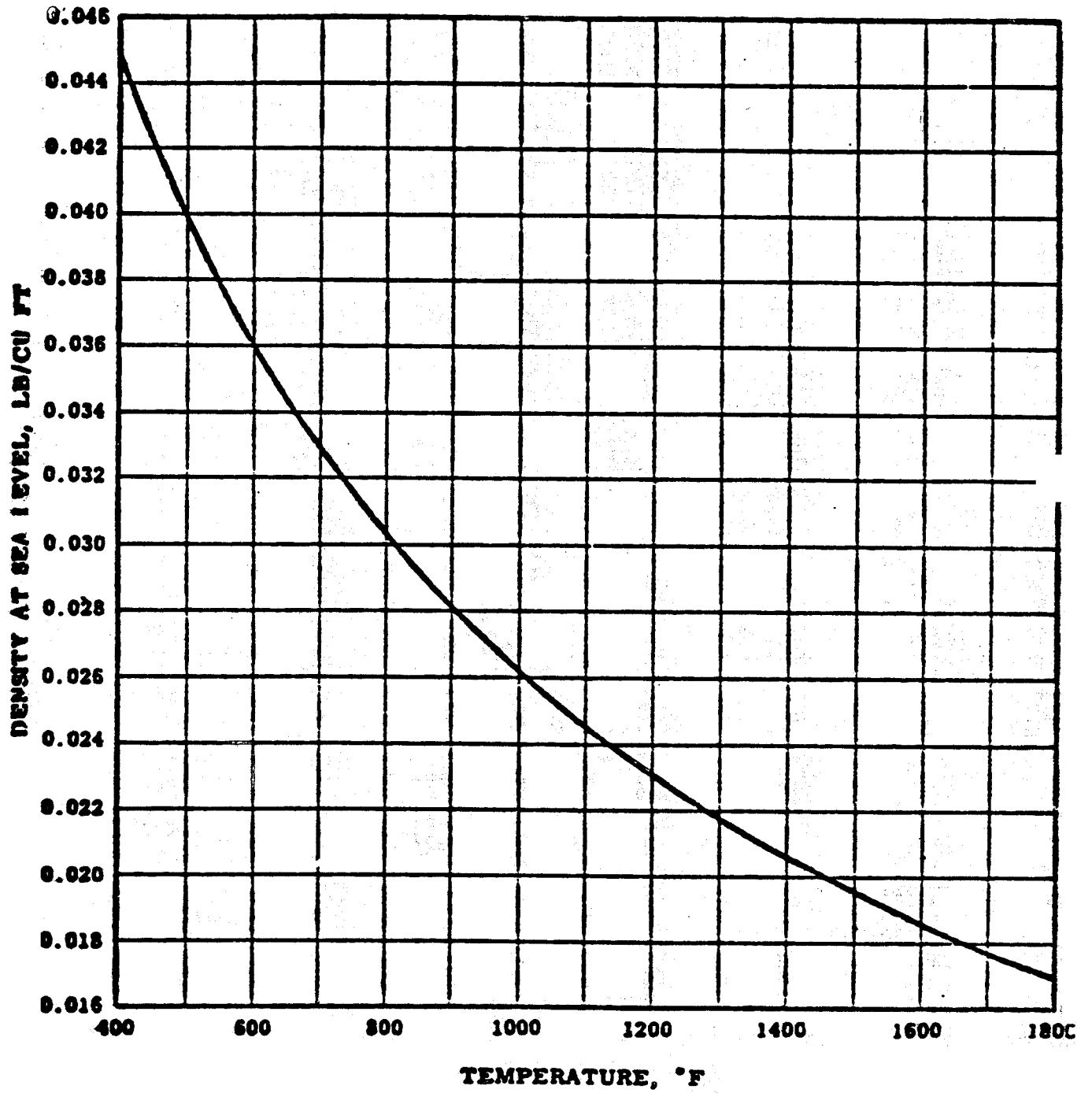


FIGURE 34

STACK DRAFT AT SEA LEVEL

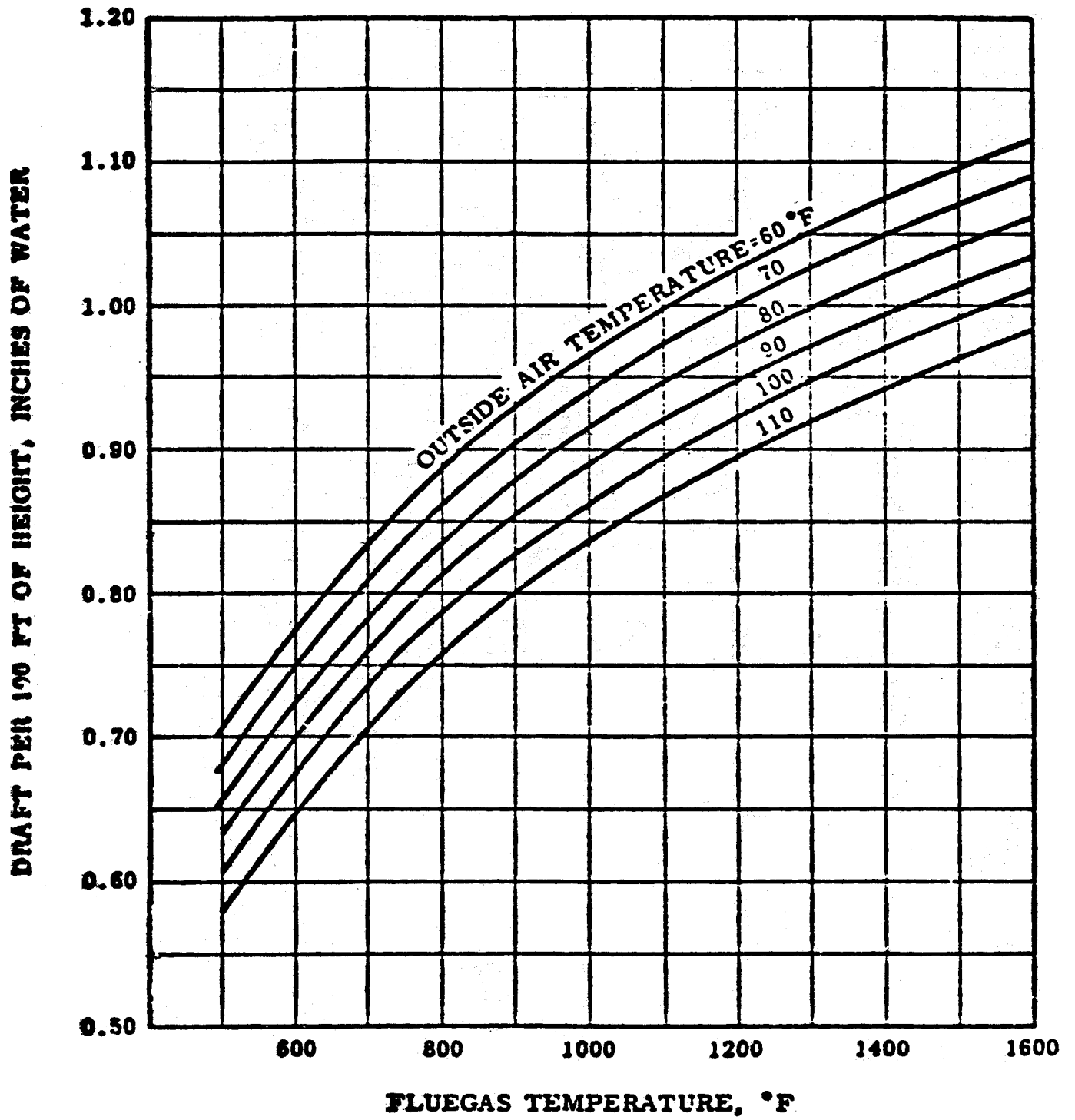


TABLE 2

FUEL OIL

FURNACE EFFICIENCY

WITHOUT AIR PREHEATER

FLUE GAS TEMP. °F	B.T.U. PER LB. FLUE GAS	B.T.U. PER LB FUEL	FURNACE EFFICIENCY PER CENT
40% EXCESS AIR - 20.83 LB. FLUE GAS PER LB. FUEL			
500	114.1	2377	83.12
550	127.7	2660	81.47
600	141.4	2945	79.81
650	155.1	3231	78.14
700	168.8	3516	76.47
750	182.6	3804	74.79
800	196.4	4091	73.12
850	210.5	4385	71.40
900	224.5	4676	69.70
950	238.7	4972	67.97
1000	253.0	5270	66.23
1050	267.3	5568	64.50
1100	281.7	5868	62.74
1150	296.3	6172	60.97
1200	311.0	6478	59.18
1250	325.9	6788	57.37
1300	340.9	7101	55.55
1350	355.9	7413	53.72
1400	370.9	7726	51.90
1450	386.0	8040	50.06
1500	401.3	8359	48.20

 BASIS

FUEL: 10° A.P.I. FUEL OIL. 17,130 B.T.U. PER LB. LOWER HEATING VALUE.

NO AIR PREHEATER: AIR AT 50% RELATIVE HUMIDITY AND 60° FAHR.

RADIATION LOSS: 3.00% OF TOTAL HEAT LIBERATED BY FUEL.

STEAM FOR ATOMIZATION OF FUEL: 0.30 LB. PER LB. OF FUEL.

FUEL OIL ANALYSIS

CARBON 85.0% BY WEIGHT
HYDROGEN 12.0% " "
SULPHUR 1.5% " "

OXYGEN 0.7% BY WEIGHT
NITROGEN 0.8% " "

HEATER EFFICIENCY % VS FLUID INLET TEMPERATURE

FIGURE 35

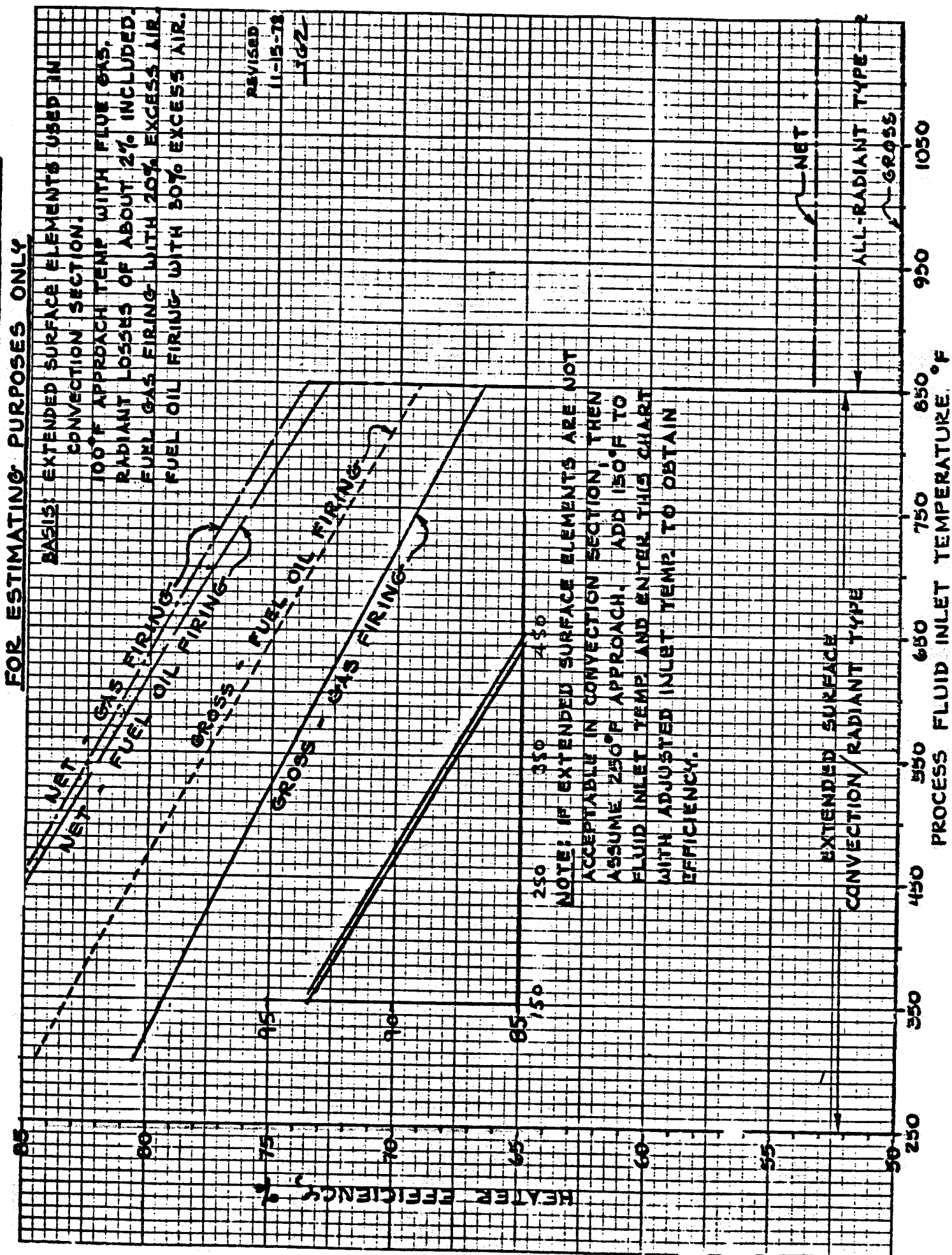


FIGURE 36

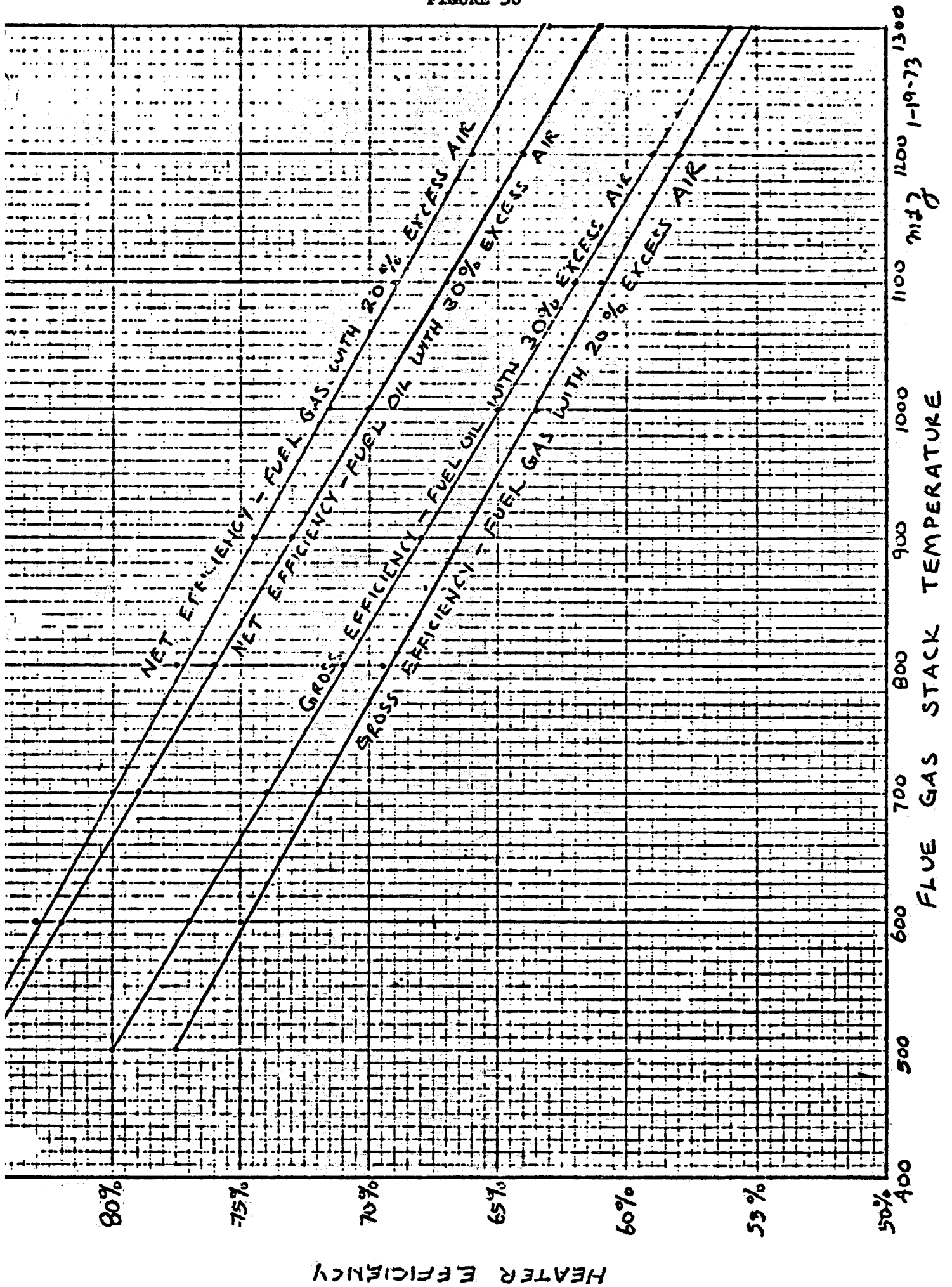


FIGURE 37

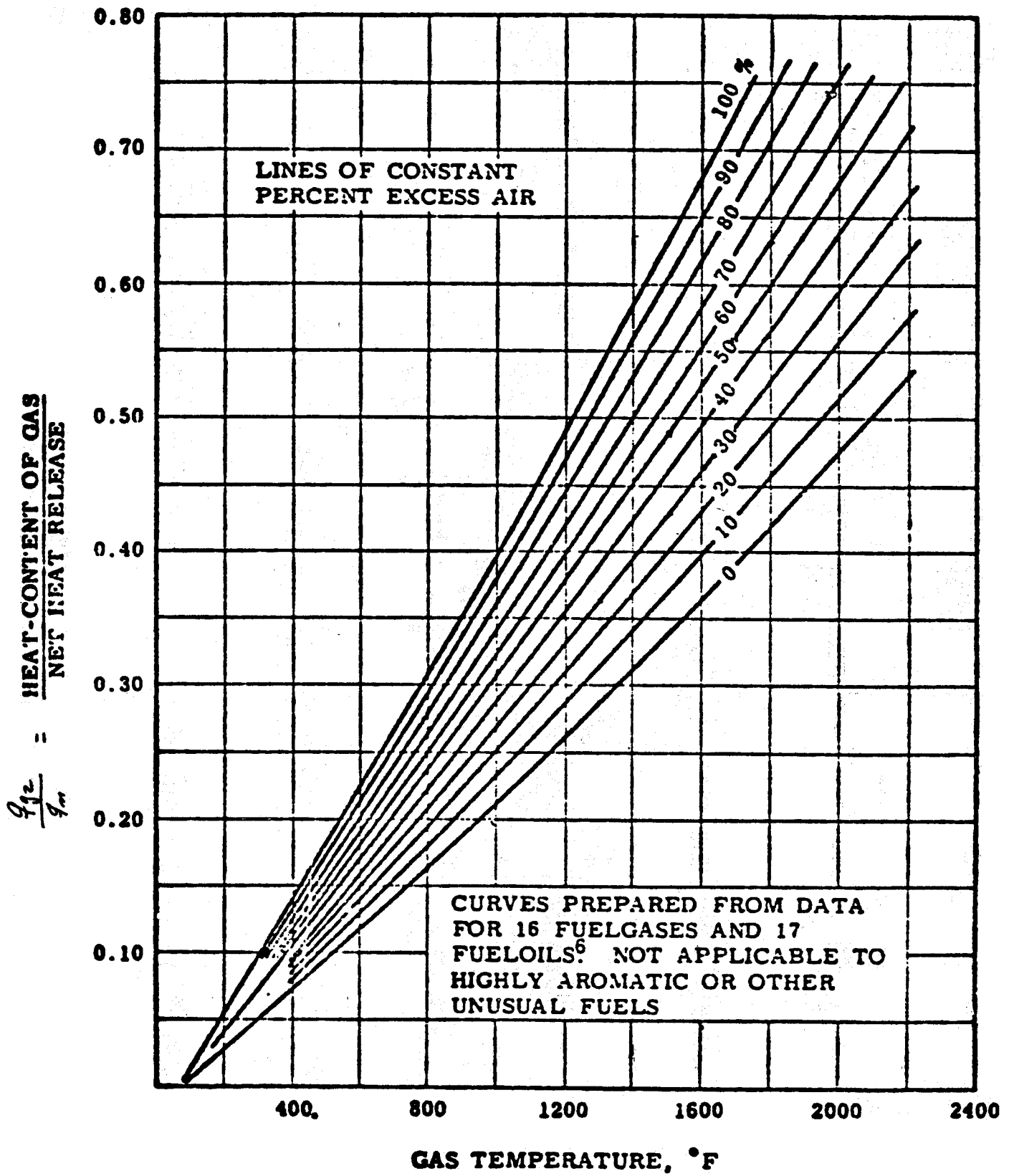


FIGURE 38a

LICENSEES' TRAINING COURSE

HEATER DUTIES

	<u>Crude Heater</u>	<u>Stabilizer Reboiler</u>	<u>Heater</u>
		<u>Summer Operation</u>	<u>Winter Operation</u>
Charge to Heater:	Crude	Gasoline	Gasoline
BPSD	99,900	69,425	88,200
Lb/Hr	1,240,000	744,600	946,700
°API	34.8	60.5	61.5
Wt. % S	0.56	0.13	0.13
Temperatures, °F			
Inlet	490	350	390
Outlet	685	365	410
Outlet Pressure, PSIG	15	150	250
Vaporization at Outlet, Wt. %	55.0	35	35
Vapor, M.W.	131.4	98.8	99.7
Heater Duty, 10 ⁶ Btu/Hr	237.92	37.22	42.52

Assumptions

Fuel Oil Firing @ 30% Excess Air
Stack Flue Gas Approach 300°F
Bare Tube Convection Section

DISTILLATION CHART

CUSTOMER Licensee's Training Course
 SUBJECT Redwater Crude - Flash Curves

PAGE NO. _____
 DATE _____
 PROJECT 6626
 BY JNW

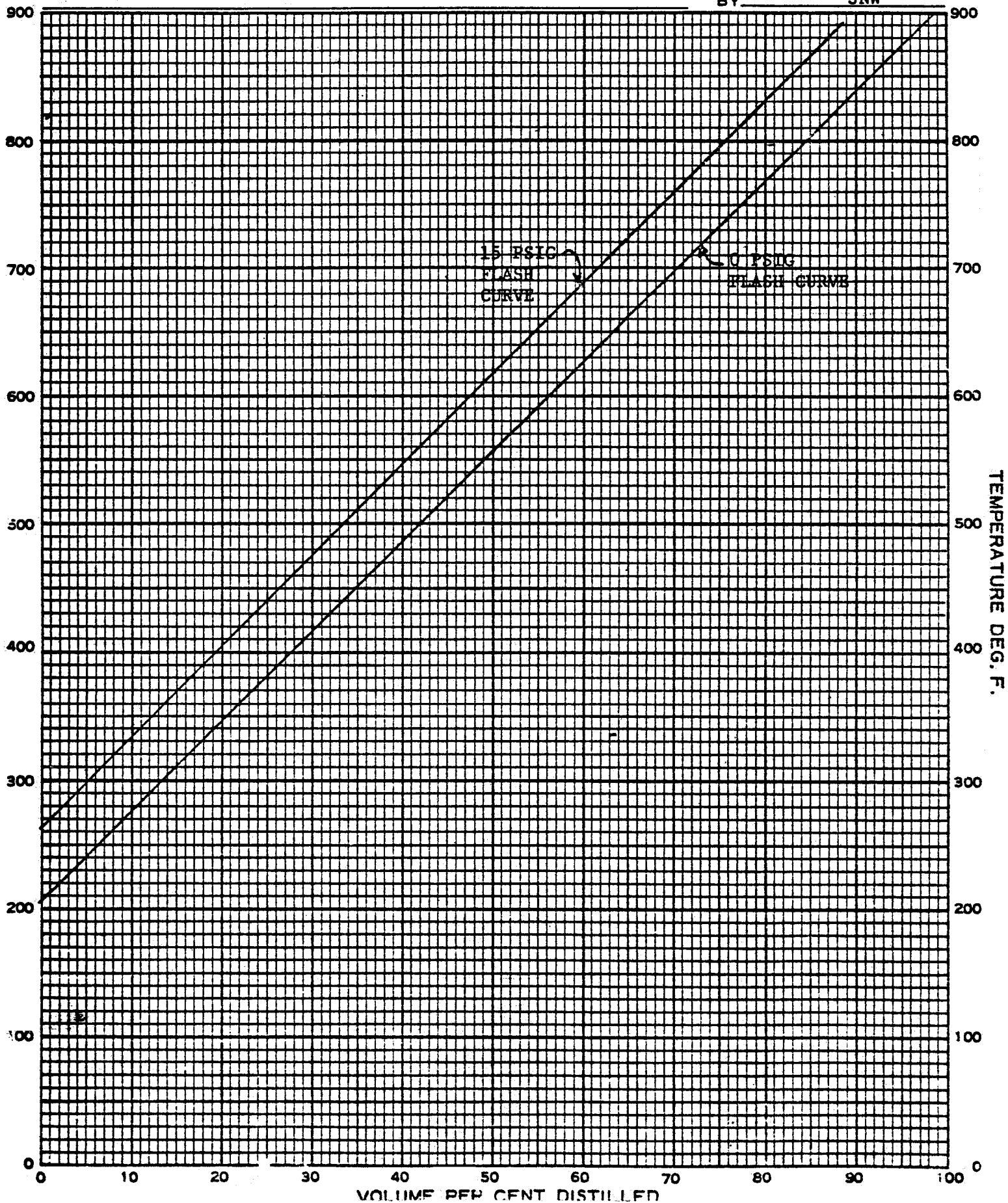


FIGURE 39

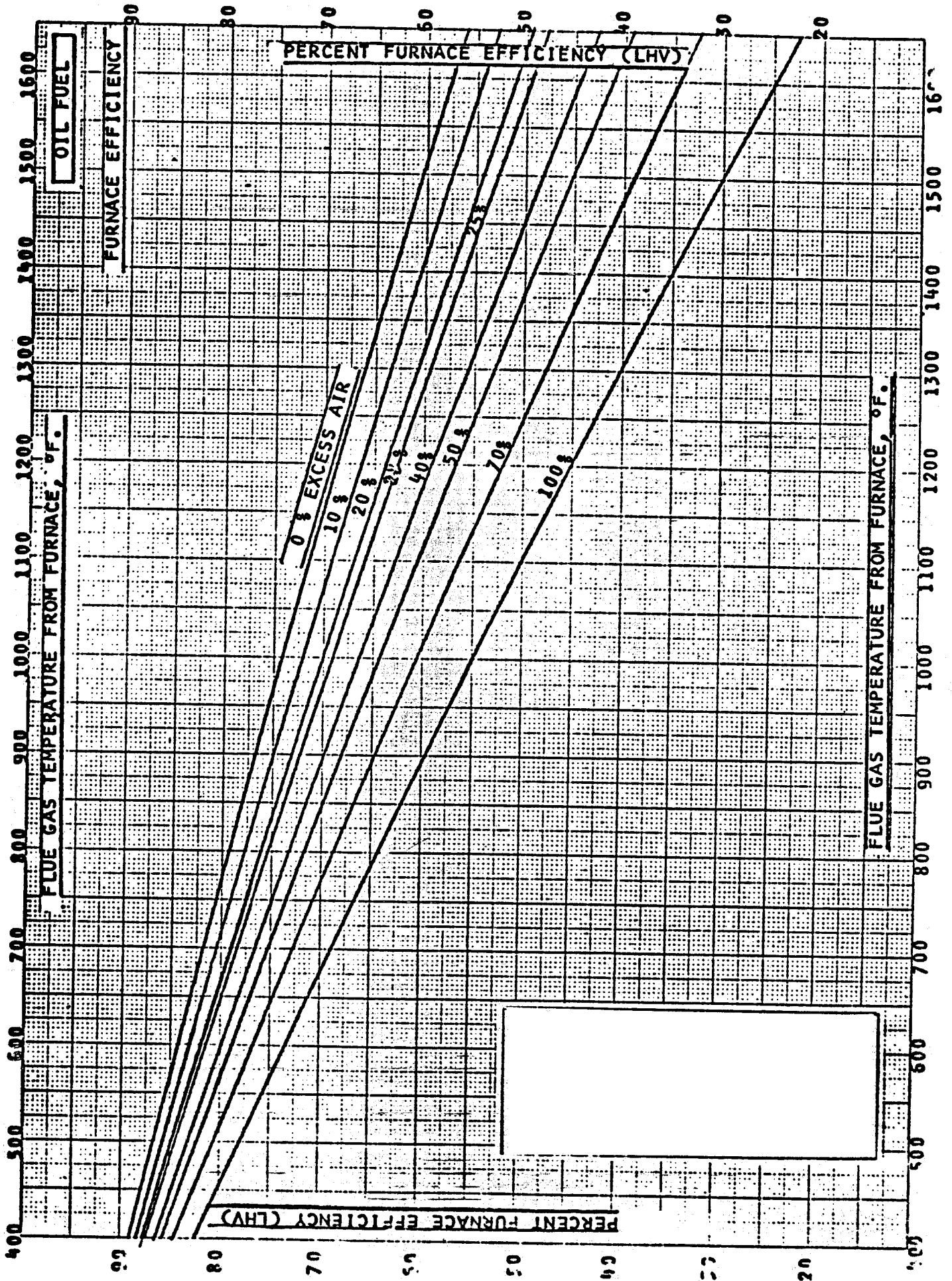


FIGURE 40a

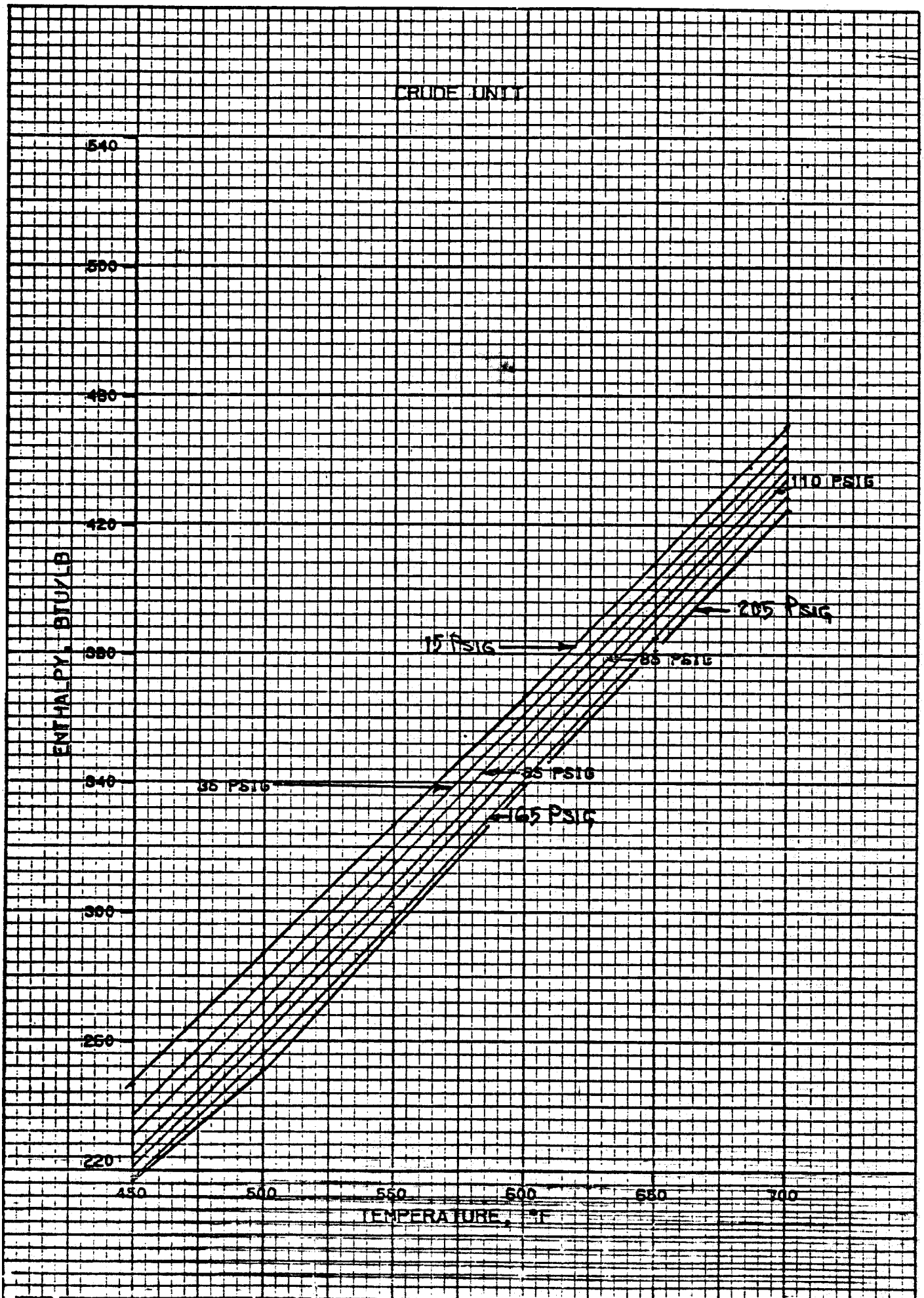


FIGURE 40b

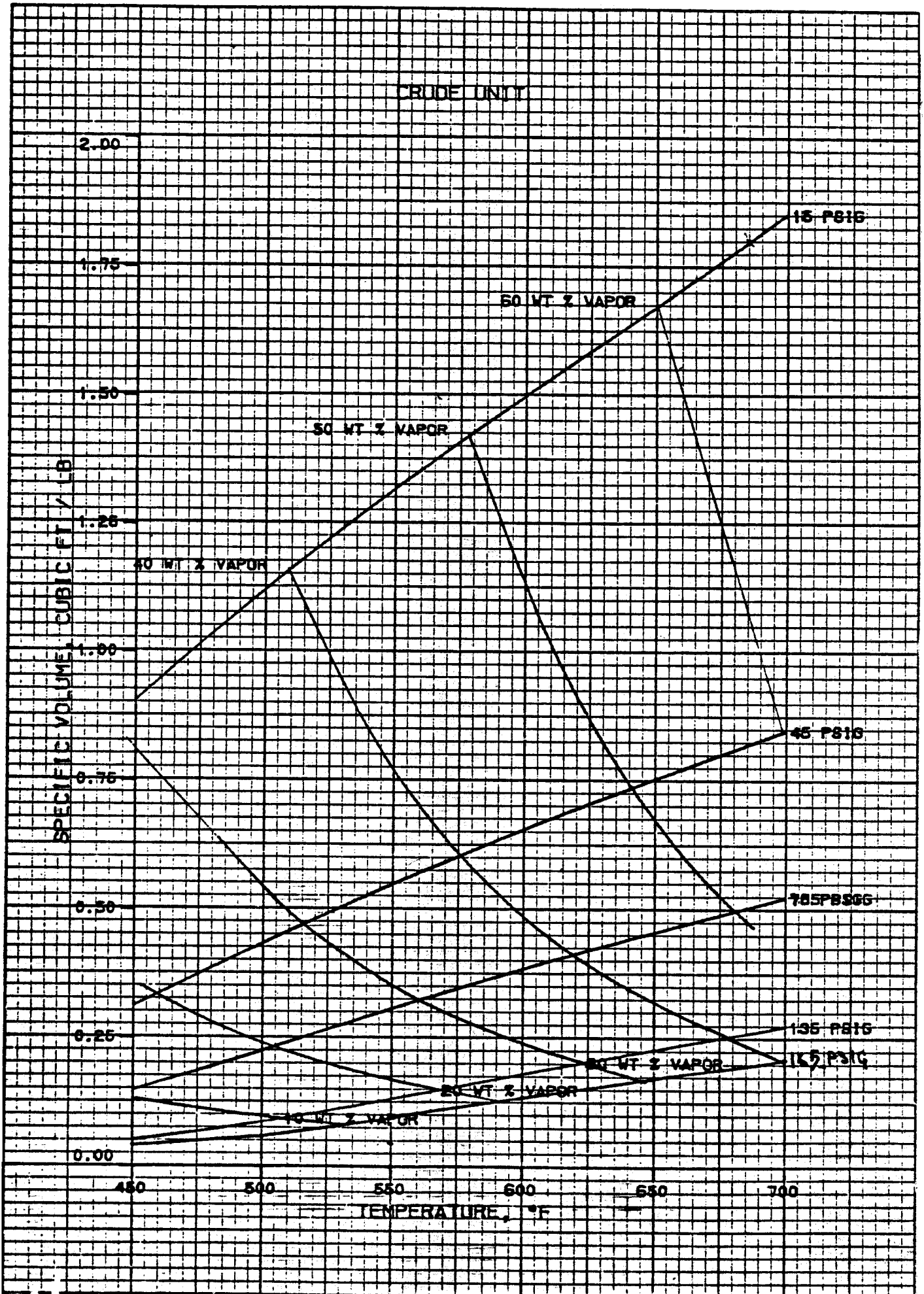


FIGURE 41

AIR-FUEL AND FLUE GAS-FUEL RATIOS

<u>PERCENT EXCESS AIR</u>	<u>FUEL OIL</u>	
	<u>AIR LB. PER LB. FUEL</u>	<u>FLUE GAS LB. PER LB. FUEL</u>
0	13.95	15.25
5	14.65	15.95
10	15.34	16.64
15	16.04	17.34
20	16.74	18.04
25	17.44	18.74
30	18.13	19.43
35	18.83	20.13
40	19.53	20.83
45	20.23	21.53
50	20.92	22.22
55	21.62	22.92
60	22.32	23.62
65	23.02	24.32
70	23.72	25.02
75	24.41	25.71
80	25.11	26.41
85	25.81	27.11
90	26.50	27.80
95	27.20	28.50
100	27.90	29.20
105	28.59	29.89
110	29.29	30.59
115	29.99	31.29
120	30.69	31.99
125	30.38	32.68
130	32.08	33.38
135	32.78	34.08
140	33.48	34.78
145	34.18	35.48
150	34.87	36.17

FUEL OIL ANALYSIS

CARBON	85.0%	BY WEIGHT
HYDROGEN	12.0%	" "
SULPHUR	1.5%	" "
OXYGEN	0.7%	" "
NITROGEN	0.8%	" "

PERFECT COMBUSTION ASSUMED.

STEAM FOR ATOMIZATION OF FUEL: 0.30 LB. PER LB. FUEL

AIR AT 50% RELATIVE HUMIDITY AND 60° FAHR.

LOWER HEATING VALUE = 17,130 Btu/Lb

FIGURE 42

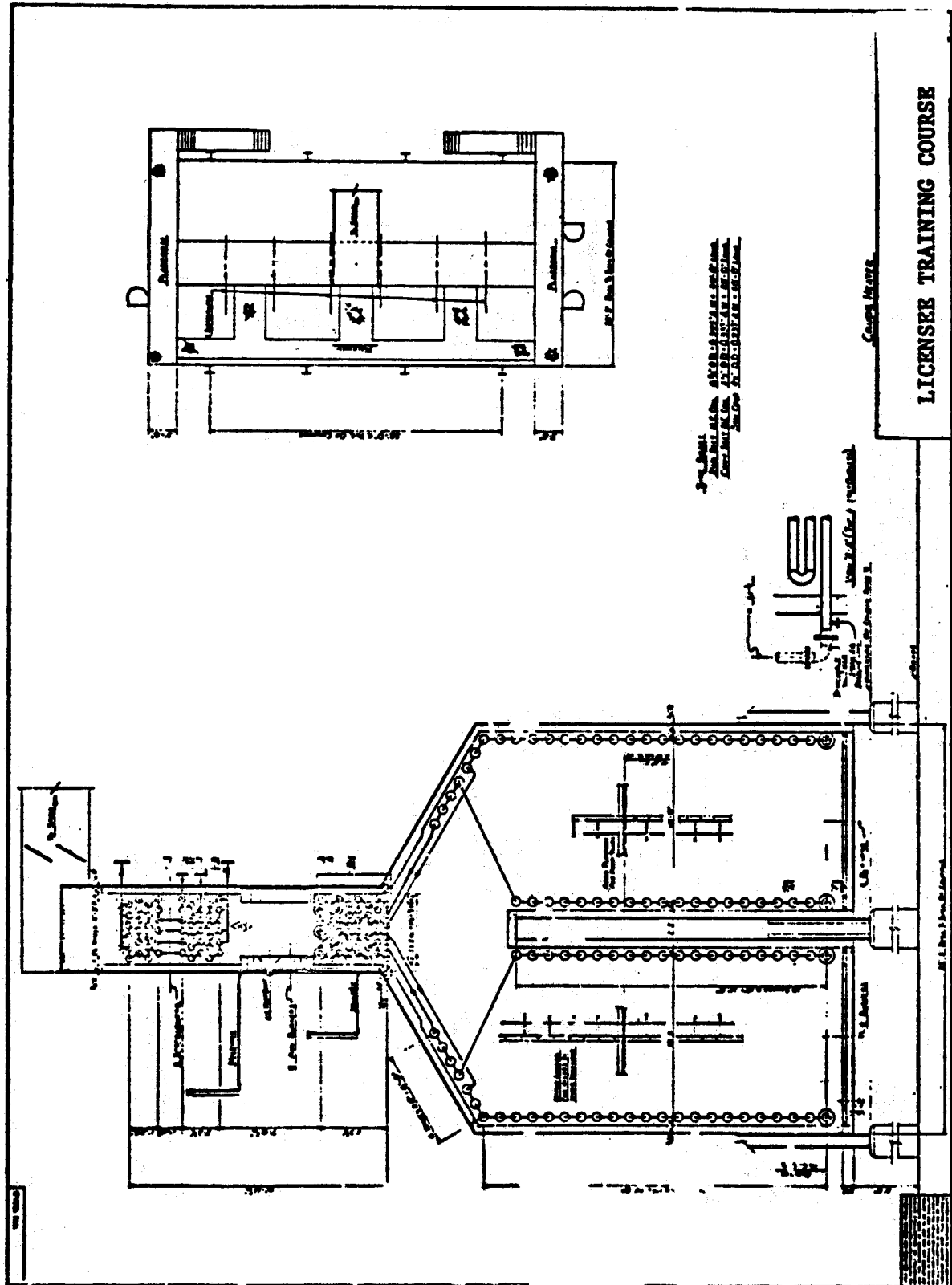


Figure 43

TABLE 4

Typical Design Heat Absorption Rates
Average for Radiant Section
For 20 x 10⁶ and Greater

<u>Service</u>	<u>Average Heat Abs.</u> <u>BTU/Hour/Sq. Ft.</u>
Crude Oil Distillation	12,000
Vacuum Distillation	8,000 - 10,000
Reboiling	10,000 - 12,000
Lube Oil Solvent	8,000
Propane Deasphalting	9,000
Catalytic Reforming	11,000 - 13,000
Hot Oil Belt	12,000
Black Oil Cracking	10,000 - 12,000
Fluid Catalytic Feed	12,000
Hydro-Treating	10,000 - 12,500
Pipe Line Heating	10,000