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HEATERS

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I. REASON FOR DISCUSSION

A. Why do we want to discuss this equipment?

1. It is one of the most neglected pieces of equipment in a unit.
2. It is usually purchased as a least cost bid. This means it is probably the most tightly designed equipment.
3. It is the only place in the unit where exposed fires are used to process flammable fluids.
4. It often is pushed over design during operation without operator knowledge.
5. Operator judgments are frequently made with a lack of understanding.
6. Improper operation can lead to disastrous consequences.

B. Figure 1a, 1b, 1c

This heater is a two service, charge and reboiler, heater. It is of the cabin type with horizontal tubes and an air preheat system. The fire started when a tube in the hip section of the reboiler coil ruptured during a startup. The liquid drained to the floor and into the common air ducting. The fire spread underneath and over the whole heater, stacks and forced-air system. The heater was damaged from an explosion after the tube failure. Fire damages had been minimal.

The operating group may have assumed the reboiler side was too large soon after the unit was commissioned. There may have been an under-estimation of the convection section duty in the design stage. In future startup's, the charge side was operating while the reboiler radiant section may not have been. Possible contaminated condensate in the cooler, parallel, reboiler radiant coil tubes vaporized, in the zones shown on sketch no. 1a, from the hot flue gases of the reactor charge coils section.

The tube wall thinned on the inside. Externally inspection revealed the tubes had never been exposed to excessive temperatures.

The corroded/eroded surface was caused by vapors contaminated with traces of chlorides. This was influenced by the flue gas temperatures from the reactor charge section of the heater. The external 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 was about 7 years old. It is completely steamed-out at each shutdown. It was shown how small traces of chloride could reach the reboiler coils from the stripper column if the unit was not properly water washed and drained before each startup. Figures 1b and 1c are the results of this occurrence. The unit remained ddown one year until repairs were completed.

C. Figure 2

A typical aftermath of a heater firebox explosion. Leaking fuel gas valves allowed an explosive build-up of gas into the heater. The operator's torch caused ignition while three feet away from the firebox. The heater was not purged prior to ignition

D. Figure 3

A typical common breeching situation. The fire originated in the center heater. It damaged 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. This tube was inaccessible for inspection.

G. Figure 6a, 6b, 6c, 6d, 6e, 6f, 6g

These failures are the results of a tube rupture at high pressure. Back flow resulted in erosion of the exterior elbow. This caused most of the external damage.

The fire was contained in the heater.

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

Results of a tube rupture. The consequences were 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 have happened with a refinery furnace. It demonstrates that more attention to the equipment and superior equipment should be considered in your units.

II. HEATER NOMENCLATURE

Figures 8 through 13 describe roughly the nomenclature used by heater designers.

III. HEATER CLASSIFICATION

A. By Service

1. No decomposition

a. Atmospheric distillation

b. Vacuum distillation

c. Catalytic processes

i. Hydrodesulfurization

ii. Catalytic cracking

iii. Catalytic reforming

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

d. Reboilers

e. Miscellaneous

2. Partial conversion

a. Thermal cracking

i. Single coil

ii. Selective

b. Continuous coking

3. Total conversion

a. Thermal reforming

b. Viscosity breaking

c. Pyrolytic cracking

i. 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

IV. FUNCTION OF HEATER

A. General

Most refining processes require a source of heat to raise temperatures of the streams to be processed. Heater design has had to become more sophisticated to keep pace with process demands and economics.

Petroleum heaters can be divided into three main groups. Each can be differentiated according to the amount of decomposition obtained:

1. Heaters used for heating only with little or no decomposition of the processing fluid.
2. Heaters where in addition to the heating, substantially all of the processing fluid decomposition desired for the refining process is obtained in the heater.
3. Heaters where only partial decomposition is obtained in the heater. The remainder of the reaction proceeds in the reaction chambers or soaking drums. These downstream equipment are usually not heated externally.

Heaters of the first group are designed to give minimum time-temperature effect at the highest operating temperatures employed. They are used for non-destructive distillation where no chemical change of the overhead and residual products is desired. Examples are topping, atmospheric or vacuum distillation, rerunning and pre-reaction heating .

Heaters of the second group are designed to give maximum time-temperature effect at the highest operating temperatures employed. They are used with cracking units for lighter overhead stocks when no reaction chamber is employed. These heaters are usually designed such that the heating surface nearest the heater outlet is subjected to lower rates of heat input than the rest of the tube bank. This provides 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. They are particularly difficult for residual cracking stocks that 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. This is to assure effective operation of the chamber, without excessive decomposition and resultant coke deposition in the coil.

B. Historical Development

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

Shell heaters were limited to low rates of heat input. There was a large volume of oil per unit of heat-absorbing surface. Low oil velocities washed the surfaces. The permissible maximum heat absorption rate was determined by the rate of free convection or convection. This rate was induced by steam jets within the body of the oil. These rates remained low. They were not in excess of 3,000 Btu/ft²hr without overheating the oil in contact with the heated portion of the shell.

Coke deposition and burning out of still bottoms almost invariably followed attempts to operate above these rates. The addition of internal flues to shell heaters (to simulate a fire tube boiler) improved their operation and increased capacity.

Shell heaters were primarily batch heaters. Continuous operation, which gradually displaced batch operation in the larger refineries, could be accomplished only by operating a battery of shell heaters in series. Each shell operated at a slightly higher temperature than its predecessor. Batteries of shell heaters became too cumbersome with large scale operation.

Tubular heaters became comparatively successful on a small scale. Successful dehydration and refining of emulsified oils applications led to the gradual adoption of tubular heaters for general refining purposes. Eventually tubular heaters were substituted for shell heaters in large-scale refining operations.

The introduction of cracking contributed greatly to the development of tubular heaters. It definitively demonstrated their superiority over the shell type. Cracking called for heaters that could withstand higher pressures and higher metal temperatures than could be withstood by an externally heated shell.

The earlier tubular heaters varied little in their setting from shell heaters. The shell was displaced by a bank of tubes placed over the firebox. Average rates of 5,000 - 6,000 Btu/ft²hr were obtained. This was due to the a velocity was imparted to the oil by pumping in continuously through the tube bank. It was also due to the increase in the ratio of heating surface to volume. These earlier tubular heaters presented zones of local over-heating, or "hot spots". Actual rates of 15,000 to 20,000 Btu/ft²hr were present in some sections of the tubes directly over the fires .

The hot spots became more pronounced as the average rates of heat input and oil temperatures were increased. Tubes failed 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. 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. Greater average rates of heat input were obtained in these heaters because of more uniform heat distribution. Overheating the shielded bank still occurred. The tubes first coming in contact with the invisible products of combustion easily overheated. It was necessary to reduce the gas temperature for successful operation.

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 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 recirculated flue gases. The desired temperature of the gases entering the tube bank or the "heating chamber" was obtained in this manner.

In other heaters, "radiant" tubes were installed in the combustion chamber to cool the products of combustion. They absorbed 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. The convection section was intended to supply the major portion of the total heat. It was usually separated from the combustion chamber by a refractory wall. Particular care was taken in these heaters to place the "radiant" tubes out of the path of the products of combustion. The flames or the small number of radiant tubes were shielded. This reduced the heat input rates since the tubes had been subjected to the full radiation of the combustion chamber. Refractory shields cut radiant rates approximately in two.

Flue gas recirculation and the shielded "radiant tube" heaters were gradually abandoned. High maintenance and operating costs of flue gas recirculating equipment had been required. There were frequent failures of the enclosures around the tubes or the flame. These heaters were replaced with the open flame heaters that had had comparatively satisfactory operation.

The open flame heaters did not employ flue gas recirculation. Neither the tubes nor the flames were shielded by refractory.

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. This led to 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 percent of the total in the latest designs.

Fig. 14a, A and B, are flue gas recirculating heaters of an early type widely used in petroleum refining. The fresh products of combustion generated in the combustion chamber are mixed with flue gases withdrawn from the flue to the stack. This is accomplished with a flue gas recirculating fan. The flue gas is distributed along the length of the heater through openings in a distributing duct.

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

Fig. 14a, B, is an early shielded, radiant heat oil heater. A roof bank was installed to absorb sufficient radiant heat from the products of combustion prior to entering the "main" or convection bank of the heater. This permitted their safe use in this bank without flue gas recirculation or high amounts of excess air. The portion of the tubes over the combustion chamber where radiation was the most intense was covered with ceramic enclosures. This protected the relatively small number of tubes in the roof bank from the full radiation in the combustion chamber. Cast-iron enclosures were used on the tubes over the main bank where radiation was comparatively low.

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

Fig. 14a, C, shows another type of an early shielded, radiant heat oil heater. Shielding enclosures or "muffles" were placed around the flames and the products of combustion. This protected the comparatively small amount of radiant heat absorbing surface from direct radiation. The enclosures were made of carborundum- the enclosures were subjected to high temperatures. Comparatively high rates of heat transfer were obtained through these enclosures to

the radiant tubes. The heater was intended for low excess air operation to obtain high flame-burst temperatures and highest radiation through the carborundum.

Fig. 14b, D, is an early open flame, radiant heat, oil heater. It has a much larger combustion chamber than the heater shown in Fig. 14a, A. Approximately one-half of the total tube bank is located along the roof of the combustion chamber. This permits the use of bare, radiant heating tubes and open flames.

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

Fig. 14b, F is a partially shielded flame heater. Muffles shield only a small portion of the flames. They promote and complete combustion rather than shield the flames and the tubes. Approximately 75 percent of the total surface is exposed to radiation. The floor, back walls and roof are covered with tubes.

Fig. 14b, G, is an early down-draft heater. It consists of a vertical cylindrical firing area with tubes placed along its whole circumference. A secondary, horizontal cylindrical section with tubes located along the lower half of this section is provided. Downstream is a convection section. An air preheater was not generally used. The later down-draft heaters consist only of a vertical, cylindrical section and an air preheater.

Fig. 14c, H, is an updraft heater of the same general type as Fig. 14b, G. The convection section when present is very small. 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 ratios of radiant heat-absorbing surface to total surface used. The heaters shown on Fig. 14a, B and C, have too little radiant heat-absorbing surface. Shielding the surface was required to protect the tubes from excessive rates, tube overheating, coke deposition and tube failure.

The heater shown on 14c, H, has too much radiant heat-absorbing surface in proportion to the furnace envelope. The effectiveness of the surface will be low, unless the furnace gases wash the upper portion of the tubes. This supplies an appreciable amount of 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 form approximately two-fifths of the furnace envelope area.

The optimum effectiveness of tubes irradiated from one side is obtained with single rows of tubes in the combustion chamber. The floor tubes of this heater are the tubes of the highest rates of heat input.

The heaters preceeding Fig. 14c, I, had fixed heat distributions and were inflexible as to a change in the type of service. Design was strictly for a certain process operation.

The heater on Fig. 14c, I, can be operated to change 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 firing tunnel compartments. The center tunnel, F, is the firing tunnel proper. The upper and lower tunnels, A and B, admit secondary air to the fuel after its ignition in tunnel F. The rates of heat input into the floor tubes and the roof tubes near the firing wall are proportionally lower when all the air passing through compartment F is primary air and short non-luminous flames are present. 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 will be proportionally higher.

Restricting the flow of air through compartment F and admitting a large portion of the air as secondary through either compartment A or B will produce semi-luminous flames. Admitting the secondary air through compartment A will produce greater heat transfer to the floor tubes. Admitting the secondary air through compartment B will produce greater heat transfer to the roof tubes.

Fig. 14c, J, is an updraft, rectangular heater in which the radiant heat absorbing surface is located along the roof and the side wall of the heater. The tubes in the roof are subject to both radiant and convection 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. It directs the gases at high velocities along the shielded faces of these tubes. It imparts a high convection component to the shielded faces. It also partially equalizes the heat pickup along the circumference of the top row of the roof tubes. A parallel flow arrangement is used for large capacities. Series flow arrangement is shown by the dashed lines.

A wide variation in heat distribution can be obtained by changing the type and firing rate of the two sets of burners in the combustion chamber. 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. This type of heater may be used without changes for a wide range of charge stocks that the heaters shown on Figs. 14a,B - 14c,I inclusive. 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 a double end-fired heater. The two combustion chambers are completely separated. The floor and

roof tubes of one chamber function independently of those in the other. This provides more definite control of the heating curve than 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. It can be used to process a single stream in series through both compartments, as shown in dotted lines.

In the early 1930's, UOP developed the Equiflux Heater. It was developed to overcome the deficiencies of the earlier box type heaters. It still is the best heater for thermal cracking. It was developed for that service. Controllability of the heating curve was the main reason for its use. It cost more than the box type. This heater is shown in Fig. 15. The first designs were the down-fired as shown in 15A and B. Later in the 1930's, the updraft type, Fig. 15C, was designed.

The cellular structure of the heaters permitted the use of any number of streams. Each stream was subjected to the optimum thermal treatment. The cellular structure permits an the optimum dimensional relationship of the heater structure for the best gas hydraulics. This is 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 diameter. The radiant tubes were 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 a 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 a 20% variation in radiant rates.

The Equiflux radiant tubes were usually designed for 15,000 Btu/ft²hr when thermal cracking black oil.

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

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

The average radiant rates in the centerwall updraft heater were usually 11,000 - 11,500. They were 10,000 -10,500 Btu/ft²hr in the heater without the center wall. The floor fired types were good when gas was fired. They not so good for oil firing. This is one reason why the updraft heater without the centerwall was designed, Fig. 17. It could be fired from the ends horizontally or vertically from floor. UOP then built a side-fired, centerwall updraft heater. It was wider than previously. The bottom and end-fired heaters were about 10'-6" wide. The side-fired heater was about 19 ft. wide It was built in about 1955.

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". 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/ft²hr. The centerwall updraft in crude service used 11,500 12,000 Btu/ft²hr.

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.

Other acceptable heaters are shown in 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 center wall (UOP)
- F. Floor fired horizontal (BORN)
- G. Equiflux downfired (UOP)
- H. Circular (De Florez)
- J. Isoflow (Petrochem)
- K. Isoflow (Integral tube) (Petrochem)
- L. Radiant wall (Selas)
- M. Inverted wicket (UOP)

There are other shapes. The above gives 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 flames can cause impingement or overheating of certain tubes in nearly all of the stills. The 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. IS.

2. Hot Tubes

The rate of heat absorption tends to be high at the entrance to the convection section. Heat is delivered by both radiation and convection (black tubes in type (A), (B), (D), (E), and (F)) in this area. The feed, because it is cooler, is sometimes introduced through the tubes (a screen bank). In some instances steam is superheated or water is heated in the screen bank. The tubes situated closest to the flame receive the largest amount of heat (Black tubes in (E), (F), (H), (K), and (M)).

3. Oil Firing

Oil fuel tends to produce larger flames. Additionally, larger heat release burners are used. Stills (A), (B), (C), and (D) are best suited for oil firing. 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. The rate of heating varies widely in different parts of the boxes or cavities of such stills as (A) and (B). 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. The stills with a centerwall are most effective if a different rate of heat input is needed in each coil.

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). They can be obtained in other types if enough radiant surface is provided.

7. Capacity

Still (A), (F), (G), (K), and (L) can be built for the lowest capacities. 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 (DeFlorez). 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. The convection section duty is increased at the same time. 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 a change to a fuel that radiates less. Flue gas recirculation stills were widely used during the development of thermal cracking processes. Now engineers know how to design stills for specific radiant absorption rates. Re-circulation stills are no longer installed.

V. CONSIDERATIONS IN HEATER DESIGN

A. General

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 the tubes. The rating involves a heat balance between the heat releasing and heat absorbing streams and a rate relationship. Much of the heat is transferred by radiation instead of convection in contrast to the usual heat exchanger.

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

B. Modes of Heat Transfer

1. Conduction

Conduction is the transmission of heat through a material by contact between adjacent objects or molecules at different temperatures. The metallic handle of a

metal pan will be too hot to touch though only the pan itself is cooking over a fire.

2. Convection

Convection is the transmission of heat between moving masses of matter or between moving particles (flue gas for example) and a stationary mass (heater tube). Heat will flow from a high temperature body to one at lower temperature. The motion of the fluid may be due to the natural buoyancy of the heated molecules (hot air rises). This is termed free or natural convection. The motion of the fluid may be mechanically produced (as by a fan) in which case it is termed forced convection. The radiator fan on an automobile cools the radiator by convection; cooler air is forced against the hot radiator to remove heat.

3. Radiation

Radiation is energy transmitted in straight lines without obstruction between two or more bodies. Radiant heat may pass through a vacuum, air, some gases and liquids and a few solids such as glass and quartz. Radiation is considered to be a wave phenomenon. In many instances it is considered to exemplify particulate qualities.

Radiation is entirely different from the conduction and convection. Light is a visible form of radiation. All heat radiation is not necessarily visible. A person can feel the heat radiation from a furnace wall or a hot piece of metal though neither may be glowing (that is, neither is radiating light). We can feel the radiant energy of the sun warming our body as we sit on a beach on a sunny day.

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 basic radiation concepts to furnace design. They 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:

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

Where:

W_B = Radiant Energy Rate

σ = Stefan-Boltzman Constant

T = Absolute Temperature, $^{\circ}R$

The Stefan-Boltzman constant, σ , has a value of 0.1713×10^{-8} Btu/ft²hr $^{\circ}R^4$.

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

$$q_r = \sigma \alpha A F (T_a^4 - T_b^4)$$

where: (2)

q_r = Radiant Heat Transfer, Btu/hr

A = Area (one of the surfaces), ft²

F = Radiant Exchange Factor

T_a = Temperature of One Surface, $^{\circ}R$

T_b = Temperature of Other Surface, $^{\circ}R$

A is the area of either of the surfaces. F is an exchange factor which depends on the relative area and arrangement of the 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

The heat-absorbing surface is usually well defined in a furnace. The rate of heat transfer per unit of absorbing area is important to the design. 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, multiplied by their exposed length and the center-to-center spacing, Equation (3):

$$A_{cp} = N \times L \times S$$

where: (3)

A_{cp} = Cold Plane Surface (Area)

N = Number of tubes

L = Length of tubes, effective, ft

S = Center-to-center spacing, ft

The tube bank will not absorb all the energy radiated to the cold plane area. The total cold plane area must be corrected by an absorption effectiveness factor, α . Hottel has published values for the factor 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.

The product of the actual coldplane area, A_{cp} , and the effectiveness factor, α , 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.

$$\text{Equivalent Cold Plane Area} = \alpha A_{cp}$$

where:

A_{cp} = Cold Plane Surface (Area)

α = Effectiveness Factor

The shield tubes require special handling. These tubes are not backed by a reradiating refractory wall. Any energy not directly absorbed by the shield tubes moves past and is absorbed in the convection section. Therefore, from the standpoint of the radiating gas, α for the shield tubes is unity.

The applicable value of α is applied only to the refractory-backed tubes. The cold plane area of the shield tubes is taken at a full value of 1.0.

It might be argued that the shield tubes do not pick up all this heat; the computed heat absorption in the radiant section will be too high. 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. This 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. It simplifies the calculations.

2. Exchange-Factor.

The remaining term to be evaluated in Equation 2 is the exchange factor, F . The gas in the firebox is far from an ideal black body. The only constituents in

normal flue gas that contribute significantly to radiant transfer 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 the concentration of each of these two components, the furnace dimensions, and the temperatures of the gas and 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. The emissivity can be correlated as a function of the PL product and gas temperature. This is demonstrated in Figures 23a-d. Curves are provided for different tube wall temperatures.

The exchange factor is also dependent on the amount of reradiation from exposed refractory. Energy striking such refractory is reflected back toward the tubes. A furnace with a large amount of exposed refractory will transfer more heat per unit of tube surface than one with refractory 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; it is the total envelope area of the firebox, less the equivalent cold plane area of all the tubes.

Figure 24 considers the tubes do not completely absorb all radiant energy striking them. The curves are based on a tube surface absorptivity of 0.9. This is a commonly accepted value for oxidized metal surfaces.

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

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

where:

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

F = Exchange Factor = $f(\alpha, \epsilon, L, A_r, \alpha A_{cp}, \dots)$

α = Effectiveness Factor (Hottel)

T_g = Flue Gas Temperature, °R

T_t = Tube Wall Temperature, °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).

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

where:

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

We make some simplifying approximations since convective heat transfer is not the major contributor. For the usual furnace, h_{Rc} is about 2.0 Btu/hrft²°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.

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

$$\text{since } h_{Rc} = \sim 2.0 \text{ BTU/hr ft}^2 \text{ } ^\circ\text{F}$$

$$A_{Rt} = \sim 2\alpha A_{cp}$$

$$q_{Rc} = 2(2\alpha A_{cp})(T_g - T_t) = 4\alpha A_{cp}(T_g - T_t) \quad (7)$$

4. Total Heat Transfer Rate

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

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

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

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

For the usual furnace, the exchange factor, F , is approximately 0.57.

Substituting into the right hand side of the above equation:

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

or

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

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

The distance between the tubes and the wall in a single fired heater can affect heat transfer rates. A tube placed one diameter from a reradiating wall will absorb considerably more heat than a tube contacting the wall (see Figure 25d).

The movement of gases to the walls of a furnace contact the tubes and walls. They both will be heated by convection as they are contacted by the furnace gases. The wall temperature will be increased as convective heat transfer augments radiative. The heated walls will radiate heat back to the wall side of the tubes.

Convective heat transfer to the tubes will be reduced as the distance between tube and wall is decreased; it will be more difficult for the gases to go past the tube and heat the back end. Radiant heat transfer will be reduced. Radiation reflecting and emitting from the walls will transfer heat over a full hemispherical angle. The radiant waves emanating in these directions will contact fewer tube surfaces as the tubes are placed closer to the walls. Lower heat transfer rates in the back end of the tube will result. The overall radiant heat absorption rate will decline. It will be necessary to increase the firing rate to transfer more heat to the process.

Points of maximum tube wall temperatures (TWTs) are located on the front end (side facing the burners) of the tubes. Decreased tube to wall clearances may raise TWTs to levels higher than recommended as a portion of the heat load is transferred from the back end to the front end of the radiant section tubes.

A refiner may wish to add insulation in an existing single fired heater to reduce heat losses and increase heater efficiency. The distance between tube and wall will decrease as more insulation as added to the shielded walls. This can reduce the heat absorbing capacity of the radiant section. The furnace may be forced to fire harder to increase the heat transfer rate up to previous levels. The efficiency will decrease as the fuel fired increases. Instead of increasing efficiency by limiting heat losses, the efficiency may decrease.

Tube Wall Temperature Calculation

The tube wall temperature (TWT) on an unfouled (clean) basis is defined as follows:

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

where:

T_f = the bulk fluid temperature

$\left(\frac{Q}{A_o}\right) \left(\frac{1}{h_i}\right) \left(\frac{D_o}{D_i}\right) = \Delta T_f$ = the temperature across the inside film

$\left(\frac{Q}{A_o}\right) \left(\frac{D_o}{2t}\right) \ln\left(\frac{D_o}{D_i}\right) = \Delta T_m$ = the temperature across the tube metal

Both the temperature differences across the film and metal are a function of the heat flux. The appropriate heat flux must be calculated wherever the tube wall temperature is calculated. Longitudinal and circumferential heat distribution must be considered.

ΔT Across the Tube Metal (ΔT_m)

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

$$\frac{dq}{dA} = -K \frac{\partial T}{\partial t} \quad (\text{Note: the minus sign is due to convention.})$$

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

$$\Delta T_f \propto \frac{Q}{A_o}$$

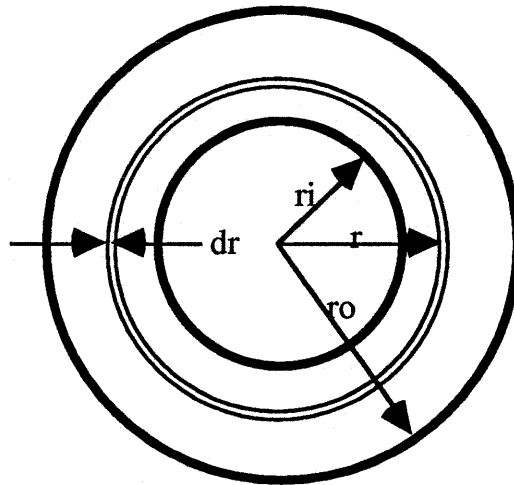
$$\Delta T_f \propto \frac{1}{h}$$

$$h \propto G^{0.8}$$

$$\Delta T_f \propto \frac{1}{G^{0.8}}$$

$$\Delta T_m \propto \frac{Q}{A_o}$$

Let us look at heat flow through a tube:



Where :

r_i = inside radius

r_o = outside radius

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.

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

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

applying this relation to our cylinder:

$$q = K (2 \pi r L) \frac{dT}{dr}$$

$$\Delta T_m = \left(\frac{Q}{A_o} \right) \left(\frac{D_o}{2L} \right) \ln \left(\frac{D_o}{D_i} \right)$$

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

$$dt = dr$$

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

$$q = \frac{KA_L(T_i - T_o)}{r_o - r_i}$$

where A_L = logarithmic mean area

Since the logarithmic mean radius, r_L , is

$$r_L = \frac{r_o - r_i}{\ln(r_o/r_i)}$$

the logarithmic mean area, A_L , is

$$A_L = \frac{A_i - A_o}{\ln(A_o/A_i)} \quad \text{or}$$

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

The temperature difference across the metal is ΔT_m . Rearranging our basic equations:

$$\Delta T_m = T_o - T_i = \frac{q}{A_L} \frac{t}{K}$$

$$\text{where :} \quad t = r_o - r_i$$

In order to place the heat flux, q/A , in terms of the outside circumferential tube area, we note:

$$A_L = \frac{A_i - A_o}{\ln(A_o/A_i)} = 2\pi r_L L = \pi D_L L$$

where:

A_i = inside circumferential area

A_o = outside circumferential area

D_L = logarithmic mean diameter

since: $A_L = \pi D_L L$ and $A_o = \pi D_o L$, then $\frac{A_L}{A_o} = \frac{D_L}{D_o}$

therefore, $A_L = A_o \frac{D_L}{D_o}$

Returning to our basic equations and substituting:

$$\Delta T_m = T_o - T_i = \frac{q}{A_L} \frac{t}{K}$$

$$= \frac{q}{A_o} \frac{D_o}{D_L} \frac{t}{K}$$

$$= \frac{q}{A_o} \left[\frac{D_o}{\frac{D_o - D_i}{\ln(D_o/D_i)}} \right] \left[\frac{(D_o/D_i)/2}{K} \right]$$

$$= \frac{q}{A_o} (D_o) \ln(D_o/D_i) \left(\frac{1}{2K} \right)$$

$$\Delta T_m = \frac{q}{A_o} \left(\frac{D_o}{2K} \right) \ln(D_o/D_i)$$

Handwritten notes:

$$\frac{dq}{dA} = -t \frac{\partial T}{\partial x}, \quad \frac{\partial T}{\partial x} = -k \frac{\Delta T}{L}$$

$$\Delta T_m = \left(\frac{D_o}{A} \right) \left(\frac{t}{K} \right)$$

ΔT Across the Inside Film (ΔT_f)

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

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

Where:

h = Inside Film Coefficient, Btu/hrft²°F

q = Heat Transferred, Btu/hr

A = Surface Area, ft²

T_w = Temperature at the Inside of the Tube Wall, °F

T_f = Average Fluid Temperature, °F

If we rearrange and place on a steady-state basis, the equation can be expressed:

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

Noting that (q/A) is the heat flux (or average transfer rate) and $T_w - T_f = \Delta T_f$, therefore:

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

where

A = area of isothermal surface

T_f = bulk temperature of fluid

T_w = temperature at wall

The inside film is located at the inside of the tube. Expressing the heat flux on an outside tube area basis:

$$\text{Since } \left(\frac{D_o}{D_i}\right) = \left(\frac{A_o}{A_i}\right)$$

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

NOTE: If there are coke deposits and scale deposits on the interior of the tube, the tube wall temperature equation becomes:

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

API RP 530

$$q_m = q_{\text{ave}} F_c F_L F_T + q_c$$

longitudinal heat flux factor

$$F_T = \frac{T_g^4 - T_s^4}{T_g^4 - T_a^4}$$

tube metal temp correction factor

Crawford heat transfer

T_g : skin temp
 T_a : ambient temp
 T_s : tube metal temp

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 in convective heat transfer. The overall heat transfer coefficient would best be 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.

Radiant heat transfer predominates in the radiant section. The outside (flue gas side) coefficient can be ignored since radiant heat transfer is predominantly a surface phenomenon. We can concentrate on the inside (in tube) film coefficient.

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

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

where:

q = Quantity of heat transferred per unit time in direction normal to surface

A = The isothermal surface area through which heat is transferred

T_f = Average fluid temperature of the stream

$T_w = T$ temperature at the wall

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

We recommend the Sieder-Tate method for determining the individual inside heat transfer coefficient for all systems except hydrocarbon vapor (single 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:

Viscosity
 $Pr = 0.78$

RP
API 530, Vapor flow

$$h_v = 0.021 \left(\frac{k}{D_i} \right) Re^{0.8} Pr^{0.4} \left(\frac{T_b}{T_w} \right)^{0.5}$$

($Re \geq 10,000$)

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

Where:

C_p = heat capacity, Btu/lb°F

G = mass velocity, lb/ft²sec

D_i = inside diameter of tube, in

Dittus-Boelter

$$h_u = \frac{k D_i}{x}$$

$$= 0.023 (Re)^{0.8} (Pr)^{0.4}$$

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 direct radiant heat on either side in diagram B of Figure 25e. 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. The overall heat transferred would be greater in the double fired (radiating planes on both sides) tubes if we fire equivalently on both sides, each side firing at a rate equivalent to that of the single fired tube

Let us compare shielded tubes vs. double fired tubes. Let us assume the tubes are spaced on two nominal diameters, center to center. The ratio of radiant maximum flux to average will be approximately 1.90 for the former and 1.22 for the latter. This means 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; the total surface area of the single fired tube will be capable of absorbing 55% more heat.

Increasing the distance between the tubes will decrease the maximum to average heat flux ratio. As the centerline to centerline distance increases, the ratio of the maximum to average flux will decrease. This is more pronounced in single fired tubes. With single firing, more heat is transferred to the tube by reradiation of the refractory wall as the tube spacing is increased. Although the peak will remain unchanged, the back of the tube will absorb more radiant heat. API RP 530, Recommended Practice for Calculation of Heater Tube Thickness in Petroleum Refineries, provides a graph of the ratio of the circumferential maximum to average heat flux on a 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 most 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 1.2 and 1.5 and may vary considerably between different sections of the furnace.

Tube Thicknesses

API RP 530, Recommended Practice for Calculation of Heater Tube Thickness in Petroleum Refineries, provides a method for the determination of tube metal temperatures. More importantly, it also provides a method to determine required heater tube thicknesses. The equations for heater tube thickness determination are as follows:

For tube metal temperatures in the elastic (low temperature) zone -

$$t_{\min} = \frac{P_E D_o}{2S_E + P_E}$$

where:

$t_{\min}$ = minimum tube wall thickness required, in

P_E = design pressure, psig

D_o = tube outside diameter, inches

S_E = elastic allowable stress at maximum tube wall
temperature, psi

For tube metal temperatures in the plastic (high temperature) zone -

$$t_{\min} = \frac{P_R D_o}{2S_R + P_R}$$

100000 hr life $\approx$ 11 years
(typical)

where:

P_R = maximum operating pressure at any normal
operation condition, psig

S_R = rupture allowable stress at maximum tube wall
temperature, psi

Both elastic and rupture allowable stresses at their respective temperatures are provided in API RP 530.

During the determination of the required tube wall thickness, both equations should be used. The maximum thickness calculated governs and is added to any required corrosion allowance to give the overall thickness required.

Tube thickness is described as either average wall (see ASTM Specification No. A 530) or minimum wall (see ASTM Specification No. A 450). If average wall thickness is specified, the smallest permissible thickness at any point cannot be less than 87.5% of the average (12.5% lower). If minimum wall thickness is specified, the tube wall thickness cannot be less than that specified at any point.

5. Radiant Heat Balance

To determine the temperature of the flue gas in the firebox, we must 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_g . Expressed as Equation 9:

$$q_n + q_a + q_f = q_R + q_L + q_g \quad (9)$$

$$q_R = q_n + q_a + q_f - q_L - q_g \quad (10)$$

where:

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_g = Sensible heat of exit flue gas

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

$$\frac{q_R}{\alpha A_{cp} F} = \left[1 + \frac{q_a}{q_n} + \frac{q_f}{q_n} - \frac{q_L}{q_n} - \frac{q_g}{q_n} \right] \frac{q_n}{\alpha A_{cp} F} \quad (11)$$

The net heat release q_n is established as soon as the total heater duty and efficiency are determined. 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_g as the only variable on the righthand side of Equation 11. 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. 37, 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 actual furnaces, neither of these temperatures is constant. Average effective temperatures must be selected in order to use Figure 28.

Tube wall temperatures depend 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 estimated as the arithmetic mean of the radiant section inlet and outlet unless there is a phase change. 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 for heaters with an approximately square cross-section and no large areas of refractory with direct flame impingement, the average effective gas temperature is very close to that of the exit gas. For some other types, such as a cylindrical furnace with a high length/diameter ratio, the average effective 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. 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. 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

$$q_c = q_{rL} + U_o A_o (LMTD)$$

1. General

$$h_c = \Phi(h_{ce}, h_{cr}, h_{cw})$$

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 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:

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

where:

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

T_f = Temperature of film, °R

G = Mass Velocity flue gas, lbs/ft²sec

d = Tube Outside Diameter, in

Figure 29 shows the convection coefficient for several common tube sizes. The average gas film temperature can be estimated as the average intube temperature plus one-half the log mean temperature difference from flue gas to fluid. The flue gas mass velocity per row is evaluated at the minimum cross-section in the convection section. The minimum cross-sectional area occurs at each row where a straight line intersects the center of each tube.

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.

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, Heat Transmission.

The apparent heat transfer coefficient for radiation from the hot gases to the convection section is

$$h_{Cr} = \frac{\sigma \left(\frac{1 + \epsilon_t}{2} \right) (\epsilon_g T_g^4 - \alpha_g T_t^4)}{T_g - T_t} \quad (13)$$

where:

- T_g = absolute gas temperature
- T_t = absolute average tube wall temperature
- α_g = gas absorptivity
- ϵ_t = tube emissivity
- ϵ_g = gas emissivity
- σ = Stefan-Boltzman constant.

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.

$$h_{cw} = 9.46 (T_w/1000)^3$$

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 14:

$$f = \frac{h_{Cw}}{h_{Cc} + h_{Cr} + h_{Cw}} \left(\frac{A_{Cw}}{A_{Ct}} \right) \quad (14)$$

where:

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 per row in convection section

A_{Ct} = Bare 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 15:

$$h_c = (1 + f) (h_{Cc} + h_{Cr}) \quad (15)$$

The intube coefficient is calculated by standard methods, based on the flow rate and physical properties of the fluid. The over-all outside heat transfer coefficient in the convection section is determined by Equation 16.

$$U_{co} = \frac{1}{\frac{D_o}{D_i h_{di}} + \frac{D_o}{D_i h_i} + \frac{t_w}{k_m} \frac{D_o}{D_L} + \frac{1}{h_c} + \frac{1}{h_{dc}}} \quad (16)$$

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.

We have expressed the overall heat transfer coefficient on an outside tube basis, It can be also described on an inside tube basis:

$$U_{co} = \frac{1}{\frac{D_o}{D_i h_{di}} + \frac{D_o}{D_i h_i} + \frac{t_w}{k_m} \frac{D_o}{D_L} + \frac{1}{h_c} + \frac{1}{h_{dc}}} \quad (16) \text{ or}$$

$$U_{ci} = \frac{1}{\frac{D_i}{D_o h_{dc}} + \frac{D_i}{D_o h_c} + \frac{t_w}{k_m} \frac{D_i}{D_L} + \frac{1}{h_i} + \frac{1}{h_{di}}}$$

where:

U_{ci} = overall heat transfer coefficient based upon inside tube area

U_{co} = overall heat transfer coefficient based upon outside tube area

D_i = inside tube diameter

D_L = logarithmic mean tube diameter

D_o = outside tube area

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

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

h_i = inside heat transfer coefficient

h_c = outside heat transfer coefficient

K_m = thermal conductivity of tube

t_w = tube wall thickness

Let us look at this overall coefficient more closely and make the following assumptions and observations:

fouling factor $< \begin{matrix} \text{Gas} & 0.002 \\ 0.1 & -44- & 0.005 \end{matrix}$

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

$$U_{co} = \frac{1}{\frac{1}{h_{di}} + \frac{1}{h_i} + \frac{t_w}{k_m} + \frac{1}{h_c} + \frac{1}{h_{dc}}}$$

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

$$U_{co} = \frac{1}{\frac{1}{h_{di}} + \frac{1}{h_i} + \frac{1}{h_c} + \frac{1}{h_{dc}}}$$

- c) Fouling is negligible —

$$U_{co} = \frac{1}{\frac{1}{h_i} + \frac{1}{h_c}}$$

Uop : take into consideration of fouling factor.

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

$$U_{co} = \frac{h_c(h_i)}{h_c + h_i} \quad \text{or equation (17), } U_c = \frac{h_c(h_{ci})}{h_c + h_{ci}} \quad (17)$$

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

6. Extended Surface

Extended surface (fins or studs) on convection section tubes provides additional surface to augment heat transfer. The presence of fins or studs increases the amount of heat transfer per unit of prime (bare) tube surface. Convective heat transfer coefficients for extended surface tubes depend upon the exact size and arrangement of the fins or studs.

Extended surface increases only convective heat transfer. Radiant transfer is computed on the basis of prime tube surface. Radiant transfer to the convection

$$A = \frac{Q}{U_o (LMTD)}$$

$$\text{Number of Rows} = \frac{A}{A_s}$$

section extended surface tubes is small. It is often can be neglected in refinery heater calculations

a. Extended Surface

i. History

An early method of applying extended surface to a tubular product involved cast iron rings. The rings were shrunk to the tube in such a fashion as to conduct heat from the rings to the tube. This type of extended surface, known as "gill rings", was very expensive and extremely heavy. Early methods of fastening included the brazing of both longitudinal and square plate-type extended surface to tubular products. This was also a costly and less attractive method of attachment.

Two methods of attaching a spiral wound fin to a tubular product were subsequently introduced to the industry. They received immediate acceptance and widespread use. One involved grooving a tubular product and rolling a smooth, continuous fin into the groove. This resulted in a tight mechanical bond. The other method used cut-type fins with a rolled underfoot. The fins were spirally wound around the tube and spot welded by resistance-welding equipment to the tubular product. Longitudinal type fins were being attached to tubular products in the same manner as the spiral wound product.

A "stud" type fin was also available. It was resistance welded to the tube. This product found applications in vertical-type furnaces and in some high temperature services. Owing to its low external surface ratio, it did not compete in general applications with the spiral-wound

type fins. Another spiral-wound fin tube employed a heli-arc weld at the base of the fin stock providing a continuous fillet weld.

Currently the production of finned tubes involves a smooth continuously, spiral wound fins. They are tension wrapped around the tube and high-frequency welded. This is an advancement over previous methods.

ii. Serrated Fins (Fig, 48)

Serrated, segmented or cut fins, are a relatively recent addition to the marketplace. Most segmented fin structures can match the effective surface areas of their solid counterparts. Some users have a preference for these type of fins based upon the ease of cleaning. The space between the segments permits lateral flow of the gas during operation (as well as the steam during cleaning). This permits more effective steam cleaning. However, these fins foul quicker than solid fins. This makes the benefits a matter of personal preference.

I'. 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.

II'. 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 $\frac{1}{8}$ " apart.

III'. Solid Fin (Fig. 48)

A'. 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.

B'. Thicker Fins

There is a great advantage of a solid fin welded to the tube by the high frequency welding process. The fins can be made much thicker than by the standard frequency process. Fin thicknesses can reach 0.1875". One raises the fin efficiency by increasing the fin thickness. This allows the individual to use fins at higher flue gas temperatures; at temperatures where the maximum fin tip temperature is exceeded for thinner fins.

Recommended fin tip temperatures for a given material should not be exceeded; the fins can corrode and decay. They can lose their effectiveness as corrosive products plug tube surfaces and flue gas access areas. They can lose their effectiveness when their surfaces are destroyed.

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

Carbon Steel	875°F
11% Chrome	1100°F
18 - 8	1500°F

C'. 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.

IV'. Studs (Fig. 51, 52 & 53)

Studding followed essentially the same development as finning. They were developed for boilers and then adapted to process heaters. A studded tube with maximum stud density will frequently have less effective surface than a comparable finned tube. Studs are frequently used when the user is concerned about heavy fouling. Studded tubes can be cleaned more easily than finned tubes. However, frequent sootblowing, limiting the fin structure and spacing sootblowers properly) often negate the need for studded tubes.

V'. Figures 49 & 50 - Tube Failures

VI'. Calculations

A'. General

It is increasingly important to improve the operating efficiency of refinery heating equipment as the cost of fuel increases. 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 previous bare tube example would often be totally redone. An exception involves anticipated severe fouling with poor oil as a fuel. We will just substitute tube banks for simplicity.

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

B'. Rating Method

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

- 1'. The heat transfer for flow of gases across banks of finned tubes.
- 2'. The radiation leakage from the combustion chamber to the convection section
- 3'. 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:

$$q_C = q_{SL} + U_C A_t (\text{LMTD}) \quad (18)$$

where:

q_C = Convective heat rate, Btu/hr

q_{SL} = Heat Rate Shield leakage, Btu/hr. The use of three bare rows of tubes shielding an extended surface bank usually this factor.

U_C = Convective overall heat transfer coefficient, Btu/hrft²°F

A_t = tube area, ft²

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.

$$\frac{h_C D_o}{k} = 0.36 \left(\frac{C_p \mu}{k} \right)^{1/3} \left(\frac{D_o G_{\max}}{\mu} \right)^{0.6} \quad (19)$$

where:

h_C = Convective Film heat transfer coefficient and Btu/hrft²°F

D_o = Outside diameter, ft.

k = Thermal Conductivity, BTU/hrft²°F/ft.

c_p = Specific Heat, Btu/lb°F

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-section, helical fins is:

$$E = \frac{\tanh(u_b)}{u_b} \quad (20)$$

where:

E = fin efficiency

u_b = parameter determined at base of fin

$$= x \sqrt{\frac{2h_c}{kY}}$$

where:

x = fin height, ft.

Y = cross-sectional width of fin, ft.

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

$$h_f = h_c \left(\frac{EA_f + A_t}{A_t} \right) \quad (21)$$

where:

h_f = fin convective heat transfer coefficient,
Btu/Hrft²°F

A_f = Fin Area, ft²/ Row

A_t = Total Surface Area of bare tube, ft²/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 (20), Gardner made the following assumptions:

- 1'. The heat flow and temperature distribution throughout the fin is independent of time; i.e., heat flow is steady.
- 2'. The fin material is homogeneous and isotropic.
- 3'. There is no heat source in the fin itself.
- 4'. 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.
- 5'. The thermal conductivity of the fin is constant.
- 6'. The heat transfer coefficient is the same over the entire fin surface.
- 7'. The temperature of the fluid surrounding the fin is uniform.
- 8'. The temperature of the fin base is uniform.
- 9'. The fin to tube offers no bond resistance.

10'. The fin thickness is so small compared with its height that the fin width temperature gradients may be neglected.

11'. The heat transferred through the outermost edge of the fin is negligible compared with that passing into the fin through the sides.

Figures 54 and 55 graphically present Equation (20) for annular fins and studs, respectively. The use of a fin efficiency factor makes Equation (21) useful for both annular fins and spines.

Monrad considered flue gas radiation and wall radiation when evaluating the total heat radiation to bare tubes. Theoretically, this would be applicable to extended surface as well. However, the use of both gas radiation and wall radiation coefficients has become questionable when applied to extended surface heat transfer. The use of such coefficients with extended surface has resulted in the underprediction of the necessary surface. The heat pick-up frequently did not match the predicted.

There are several theories why this occurs. One suggests the fin itself obstructs the radiant beams from reaching the bare tube surface. It also surmises the additional radiant heat transfer to the fins is insufficient to affect the resultant convective heat transfer.

There are a number of tube arrangements possible. For this presentation, equilateral triangular pitches of 2.0 times the nominal diameter of the tube is chosen as standard. The tube

spacing is fixed by the tube size selected. 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.

The tube size in the radiant section is often identical to that in the convection section. The number of tubes per row is such that the flue gas rate (natural draft operation) is about or below 15 ft/sec. These rates are frequently doubled when draft is mechanically induced or a huge stack is employed.

The surface 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 cross-sectional area is evaluated at the dimension where a straight line can be drawn through the tube centers of the row.

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. They must fit with the radiant section rating. 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:

$$h_{Cc} = \frac{2.14 T_f^{0.28} G_{max}^{0.6}}{d^{0.4}} \quad (22)$$

where:

T_f = Outside film temperature, °R

G = Flue gas mass velocity, lb/ft² sec

d = Outside diameter of tube, in

This equation is used to determine the effective heat transfer coefficient for a finned tube. The effective heat transfer coefficient for a finned tube is given by Equation (23). Note this coefficient becomes the outside coefficient, h_{Cc} , used in the bare tube rating procedure. Wall radiation and gas radiation are ignored.

Radiation Leakage from Combustion Chamber

The Wimpress method expressed here is based upon the Lobo and Evans method for calculating the heat absorption in furnace radiant sections. It 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:

$$q_{sL} = \sigma (1 - \alpha) A_{cp} (T_g^4 - T_t^4) \quad (29)$$

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 is a factor of comparison for direct radiation.

Figure 58 is a plot of Equation (29) rearranged as:

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

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:

$$f = \Delta P \left(\frac{2g_c \rho_g}{(G_{max})^2} \right) \left(\frac{\mu}{\mu_w} \right)^{0.14} \left(\frac{D_v}{S_T} \right)^{-0.4} \left(\frac{S_L}{S_T} \right)^{-0.6} \quad (31)$$

where:

ΔP = Pressure Drop, inches H_2O

g_c = Gravitational constant, 32.17 $lb_m \text{ ft}/lb_f \text{ sec}^2$

ρ_g = Density of gas, lbs/ft^3

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

L = Mean length of radiant beam, ft

μ = Flue Gas Viscosity, lb/ft hr

μ_w = Flue gas viscosity at tube wall, 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 (31) is based on the intensive data of Seider and Scott, Hoge and Pierson. The viscosity ratio to the 0.14 power is approximately 1.0. Therefore, for equilateral spacing, S_L equals S_T . Then:

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

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

$$\Delta P = 0.003 \left(\frac{f(G_{\max})^2 L}{\rho_g D_v} \right) \left(\frac{D_v}{S_T} \right)^{0.4} \quad (33)$$

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

PROBLEM

Determine the size of a convection bank required to transfer 23.4×10^6 Btu/hr from flue gases to a process coil.

- 1) Use 2 bare rows of shield tubes at the bottom of the convection section. Assume there is an additional row (considered to be in the radiant section) below the two shield rows.
- 2) Remaining tubes are finned with 1" high, 2 fins per inch of 0.05" thick carbon steel fins.

PROCESS CONDITIONS:

98,640 lb/hr flue gases in at 1610°F, out at 483°F

Process fluid in at 380°F, out at 483°F

$$h_i = 100 \text{ Btu/hrft}^2\text{°F}$$

use 6, 4.5" O.D. tubes per row, 8" center to center spacing, triangular pitch, effective length = 38.5 ft.

Convection bank width = 52"

Firing occurs at 25% excess air at the rate of 97.5×10^6 Btu/hr .

Assume there are 6, 4" (4.5" O.D.) tubes in the shield row atop the radiant section. Center to center tube spacing is 8". Each tube has an effective length of 38.5'.

The radiant section has an exchange factor equal to 0.635.

Solution

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: Shield row in radiation section-

$$A_{cp} = \frac{6 \text{ tubes } (8 \text{ in C to C}) (38.5 \text{ ft})}{12 \text{ in/ft}} = 154 \text{ ft}^2$$

$$\frac{C \text{ to } C}{D_o} = \frac{8}{4.5} = 1.78; \quad \alpha = 0.74 \quad (\text{See Fig. 18})$$

$$(1 - \alpha) A_{cp} F = (1 - 0.74) (154) (0.635) = 25.4 \text{ ft}^2$$

Assuming TWT = 483 + 100 = 583°F (Figure 58):

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

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

- 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
Net heat release @ 1610°F & 25% excess air = 0.45

$$= 0.45 - \frac{q_{SL}}{q_n} = 0.45 - \frac{0.75 \times 10^6}{97.5 \times 10^6} = 0.44$$

From Figure 37 : T = 1585°F

- C) Determine convective heat transfer to the 2 bare rows:

Gross width	=	=	52
Projected width	=	6 (4.5) =	27
Free width	=	25" =	2.08'

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

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

l) Assume convective heat transfer = 6×10^6 Btu/hr

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

$$\frac{\text{Heat content of flue gas}}{\text{Net heat release}} =$$

$$0.45 - \frac{(0.75 + 6.00) \times 10^6}{97.5 \times 10^6} = 0.38$$

flue gas T out = 1398°F, Figure 37

b) Calculate fluid temperature in:

$$T_{fl.in} = 483 - \frac{(0.75 + 6.00)}{23.4} (483 - 380) = 453^\circ\text{F}$$

c) Temperature Differences:

$$\text{Hot} = 1585 - 483 = 1102$$

$$\text{Cold} = 1398 - 453 = 945$$

$$\text{LMTD} = \frac{1102 - 945}{\ln \frac{1102}{945}} = 1021.5$$

d) Average Temperatures

$$\text{Oil} = (483 + 453)/2 = 468$$

$$\text{Tube Wall} = 468 + 100 = 568$$

$$\text{Gas} = 468 + 1022 = 1490$$

$$\text{Gas Film} = 468 + 1022/2 = 979$$

e) Calculate surface required

$$h_{cc} = \frac{2.14 (979 + 460)^{0.28} (0.342)^{0.6}}{4.5^{0.4}} = 4.72$$

$$h_{cr} = 2.93 \text{ From Figure 56}$$

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

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



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

$$h_{C_g} = (1 + 0.0984)(4.72 + 2.93) = 8.40$$

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

$$\text{Surface Area Required} = \frac{6 \times 10^6}{7.75(1022)} = 758$$

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

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:

$$\frac{\text{Heat content of flue gas}}{\text{Net heat release}} =$$

$$0.45 - \frac{(0.75 + 4.45) \times 10^6}{97.5 \times 10^6} = 0.40$$

flue gas T out = 1445°F, Figure 37

b) Calculate fluid temperature in:

$$T_{fl.in} = 483 - \frac{(0.75 + 4.45)}{23.4}(483 - 380) = 460^\circ\text{F}$$

c) Temperature Differences:

$$\begin{aligned} \text{Hot} &= 1585 - 483 = 1102 \\ \text{Cold} &= 1445 - 460 = 985 \end{aligned}$$

$$\text{LMTD} = \frac{1102-985}{\ln \frac{1102}{985}} = 1042$$

d) Average Temperatures

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

e) Calculate surface required

$$h_{cc} = \frac{2.14 (993 + 460)^{0.28} (0.342)^{0.6}}{4.5^{0.4}} = 4.73$$

$$h_{cr} = 3.05 \text{ From Figure 56}$$

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

$$f = \frac{10.40}{4.72 + 10.40 + 2.93} \left(\frac{44.45}{272} \right) = 0.0935$$

$$h_{C\phi} = (1 + 0.0935)(4.73 + 3.05) = 8.51$$

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

$$\text{Surface Area Required} = \frac{4.45 \times 10^6}{7.84(1042)} = 758$$

$$\text{No. of rows} = \frac{544.7}{272} = 2.003 \quad \underline{O.K.}$$

D) Determine the remaining number of finned rows required using

1" high, 0.05" thick carbon steel fins, 2 fins per inch.

1) Assume convective heat transfer = 4.45×10^6 Btu/hr

$$\begin{aligned} \text{Gross width} &= 52 \\ \text{Projected width} &= 6 [4.5 + 2(0.05)(1)(2)] = 28.2 \\ \text{Free width} &= 23.8" = 1.98' \end{aligned}$$

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

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

2) Temperature Differences:

$$\begin{aligned} \text{Hot} &= 1445 - 460 = 985 \\ \text{Cold} &= 820 - 380 = 440 \end{aligned}$$

$$\text{LMTD} = \frac{985 - 440}{\ln \frac{985}{440}} = 675$$

3) Average Temperatures

$$\begin{aligned} \text{Oil} &= (460 + 383)/2 &= & 420 \\ \text{Tube Wall} &= 420 + 100 &= & 520 \\ \text{Gas} &= 420 + 675 &= & 1095 \\ \text{Gas Film} &= 420 + 675/2 &= & 757 \end{aligned}$$

4) Calculate surface required:

$$h_{cc} = \frac{2.14 (757 + 460)^{0.28} (0.359)^{0.6}}{4.5^{0.4}} = 4.64$$

fin efficiency:

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

$$d_f/d_o = 6.5/4.5 = 1.445$$

From Figure 54 :

$$\text{Efficiency} = 0.84$$

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

$$h_{Cc} = h_f = 24.56$$

$$U_c = \frac{24.56(100)}{24.56 + 100} = 19.7$$

$$\text{Surface Area Required} = \frac{[23.4 - (4.45 + 0.75)] \times 10^6}{19.7(677)}$$

$$\begin{aligned} &= 1364 \\ \text{No. of rows} &= \frac{1365}{272} = 5.02 \end{aligned}$$

use 5 finned rows

*h_{cc} : 24.56 W/m² , negligible
Temp in gas*

F. Stack Design

1. General

The stack must create sufficient draft to overcome the flue gas side frictional drop. This includes the ΔP through the burners, the convection section, the damper, and the stack in a natural draft heater

2. Burners

The pressure drop across the burners is required to draw in air. Its magnitude is usually set by the burner manufacturer. It usually is between 75 and 85% of the available differential across the burner. It depends on the type of burner and the fuel. A minimum draft required at the burner is typically 0.25 inches 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.

$$P_v = 12 \left(\frac{V^2}{2g_c} \right) \left(\frac{\rho_g}{\rho_w} \right) = 0.0030V^2\rho_g = 0.0030 \left(\frac{G^2}{\rho_g} \right)$$

where:

V = Velocity, ft/sec

g_c = Gravitational constant, 32.17 lb_mft/lb_fsec²

ρ_g = Density of flue gas, lbs/ft³

ρ_w = Density of water, lbs/ft³

Flue gas densities are shown in Figure 33. In the convection section, The gas density is evaluated at the same average gas temperature 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
Damper	1.5 D
Stack friction	H/50D
Stack exit	<u>1.0</u>
TOTAL	3.0 plus H/50D

where:

H = Stack Height, ft

D = Stack diameter, ft

Equation 16 and Figure 33 are used to calculate the stack velocity head as in the convection section.

5. Draft

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

$$\Delta P = 12 H \frac{(\rho_a - \rho_g)}{\rho_w} = 0.187 H (\rho_a - \rho_g)$$

The available draft per 100 feet of height is shown in Figure 34. The draft is a function of the flue gas temperature. The draft should be calculated based on a corrected temperature. This corrected temperature is lower than the flue gas temperature leaving the convection section. This accounts for heat losses as the flue gas rises up the stack. The heat losses are a function of stack insulation and dimensions, ambient conditions and the flue gas temperature entering the stack. The corrected temperature should be no less than 35°F below the entering flue gas temperature.

6. Furnace Pressure

All sections of the heater should be below atmospheric pressure. Leaks through peepholes, tube-openings, or cracks in the setting will be air moving into the furnace. A positive pressure (greater than atmospheric) can force hot gases out of the heater. This can weaken structural members. It can pose a safety hazard to operators.

The bridgewall area (area between the radiant and convection sections) is the point of highest furnace pressure (minimum draft). A draft of 0.05 to 0.1 inches of water is acceptable in this area. Greater draft can cause leakage problems.

Air leakage will increase as the draft increases. This can give misleading readings of excess air. Operators may assume excess air is sufficient if the O₂ analyzer implies sufficient air is present. This can be a serious error. The analyzer can be reading the total air to the radiant section- the burner air plus the leakage air. It may not be reading the air available at the burners. Burners can become unstable as insufficient air enters them. Afterburning could occur.

7. Stack Size

Minimum stack heights are set by draft, safety and pollutant dispersal requirements. Flue gases must not interfere with operators on nearby platforms. The proximity of tall structures may produce downdrafts that can affect personnel. Pollutants are diluted over a wider area as stack heights are increased. The stack is sized so that the total friction drop is no greater than the available draft if there are no additional limitations. The most economical combination of diameter and height may be selected. Usual design flue gas stack velocities are typically 25 - 30 feet per second.

F. Furnace Rating Procedure

1. General

The principles outlined above can be applied to specific furnace designs. The following describes general concepts that are used. They present a procedure requiring a minimum of trial and error.

2. Basic Design Considerations

The function of a process furnace is to increase the heat content of a prescribed quantity of material. Other contingencies are often imposed. These include provision for upstream or downstream fouling, start-up, future operation, etc.

Furnace efficiency determines the basic operating cost of a heater. The optimum efficiency for a particular application depends on a number of factors. These include fuel costs, the process fluid temperatures, the tube material and the user's policy on investment payout. Overall efficiencies are usually in the 82 to 87 percent (LHV) range for process heaters without air preheat.

Each fuel has its own characteristics. An efficiency curve should be developed for individual fuels. Table II shows furnace efficiencies based on fuel oil and exit flue gas temperatures. This chart is fairly representative for fuel oil firing. Another general curve for estimating efficiency for is Figure 35.

Appendix A goes into efficiency in more detail.

Excess combustion air has a significant effect on efficiency. The greater the excess air, the lower the efficiency. Too little excess air can cause operating difficulties. This includes incomplete combustion, poor flame characteristics, instability, etc. Typical excess air rates are 10 to 25 percent for gas firing and 25 to 40 percent for oil.

The maximum heat absorption rate in the radiant section is usually specified by the user on the basis of past experience. High radiant rates result in smaller, less expensive heaters. High rates can limit operation if additional throughput is desired in the future. Lower rates give lower tube wall and refractory temperatures. This tends toward less maintenance. Lower rates may increase cycle times in coking services. Normal design figures for clean services with good intube heat transfer coefficients are in the range of 10,000 to 13,000 Btu/ft²hr of outside circumferential tube surface (tubes located against refractory wall).

The maximum heat release per unit firebox volume can be specified. Unit heat release is a rough measure of the probability of flame impingement on the tubes. Current practice employs fitting the burners to the heater or vice versa.

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

Certain aspects of the design can be established before the detailed design begins. The process requirements and other limitations are required. The percentage of excess air can be selected based upon the fuel and past experience. The heater efficiency can be selected based upon the process inlet temperature, the fuel fired and the excess air. The net heat release can be calculated using the specified duty and efficiency. Flue gas rates can be determined from the fuel properties.

Typical radiant-convection heaters will absorb 65 -70% of the total duty in the radiant section. Estimate the radiant heat absorption, q_R . Estimate the temperature at the crossover between the convection and radiant sections from enthalpy or specific heat data for the process fluid. Determine the average fluid

temperature in the radiant section. Calculate the average tube wall temperature. Using the specified allowable radiant transfer rate, calculate the total radiant tube surface.

The convection section, bare tube surface often approaches that in the radiant section. Select a tube size and pass arrangement that will give the estimated total surface. Estimate the pressure differential to see if it may be satisfactory. Proceed with your calculations and adjust your numbers accordingly until the design is correct.

4. Radiant Section Rating

Choose a center-to-center spacing for the radiant tubes which is compatible with the selected tube size. Short radius return bends (two nominal diameter, center to center spacing) are most frequently employed. Wider tube spacing permits greater radiant absorption rates. It can produce lower firebox temperatures, and improved circumferential heat distribution. Closer spacing permits more tubes to be installed in a given size firebox.

Determine the length and number of tubes required in the radiant section using the approximate radiant tube surface required. Use firebox dimensions that will accommodate the required total tube length. The exact proportions depend on judgement and past experience. Long furnaces minimize the number of return bends decreasing total cost. Short and wide fireboxes usually give more uniform heat distribution.

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.

Calculate the heat content of any preheated air or fuel stream. Determine heat content to total net heat release ratio of each. The heat content of these

streams can usually be neglected if their temperature is below 100°F. Estimate the heat loss through the setting.

Assume a temperature of the flue gas in the radiant section, t_g . Read the gas emissivity from Figures 23a-d at this temperature. Determine the exchange factor from Figure 24. Calculate the ratio $q_n/\alpha A_{cp}F$.

Read the corresponding value of q_g/q_n from Figure 37. Using Equation 10, calculate $q_R/\alpha A_{cp}F$. Plot it against average furnace temperature on Figure 28.

The computed point will be on the absorption curve for the average tube wall temperature if the assumed value of t_g was correct. There usually will be a discrepancy. Select another value of t_g on the other side of the absorption curve. Repeat the above procedure.

Join the two points on opposite sides of the curve with a straight line. The point of intersection of this line with the absorption curve is the firebox temperature. Apply the appropriate factor to correct to the exit temperature, t_{g2} , as experience indicates. Read q_{g2}/q_n from Figure 37. Calculate q_R from Equation 9. This is the heat actually absorbed in the radiant section.

Check the radiant absorption for the design limitations. Divide the heat absorption by the total exposed tube area to get the average heat flux. Repeat the rating if it is beyond acceptable limits. Adjust the size of the radiant section as necessary. Calculate the crossover temperature from the convection to radiant section. Use process enthalpy data for this. Repeat the radiant section calculations if the crossover temperature is appreciably different from that assumed. Tube wall temperature has a relatively small effect on the radiant absorption rate. Such a refinement is usually unnecessary.

The crossover temperature above corresponds to the transition from true convection tubes to shield tubes. Many heaters include the shield tubes as part of the convection section. The cross-over piping is actually between the shield and radiant tubes. The temperature in the crossover piping will be higher than that calculated in the rating for such heaters. It will be higher by an amount corresponding to the heat absorption in the shield tubes.

5. Convection Section Rating

The convection section duty equals the difference between the heat absorbed in the radiant section and the total heater duty. The temperature of the flue gas entering and leaving the convection section has been calculated as part of the radiant section rating. The incoming fluid temperature is set by the original design conditions. The flue gas temperature has been set by the heater efficiency.

$$q_n + q_a + q_f = q_L + (q_R + q_c) + q_s \quad (18)$$

$$\frac{q_s}{q_n} = 1 + \frac{q_a}{q_n} + \frac{q_f}{q_n} - \frac{q_L}{q_n} - \frac{q_R + q_c}{q_n}$$

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 0.3 to 0.4 pounds per square foot per second. Greater velocities are usually acceptable if there is a high stack requirement or an induced draft fan. Calculate the mass velocity, G.

Determine the gas-film coefficients using Figures 29, 30 and 31, and/or Equations 12 to 14. Calculate the intube coefficient. Calculate the overall heat transfer coefficient from Equation 15. Determine the convection area from the usual heat transfer equation:

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

The calculated convection surface rarely will correspond to an integral number of tube rows. The computed surface is rounded up to an integral number of tube rows. The excess is considered as a margin of safety. The convection section orientation may have to be changed if a more exact rating is desired.

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 convection section heat coefficients.

6. Stack Design

The stack must provide enough draft to hold the firebox at a negative pressure. The stack draft must overcome frictional losses through the convection section and stack. Calculate the draft caused by the firebox itself. Determine there is sufficient draft for the burners to function properly. Determine the draft required atop the convection section or downstream ductwork (whichever is greater).

Determine the stack temperature correcting for stack heat losses. Select a stack diameter to give a stack velocity of about 25 to 30 feet per second. Calculate the velocity head. Estimate the stack height, and calculate its friction loss. Allow 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. Calculate the required height. Confirm the stack height meets all safety and pollution requirements.

Altitude Correction Factors for Gas-Density and Draft

Altitude Above Sea Level, feet	Multiply Density and Draft by...
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

$$T_o = 685^{\circ}\text{F}$$

$$P_o = 15 \text{ psig}$$

$$H_o = 438 \text{ Btu/lb}$$

at inlet

$$T_i = 490^{\circ}\text{F}$$

$$P_i = 195 \text{ psig}$$

$$H_i = 245 \text{ Btu/lb}$$

$$\Delta H = 193 \text{ Btu/lb}$$

$$\text{Duty} = 1,240,000 \times 193 = 239.32 \times 10^6 \text{ Btu/hr}$$

$$\approx 237.92 \text{ (Figure 38a)}$$

Vaporization at outlet = 68%

Process specs = 55%

We will use the higher vapor point. It will be safer when pressure drop is considered.

We will use the heater arrangement shown on Figure 42. We will use a radiant average heat transfer rate of 12,000 Btu/Hr/ft². This is taken from from Table 4, Figure 43 for Crude service.

We base our efficiency on the inlet fluid temperature, 30% excess air and fuel oil firing. We will assume a 300°F approach with a bare tube convection bank. The stack temperature is $300 + 490 = 790^\circ\text{F}$. We can determine the heater efficiency from Figure No. 36: gross efficiency = 71.25%; net efficiency = 76.25%. Subsequent calculations will be based on the net heating value of the fuel- we will use the net efficiency rather than the gross.

The net heat release is calculated as follows:

$$\text{Net Heat Release} = \frac{239.32 \times 10^6 \text{ Btu/hr}}{0.7625} = 313.86 \times 10^6 \text{ Btu/hr}$$

The flue gas rate can be determined from Figure 27. There are 1,080 lbs flue gas/MM Btu net heat release.

$$\begin{aligned} \text{Total Flue Gas} &= 1080 \text{ lbs flue gas/MM Btu} \times (313.86 \times 10^6 \text{ Btu/hr}) \\ &= 338971 \text{ lbs/hr} \end{aligned}$$

NOTE: 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.

RADIANT SECTION

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

$$\text{Radiant Duty} = 0.7 \times (239.32 \times 10^6) = 167.52 \times 10^6 \text{ Btu/hr}$$

The radiant surface required at an average heat flux of 12000 Btu/ft²hr (Table 4) is equal to the following:

$$\text{Radiant Surface} = \frac{167.52 \times 10^6 \text{ Btu/hr}}{12000 \text{ Btu/ft}^2\text{hr}} = 13960 \text{ ft}^2$$

From Table 5, Figure 44 we will select 6"N.P.S., Schedule 40 average wall thickness. A cold oil velocity of 3.5 ft/sec can be used to approximate the number of passes required in a crude heater. The number of passes required would be:

$$\frac{99000 \text{ B/D}}{1000(3.5 \text{ ft/sec})} \times 0.31 = 8.76 \text{ - Try 8 passes.}$$

Return 1 Row of Shield section shall be considered as Radiation section.

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It would be too complicated to use more than 8 passes in this style of box. We will attempt using an 8 passes in our design.

Assume tube length = 60 ft

Surface per foot of tube (Table 5) = 1.734 ft²/ft

$$\text{Length of tubing required} = \frac{13960}{1.734} = 8051 \text{ ft}$$

Assume tube length of 60 ft,

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

$$\text{tubes/pass} = \frac{134.2}{8} = 16.8 \text{ tubes/pass- use 17 tubes/pass}$$

135 tubes
60 ft length

Tube length 1. and
Diameter = 2 in
8 shield tubes

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

$$\text{per tube} = 60 + 25.3 = 85.3 \text{ ft.}$$

$$\text{Equivalent length per pass} = 85.3 (17) = 1450.1 \text{ ft.}$$

Surface = 135 x 1.734
= 233.085 ft²
58.5 ft (1.734 ft²/ft)
= 101.66 ft²/tube

$$G = \text{Mass velocity} = \frac{1240000 \text{ lb/h}}{3600 \text{ sec/hr}(8 \text{ passes})(0.2008 \text{ ft}^2)} = 214.4 \text{ lbs/ft}^2\text{sec}$$

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, therefore $H = 438 \text{ Btu/Lb}$

Crossover conditions:

$$\Delta H_{co} = \frac{\text{Radiation section heat absorbed}}{\text{Fluid heated}}$$

$$\Delta H_{co} = \frac{167.52 \times 10^6 \text{ Btu/Hr}}{12400000 \text{ lb/h}} = 135 \text{ Btu/Lb}$$

$$\Delta H_{co} = H_o - H_{co} \quad H_{co} = H_o - \Delta H_{co}$$

$$H_{co} = 438 - 135 = 303 \text{ Btu/lb}$$

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

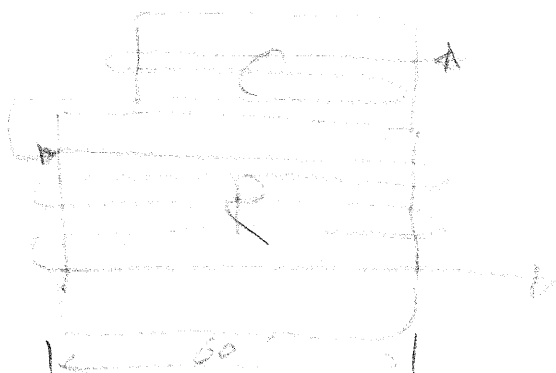
Effective Tube Area

A_{cp}

$$= (52.5)(128)(1 \text{ ft})$$

$$= 6720 \text{ ft}^2$$

$A_{cp, shield}$



Estimated Pressure Drop

$$\frac{\Delta P}{100 \text{ ft}} = \frac{0.517 f G^2}{D_i \rho}$$

where:

f = friction factor (0.004 for this example)

D_i = 6.065 in. (pipe size ID)

G = 214.42 lb/sec. sq.ft.

ρ = density @ Process Conditions = $\frac{1}{\text{specific volume}}$
 $= \frac{1}{1.8} = 0.555 \text{ ft}^3/\text{lb}$

$$\frac{\Delta P}{100 \text{ ft}} \text{ at outlet} = \frac{0.517(0.004)(214.4)^2}{6.025(0.555)} = \frac{15.6765}{0.555} = \frac{28.22 \text{ psi}}{100 \text{ ft}}$$

Estimated specific volume at crossover = 0.125 ft³/lb,

therefore, ρ = (1/0.125 ft³/lb) = 8.00 lb/ft³

$$\frac{\Delta P}{100 \text{ ft}} \text{ at crossover} = \frac{0.517(0.004)(214.4)^2}{6.625(0.125)} = \frac{15.6736}{8.00} = \frac{1.96 \text{ psi}}{100 \text{ ft}}$$

$$\log \text{ mean } \Delta P = \frac{28.22 - 1.96}{\ln(28.22/1.96)} = \frac{9.846 \text{ psi}}{100 \text{ ft}}$$

$$\text{Estimated radiant section } \Delta P = \frac{9.846 \text{ psi}}{100 \text{ ft}} (1450.1 \text{ ft}) = 142.8 \text{ psi}$$

OK (140 psi assumed)- accepted estimate deviation = 10 - 20%

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

$$\text{Average fluid temp.} = \frac{T_{co} + T_o}{2} = \frac{554 + 685}{2} = 619.5^\circ\text{F}$$

close enough

Tube Wall Temperature Check

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

$$h = \frac{K}{D} \left[\frac{CZ}{K} \right]^{1/3} \left(\frac{Z}{Z_w} \right)^{0.14} \Phi \left(\frac{DG}{Z} \right) \quad (\text{see Figure 45})$$

where:

h = Film coefficient, Btu/hrft²°F

K = Film thermal conductivity, Btu/hr/ft²/(°F/Ft)

L = Heated tube length, ft.

D = Inside tube diameter, inches

Z_w = Absolute viscosity at inside tube wall temperature, cp

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

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

Z = Absolute viscosity, cp at average temperature of fluid, cp

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

Crude properties: °API = 34.8, Characterization factor = 11.9.

$K = 0.0654$

$L = 1450.1$ ft.

$D = 6$ in. (6.065 I.D.)

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

$Z = 0.3$ Centistokes @ 620°F = 0.3 (flowing Specific gravity)

$= 0.3(0.625) = 0.1875$ cp

(specific gravity from UOP PD-194)

$G = 214.42$ lb/ft²hr

$C = 0.76 \times 1.005 = 0.7638$

Corrected by UOP K factor (UOP Chart F-I)

$$\text{Pseudo- Reynolds Number} = \frac{DG}{Z} = \frac{6.065(214.42)}{0.1875} = 6935$$

$$\text{From Fig. 45: } \phi \left(\frac{DG}{Z} \right) = 23,000$$

$$h = \frac{0.0654}{6.065} \left[\frac{0.7638(0.1875)}{0.0654} \right]^{1/3} (1) (23000) = 322$$

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

$$\Delta T_{\text{metal}} = 12000 \left(\frac{6.625}{6.065} \right) \left(\frac{1}{322} \right) = 40.7$$

The temperature difference through the tube wall, is:

$$\Delta T_{\text{film}} = 12000 \left(\frac{6.625}{2(190)} \right) \ln \left(\frac{6.625}{6.065} \right) = 18.5$$

$$\text{Total } \Delta T = \Delta T_{\text{film}} + \Delta T_{\text{metal}} = 40.7 + 18.5 = 59.2$$

$$\text{Average Tube Wall Temperature} = \frac{554 + 685}{2} + 59.2 = 679^{\circ}\text{F}$$

RADIANT SECTION LAYOUT

The lay-out of the heater is shown on Fig. 42 and 46. The heater dimensions are: 32.5 feet wide by 33 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

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

$$\text{Surface per tube} = 58.5 \text{ ft./tube} \times 1.734 \text{ sq. ft./ft} = 101.44 \text{ sq. ft./tube}$$

$$\text{Total Surface} = 136 \text{ tubes} \times 101.44 \text{ sq. ft./tube} = 13796 \text{ sq. ft.}$$

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

Effectiveness factor for wall tubes, $\alpha = 0.92$ (Fig. 19)

Effectiveness factor for shield tubes, $\alpha = 1.00$ (α for shield tubes is consider unity.)

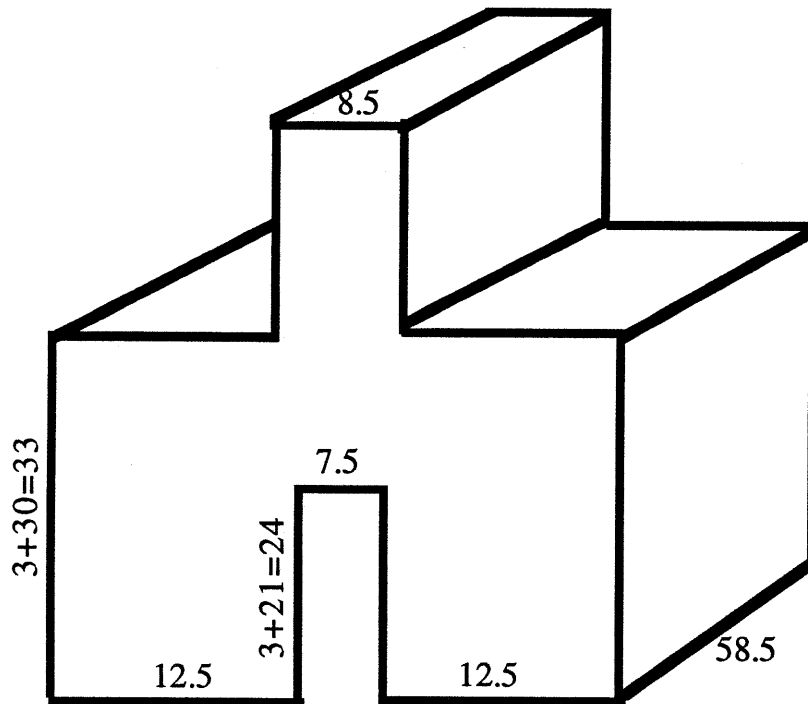
Shield tubes, $A_{cps} = \text{exposed length/tube} \times \text{No. of tubes} \times \text{center-center distance.}$

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

Wall tubes, $A_{cpw} = 58.5 \text{ ft/tube} \times 128 \text{ tubes} \times 1 \text{ ft} = 7488 \text{ sq. ft.}$

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

$$= (1) (468) + (.92)(7488) = 468 + 6889 = 7357$$



$$\begin{aligned} \text{Envelope area} &= 2(32.5)(33) + 2(58.5)(33) + 2(58.5)(32.5) \\ &\quad + 2(58.5)(24) - 8.5(58.5) - 2(7.5)(24) \\ &= 11759 \text{ sq. ft.} \end{aligned}$$

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

$$= 11759 - 7357 = 4402 \text{ sq. ft.}$$

$$\frac{A_R}{\alpha A_{cp}} = \frac{4402}{7357} = 0.598$$

Dimensions each side = 12.5 : 33 : 58.5 = 1 : 2.64 : 4.68

$$L = \text{Mean Beam Length} = \frac{1.3 + 1.8}{2} \quad (\text{Smallest dimension})$$

$$= 1.55 (12.5) = 19.38$$

P = Partial Pressure $\text{CO}_2 + \text{H}_2\text{O} = 0.225$ Figure 22 @ 30% E.A.

$$PL = 0.225 (19.38) = 4.36 \text{ atm. ft.}$$

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

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

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

Estimated average firebox temperature = 1500°F

Flame emissivity: For $T_G = 1500^\circ\text{F}$ & $PL = 4.36 \text{ atm. ft.}$

Heat losses are 2%
for single fired typical
to double fired:
2.5 to 3%

Tube Metal Temperature, °F	Flame Emissivity, P_f
600	0.545
679	0.543*
800	0.542

*By interpolation from Fig. 23a - d

Exchange factor, $F = 0.607$ at $P_f = .543$ & $A_R/\alpha A_{cp} = 0.598$

(Fig. 24)

$$\alpha A_{cp} F = (0.607)(7357) = 4466 \text{ sq. ft.}$$

$$\frac{q_n}{\alpha A_{cp} F} = \frac{313.86 \times 10^6 \text{ Btu/h}}{4466 \text{ ft}^2} = 70282 \text{ Btu/ft}^2\text{hr}$$

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

$q_{g2}/q_n = 0.43$, Fig. 37 @ 30% E.A.

$$\begin{aligned}\frac{q_R}{\alpha A_{cp} F} &= \left[1 + \frac{q_a}{q_n} + \frac{q_f}{q_n} - \frac{q_{g2}}{q_n} + \frac{q_L}{q_n} \right] \frac{q_n}{\alpha A_{cp} F} \\ &= [1 + 0 + 0 - 0.43 - 0.02] 70282 \\ &= (.55) (70282) = 38655 \text{ BTU/ft}^2 \text{ hr}\end{aligned}$$

Plot the calculated $q_R/\alpha A_{cp} F$ of 38655 at the assumed firebox temperature of 1500°F on Figure 25b). The point lies to the left of the absorption curve.

Try another calculation at 1750°F emissivity, by interpolation @ 30% E.A.

Flame emissivity: For $T_G = 1750^\circ\text{F}$ & $PL = 4.36 \text{ atm. ft.}$

Tube Metal Temperature, °F	Flame Emissivity, P_f
600	0.525
679	0.521*
800	0.520

*By interpolation from Fig. 23a - d

Exchange factor, $F = 0.588$ at $P_f = .521$ & $A_R/\alpha A_{cp} = 0.598$ (Fig. 24)

$$\alpha A_{cp} F = (0.588)(7357) = 4326 \text{ sq. ft.}$$

$$\frac{q_n}{\alpha A_{cp} F} = \frac{313.86 \times 10^6 \text{ Btu/h}}{4326 \text{ ft}^2} = 72553 \text{ Btu/ft}^2 \text{ hr}$$

Then, $t_g = 1750^\circ\text{F}$

$q_g/q_n = 0.52$, Fig. 37 @ 30% E.A.

$$\begin{aligned}\frac{q_R}{\alpha A_{cp} F} &= \left[1 + \frac{q_a}{q_n} + \frac{q_f}{q_n} - \frac{q_{g2}}{q_n} + \frac{q_L}{q_n} \right] \frac{q_n}{\alpha A_{cp} F} \\ &= [1 + 0 + 0 - 0.52 - 0.02] 72553 \\ &= (0.46) (72553) = 33375 \text{ BTU/ft}^2 \text{ hr}\end{aligned}$$

Plot this point on Fig. 25b. Joining the two calculated points with a straight line. The straight line crosses the absorption curve at 1612°F. This is the pseudo-firebox box temperature.

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.

$$\text{therefore, } t_g = t_{g2}$$

Now, again from Fig. 37

$$q_{g2}/q_n = 0.47, \text{ Fig. 37 @ 30\% E.A.}$$

$$q_R = [1 - 0.02 - 0.47] 313.86 = 160.07 \times 10^6 \text{ Btu/hr}$$

Calculate the average radiant section heat flux:

$$q_R/A_{Rt} = \frac{160.07 \times 10^6 \text{ Btu/hr}}{13796 \text{ ft}^2} = 11603 \text{ Btu/ft}^2\text{hr} < 12000$$

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

$$\text{Enthalpy change} = \frac{160.07 \times 10^6 \text{ Btu/hr}}{1240000 \text{ lb/h}} = 129 \text{ Btu/lb}$$

$$H_{co} = H_o - H_{co} = 438 - 129 = 309 \text{ Btu/lb}$$

$$\text{Crossover temperature} = 558^\circ\text{F vs } 554^\circ\text{F (Assumed) - Fig. 40a}$$

$$\text{Average radiant fluid temperature} = \frac{558 + 685}{2} = 621^\circ\text{F vs. } 619.5^\circ\text{F} \\ (\text{O.K., close enough})$$

Average radiant section tube wall temperature :

$$\Delta T_{\text{metal}} = 11603 \left(\frac{6.625}{6.065} \right) \left(\frac{1}{322} \right) = 39.4$$

$$\Delta T_{\text{film}} = 11603 \left(\frac{6.625}{2(190)} \right) \ln \left(\frac{6.625}{6.065} \right) = 17.9$$

$$\text{TWT}_{\text{ave}} = 621 + 39.4 + 17.9 = 678$$

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

$$\text{Convection section duty} = 239.32 - 160.07 = 79.25 \times 10^6 \text{ Btu/hr}$$

$$\frac{\text{Stack Heat Content}}{\text{Net Heat Release}} = \frac{q_s}{q_n} = 1 + \frac{q_a}{q_n} + \frac{q_f}{q_n} - \frac{q_R + q_C}{q_n} - \frac{q_L}{q_n}$$

$$= 1 + 0 + 0 - 0.763 - 0.02 = 0.217$$

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

$$\text{Stack Temperature} = 830^\circ\text{F} \quad (\text{Figure 37})$$

Radiation Losses Through the Shield Tubes

$$(1 - \alpha)A_{cp}F = (1 - 0.76)(468)(0.598) = 67.2 \text{ ft}^2$$

where $F = 0.598$ is determined by interpolation at our firebox temperature at 1612°F .

From Fig. 58 and $T_{W_{est}} = 621^\circ\text{F}$

$$\frac{q_{SL}}{(1 - \alpha)A_{cp}F} = 30500 \text{ Btu/ft}^2\text{hr}$$

$$q_{SL} = 30500 (67.2) = 2.05 \times 10^6 \text{ Btu/Hr}$$

Determining flue gas temperature to the convection section

$$\frac{\text{Stack Heat Content of Firebox Gases}}{\text{Net Heat Release}} @ 1612^\circ\text{F} = 0.47 \quad (\text{Figure 37})$$

$$\frac{\text{Stack Heat Content of Flue Gas to Conv. Section}}{\text{Net Heat Release}} = 0.47 - \frac{2.05 \times 10^6}{313.86 \times 10^6} = 0.46$$

From Figure 37:

$$\text{Flue gas temperature to the convection section tubes} = 1597^\circ\text{F}$$

Temperatures

Temperature Differences:

$$\Delta T_{\text{Hot}} = T_g - T_{co} = 1597 - 558 = 1039^\circ\text{F}$$

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

$$\text{Log Mean Temp} = \frac{\Delta T_{\text{hot side}} - \Delta T_{\text{cold side}}}{\ln \frac{\Delta T_{\text{hot side}}}{\Delta T_{\text{cold side}}}} = \frac{1039 - 340}{\ln \frac{1039}{340}} = 626$$

Average Temperatures:

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

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

$$\text{Gas} = \text{Oil Av. Int. Fluid} + \text{Log Mean Temp.} = 524 + 626 = 1150^{\circ}\text{F}$$

$$\text{Gas Film} = \text{Oil Av. Int. Fluid} + 1/2 \text{ Log Mean Temp.} = 524 + 1/2 (626) = 837^{\circ}\text{F}$$

Convection Section Lay-out

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

$$\text{Gross width} = (\text{No. tubes/row} + 1/2 \text{ tube/row}) \times \text{tube spacing, in.}$$

$$= (8 + 1/2) \times 12 = 102 \text{ inches}$$

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

$$\text{Free width area} = \text{Free width} \times \text{exposed tube length} = 4.5 \times 58.5 = 263.25 \text{ ft}^2$$

$$\text{Flue Gas Mass Velocity} = G = \frac{\text{Total Flue Gas Mass}}{\text{Free Width Area}} = \frac{94.2 \text{ lb/sec}}{263.25 \text{ ft}^2}$$

$$= 0.356 \text{ lb/ft}^2\text{sec} \text{ (Permissible flue gas mass velocity - .5 lb/ft}^2\text{sec)}$$

typical maximum under natural draft operation)

Film Transfer Correlation

For $G = .356$; 6.625" O.D., & Gas Film Temperature = 837°F

From Fig. 29 or the following equation:

$$h_{Cc} = \frac{2.14(T_f)^{0.28}(G_{\max})^{0.6}}{D_i^{0.4}} = \frac{2.14(837 + 460)^{0.28}(0.356)^{0.6}}{6.625^{0.4}} = 4.02 \text{ Btu/hrft}^2\text{ }^{\circ}\text{F}$$

For an average gas temperature of 1150°F and a tube wall temperature of 626°F

From Fig. 30: $h_{Cr} = 2.37 \text{ Btu/hrft}^2\text{°F}$

For an average wall temperature of 626°F

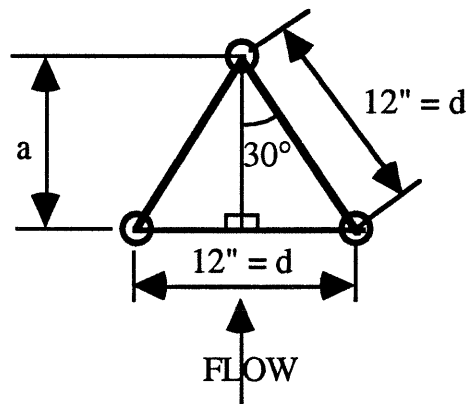
From Figure 31 or the following equation:

$$h_{Cw} = 9.46 \left(\frac{T_t}{1000} \right)^3 = 9.46 \left(\frac{626 + 460}{1000} \right)^3 = 12.12 \text{ Btu/hrft}^2\text{°F}$$

$$\text{then, } h_{Cc} + h_{Cr} + h_{Cw} = 4.02 + 2.37 + 12.12 = 18.51 \text{ Btu/hrft}^2\text{°F}$$

Row-to-Row Tube Spacing

For an equilateral convection section lay out we have:



$$a = \text{row-to-row tube spacing} = d (\cos 30^\circ) = d (\sin 60^\circ)$$

$$= 12 \text{ inches} (1\text{ft}/12\text{inches}) 0.866 = 0.866 \text{ ft}$$

$$A_{Cw} = \text{wall area per row} = \text{two side walls } (a)(L) = 2(0.866)(58.5) = 101.322 \text{ ft}^2$$

where L = effective length per convection tube.

$$A_{Ct} = \text{tube area per row} = \text{No. tubes/row (area/tube)}$$

$$= 8 (58.5)(1.734) = 811.512 \text{ ft}^2$$

$$\text{Wall radiation correction factor} = f = \frac{h_{Cw}}{h_{Cc} + h_{Cr} + h_{Cw}} \left(\frac{A_{Cw}}{A_{Ct}} \right)$$

$$= \frac{12.12}{4.02 + 2.37 + 12.12} \left(\frac{101.322}{811.512} \right) = 0.0818$$

Overall Outside Coefficient

$$\begin{aligned}\text{Total Apparent Gas Film Coefficient} &= h_C = (1 + f) (h_{Cc} + h_{Cr}) \\ &= (1 + 0.0818)(4.02 + 2.37) = 6.91 \text{ Btu/hrft}^2\text{ }^\circ\text{F}\end{aligned}$$

$$\text{Intube Film Coefficient} = h_{Ci} = \frac{K}{D} \left[\frac{CZ}{K} \right]^{1/3} \left(\frac{Z}{Z_w} \right)^{0.14} \phi \left(\frac{DG}{Z} \right) \quad (\text{see}$$

Figure 45)

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

Crude properties: °API = 34.8, Characterization factor = 11.9.

$$K = 0.0695$$

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

$$D_i = 6.065 \text{ inches}$$

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

$$Z = 0.46 \text{ Centistokes @ } 474^\circ\text{F} = 0.3 \text{ (flowing Specific gravity)}$$

$$= 0.46(0.7) = 0.322 \text{ cp}$$

(specific gravity from UOP PD-194)

$$G = 214.42 \text{ lb/ft}^2\text{hr}$$

$$C = 0.68 \times 1.005 = 0.6834$$

Corrected by UOP K factor (UOP Chart F-1)

$$\text{Pseudo- Reynolds Number} = \frac{DG}{Z} = \frac{6.065(214.42)}{0.322} = 4039$$

$$\text{From Fig. 45: } \phi \left(\frac{DG}{Z} \right) = 14800$$

$$h = \frac{0.0695}{6.065} \left[\frac{0.6834(0.322)}{0.0695} \right]^{1/3} (1) (14800) = 249$$

Convection Section Over-All Transfer Outside Coefficient = U_o

$$\begin{aligned}
 &= \frac{1}{\frac{D_o}{D_i h_{di}} + \frac{D_o}{D_i h_i} + \frac{t_w}{k_m} \frac{D_o}{D_L} + \frac{1}{h_c} + \frac{1}{h_{do}}} \\
 &= \frac{1}{\frac{6.625}{6.065(1000)} + \frac{6.625}{6.065(249)} + \frac{0.28/12}{190/12} \frac{6.625}{6.341} + \frac{1}{6.91} + \frac{1}{500}} \\
 &= 6.50 \text{ Btu/hrft}^2\text{°F}
 \end{aligned}$$

or, if we made some simplifying assumptions, $U_o = \frac{h_c h_i}{h_c + h_i} = \frac{6.91(249)}{6.91 + 249} = 6.72$

For comparison, the approximate over-all coefficient from Fig. 32 is 6.66 for 6.625" O.D. tubes and an average gas temperature of 1150°F

Convection Section Area

Convection section area is estimated from the usual heat-transfer equation (using the in-depth, overall outside coefficient):

$$A_c = \frac{q_c}{U_o(LMTD)} = \frac{(79.25 - 2.07) \times 10^6 \text{ Btu/h}}{6.50 \text{ Btu/hrft}^2\text{°F}(626\text{°F})} = 18968 \text{ ft}^2$$

Note: The subtracted value is the radiant leakage into the shield tubes. It has already been considered.

$$\text{No. of Rows} = \frac{\text{Convection Area}}{\text{Area/Tube}} = \frac{18968}{58.5(1.734)} = 187 \text{ tubes}$$

$$\text{No. of Tubes} = \frac{\text{No. of Tubes}}{\text{Tubes/row}} = \frac{187}{8} = 23.4 \text{ rows}$$

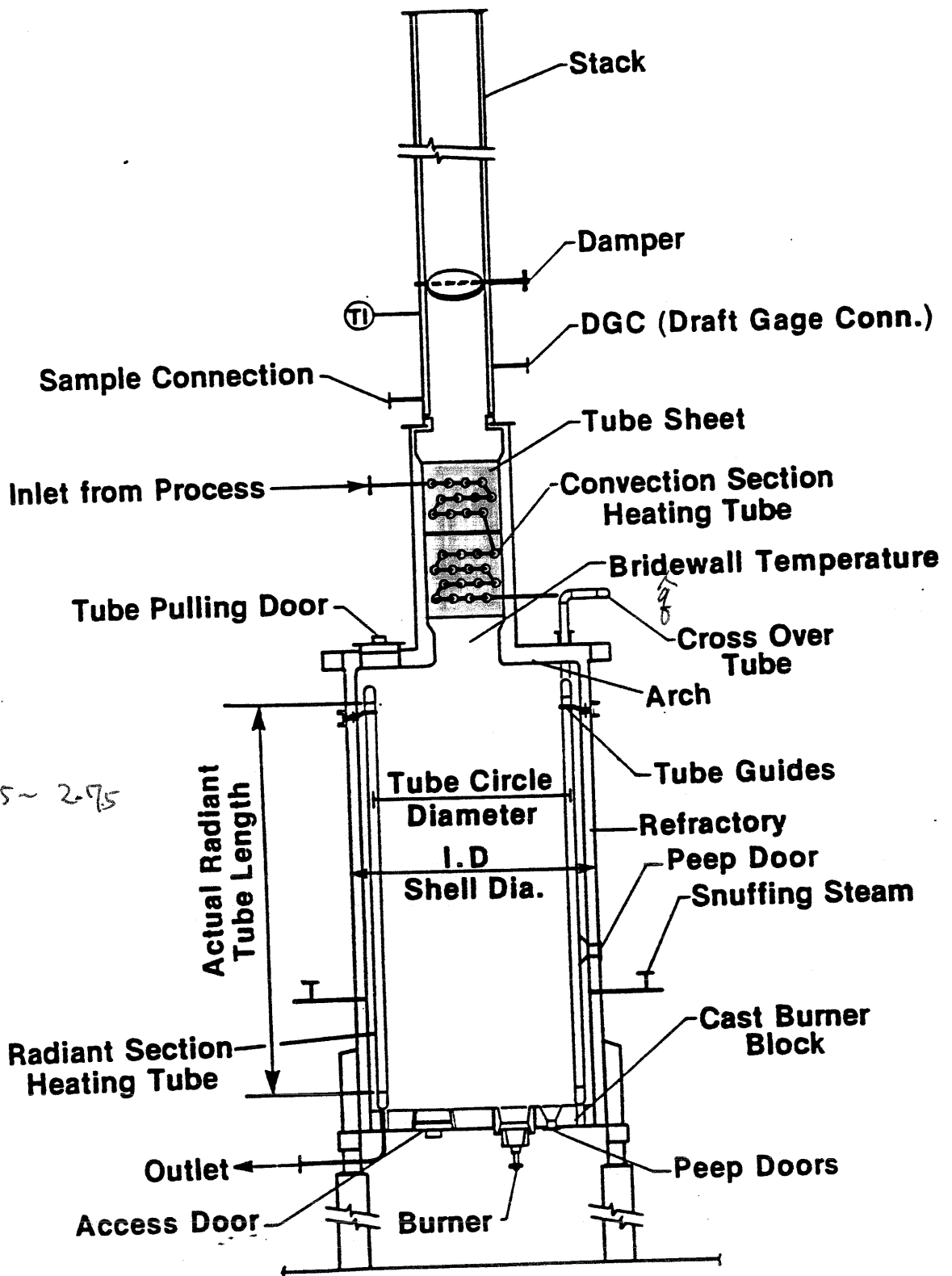
24 rows required, use 25 rows (see following)

24 convection section rows are required to meet our remaining duty requirement. Note the flow through our furnace. The heater inlet (convection section inlet) and the convection section outlet are on the same side of the furnace. The convection section outlet immediately feeds the radiant section inlet on the same side. The heater outlet (radiant section outlet) will exit the opposite side of the furnace since the radiant section has 17 tubes per pass. The furnace inlets and outlets are on the opposite sides of the furnace.

There is wasted piping, more than one furnace length, when the furnace inlet and outlet are on opposing sides. We can either add another row of tubes or remove a row of tubes to place the terminals on the same side. An additional row will have a small effect on efficiency. The removal of one row will decrease the efficiency for the same reason. The decrease will be slightly greater than the decrease due to a row addition. This decrease in efficiency will cause the furnace to fire slightly higher, increasing our average radiant flux minutely. The addition of a 25th row is warranted for piping considerations.

Note: Use of the simplified method will produce less surface, lower efficiency, greater stack temperature and a higher average radiant heat transfer rate. Our simplified example ($h_c = 6.72$) will result in 18346 ft², 181 tubes and a need for 22.6 (23) rows. The difference between the two methods can become more pronounced. This can occur if fouling is considerably higher than in our example.

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

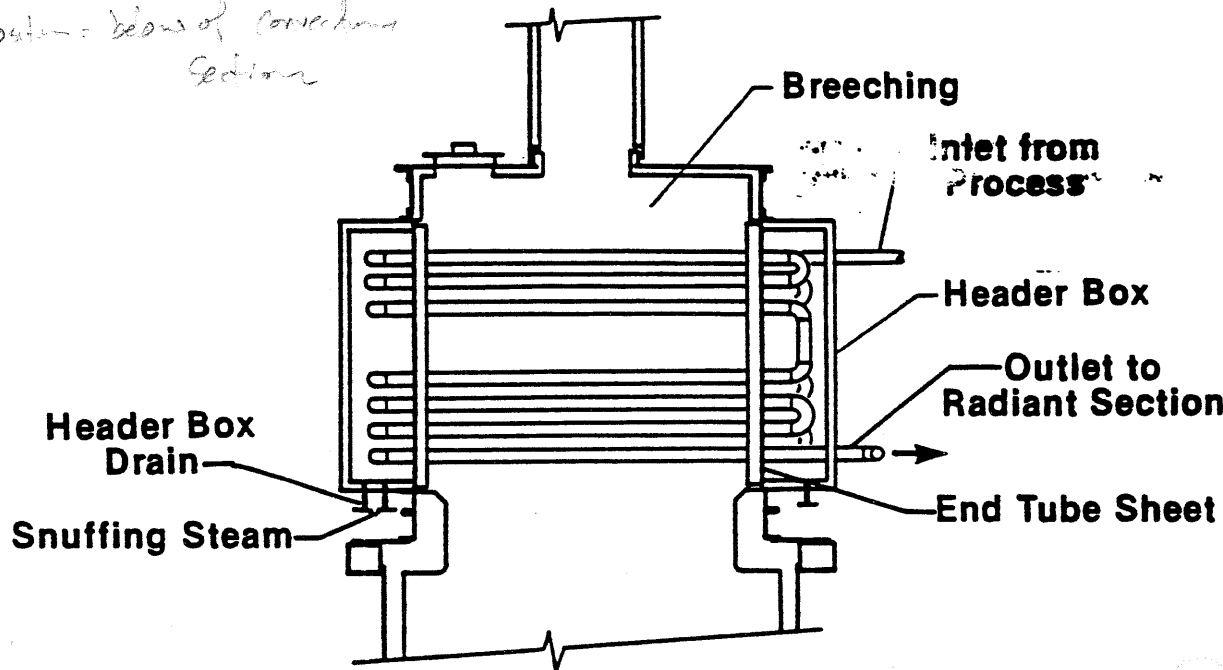


$$L/D = 1.5 \sim 2.75$$

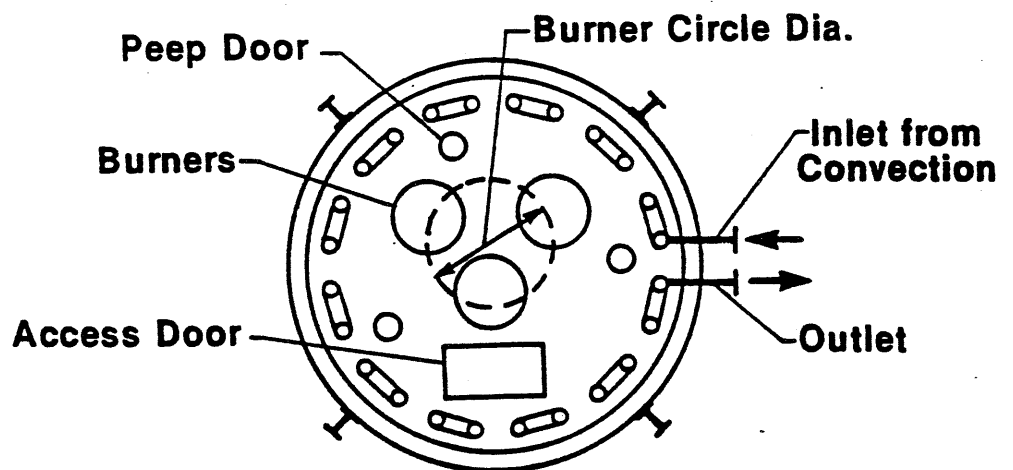
Oxygen Analyser

best position - below of convection section

FIGURE 8B

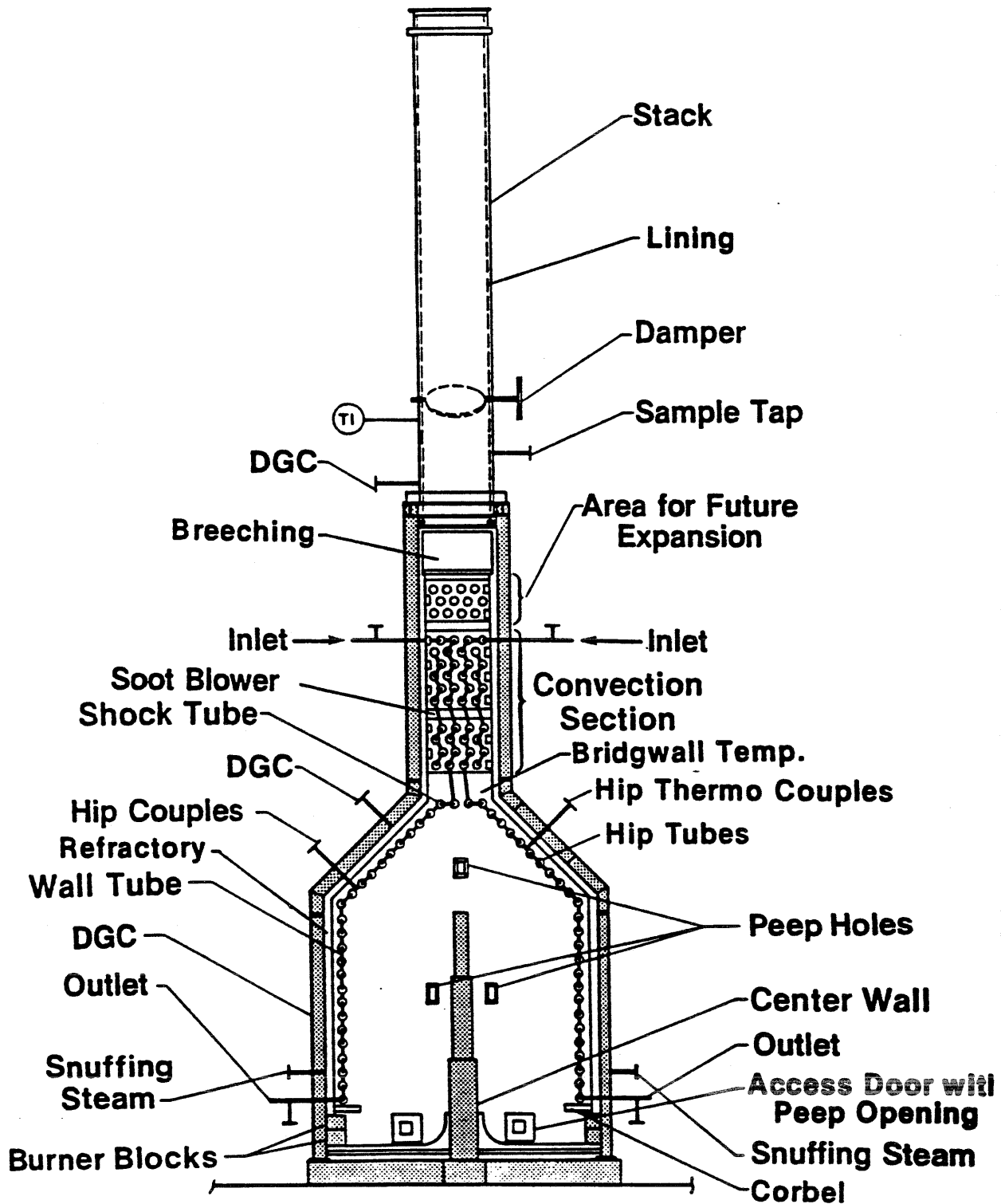


**CONVECTION SECTION
FOR VERTICAL HEATER**



RADIANT SECTION

FIGURE 9A
SIDE FIRED CENTERWALL HEATER



**FIGURE 9B
SIDE VIEW CENTERWALL**

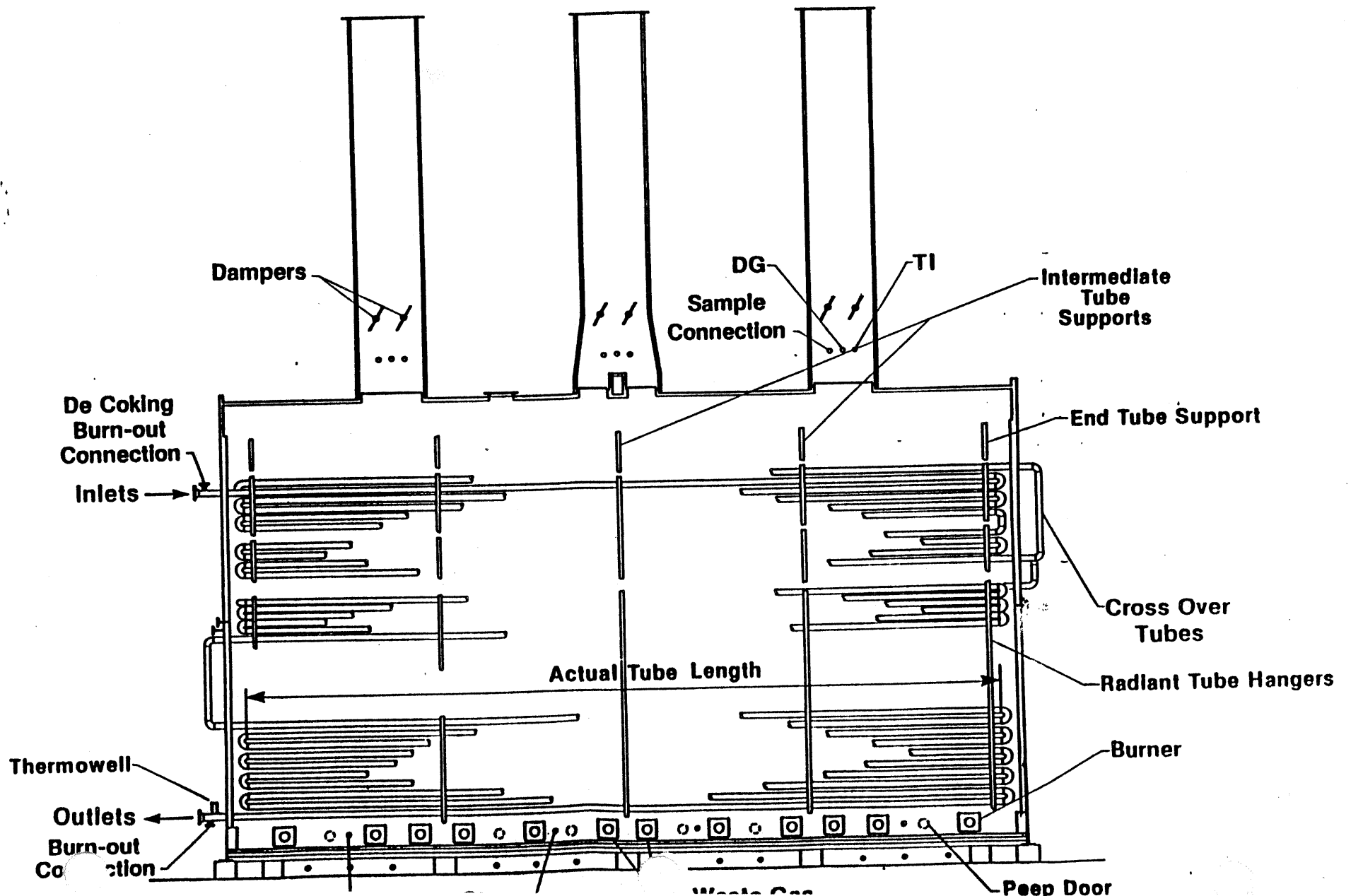


FIGURE 10A

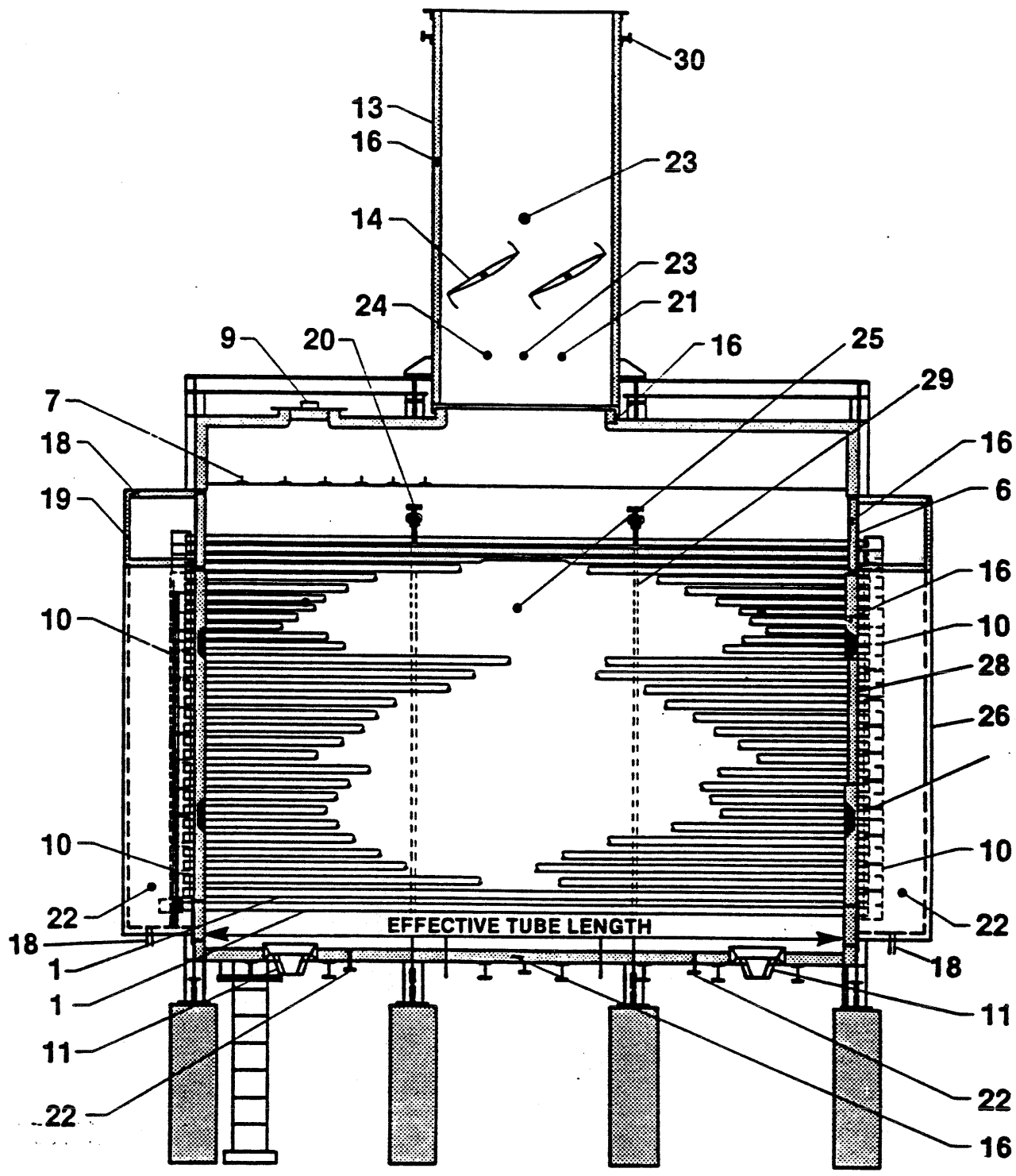


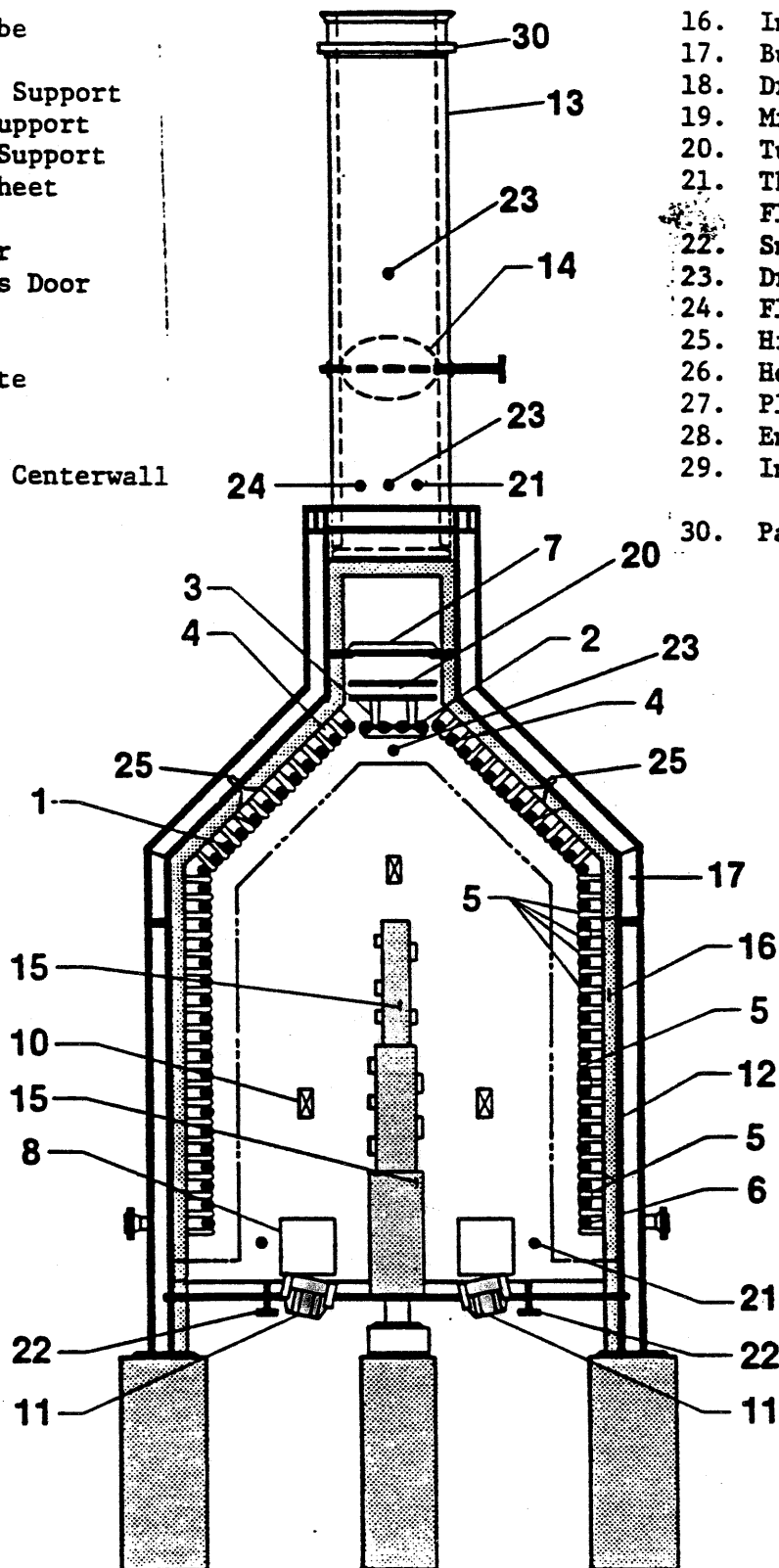
FIGURE 10B ALL RADIANT BOTTOM FIRED CENTERWALL

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
22. Flue Gas Temperature
23. Snuffing Steam Conn.
24. Draft Gage Conn.
25. Flue Gas Sampling Conn.
26. Hip Thermocouple Conn.
27. Header Box
28. Plug Type Fittings
29. End Tube Sheet
30. Intermediate Tube Supports
31. Painters Trolley



**FIGURE 11A
DOUBLE FIRED HEATER**

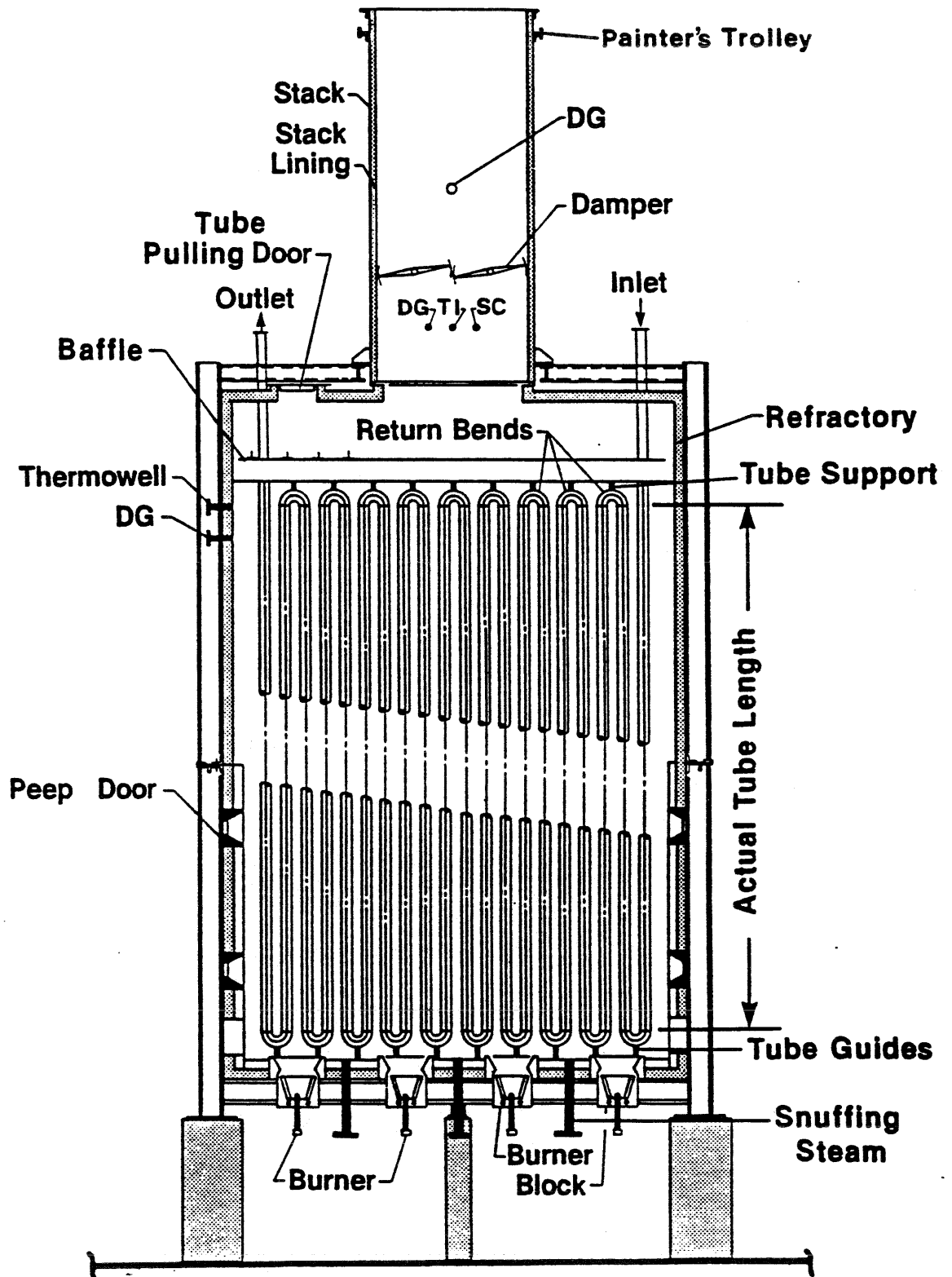


FIGURE 11B

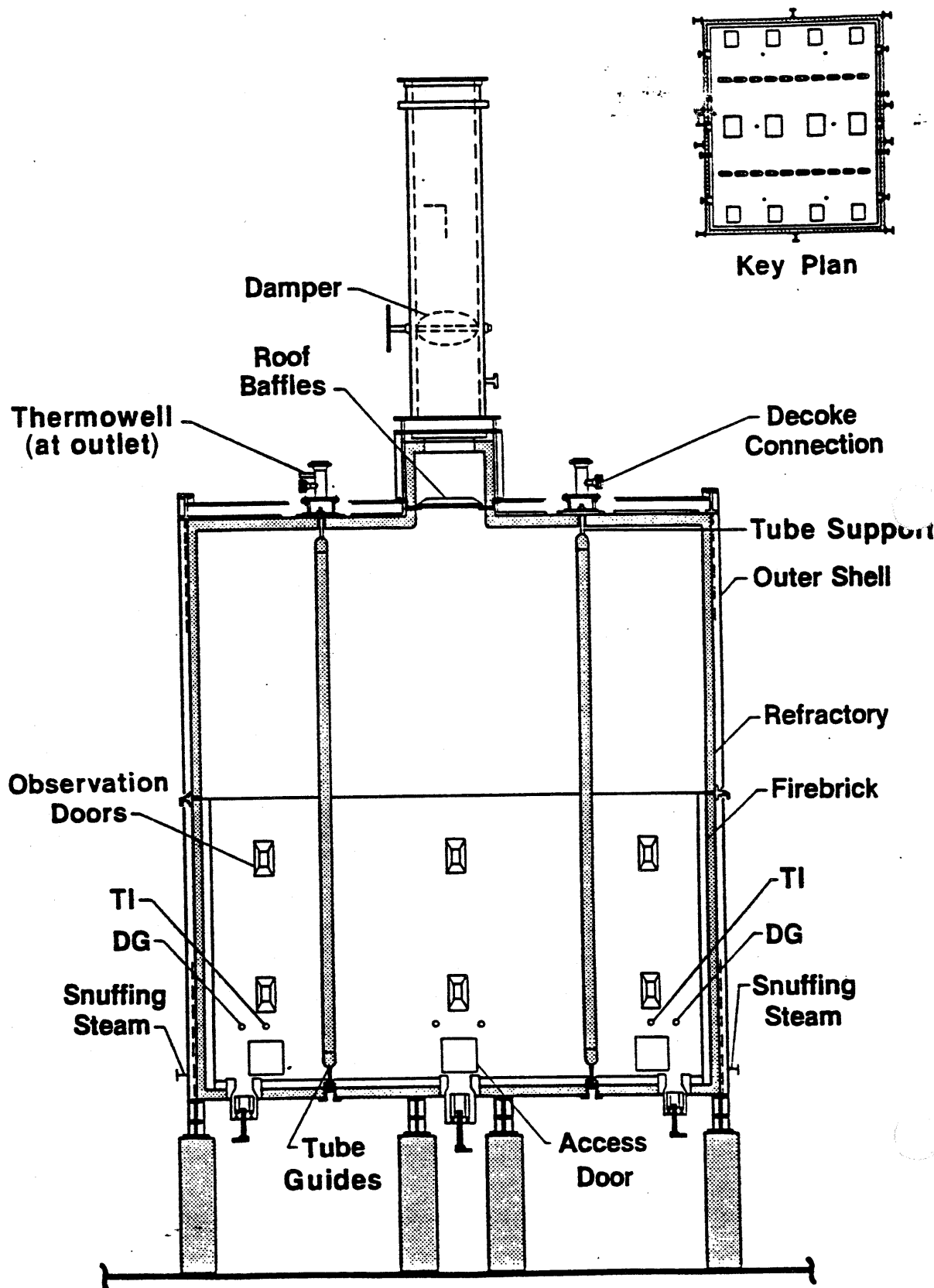


FIGURE 12 HELICAL COIL HEATER

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

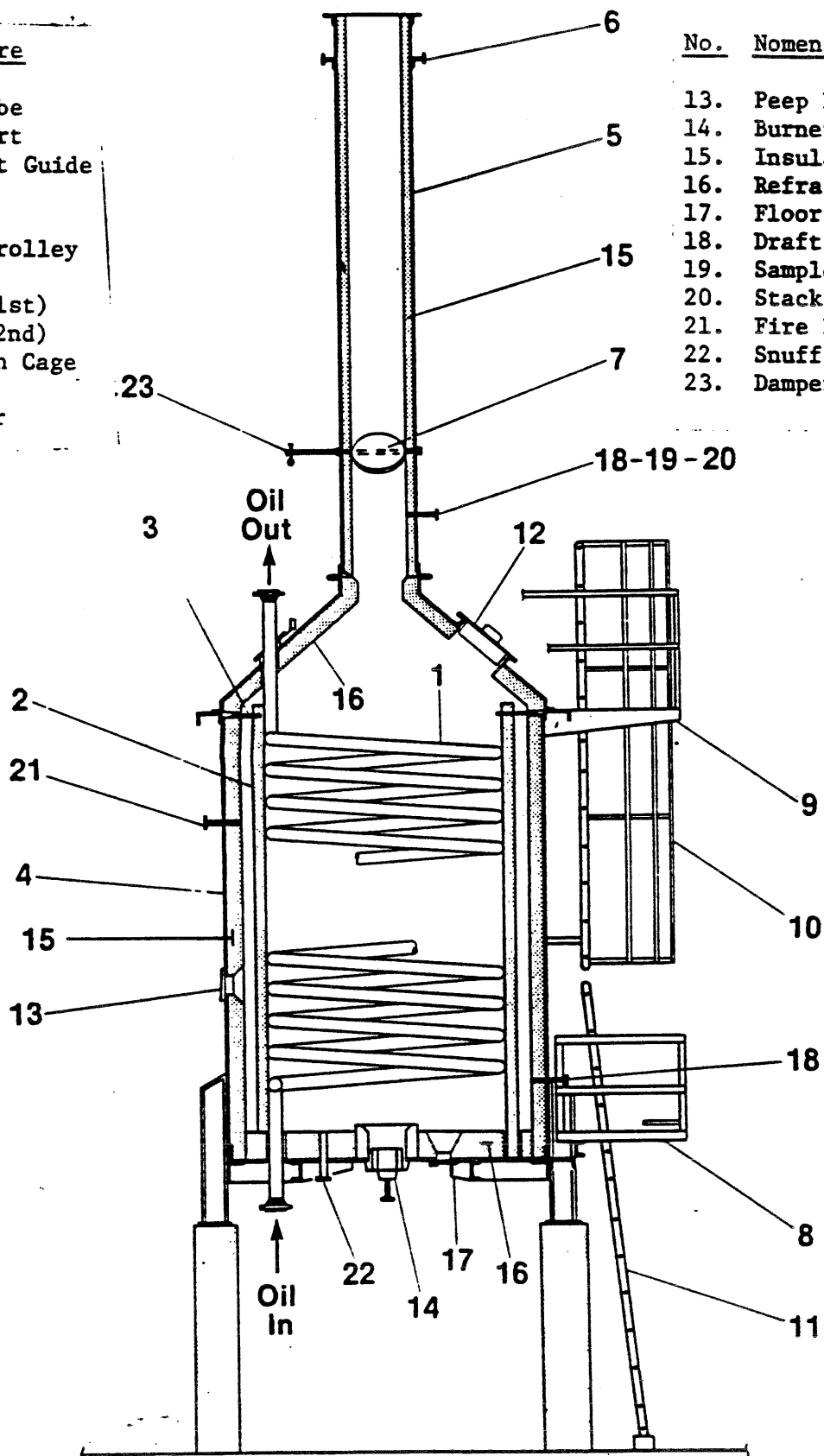


FIGURE 13

UOP
Process Division

NO.	REV.
- 201	-
SHEET	OF
BY	APP'D
DATE	

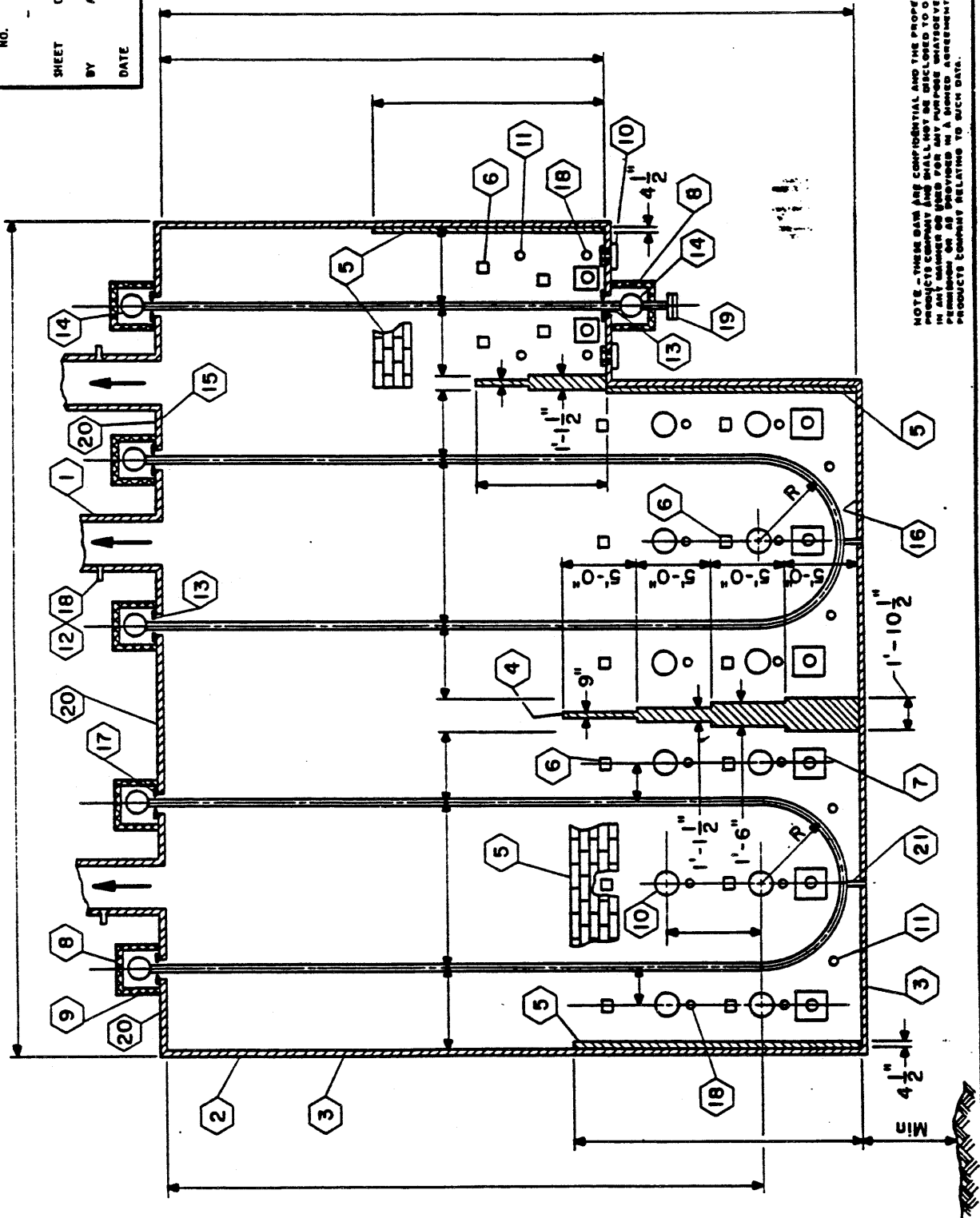
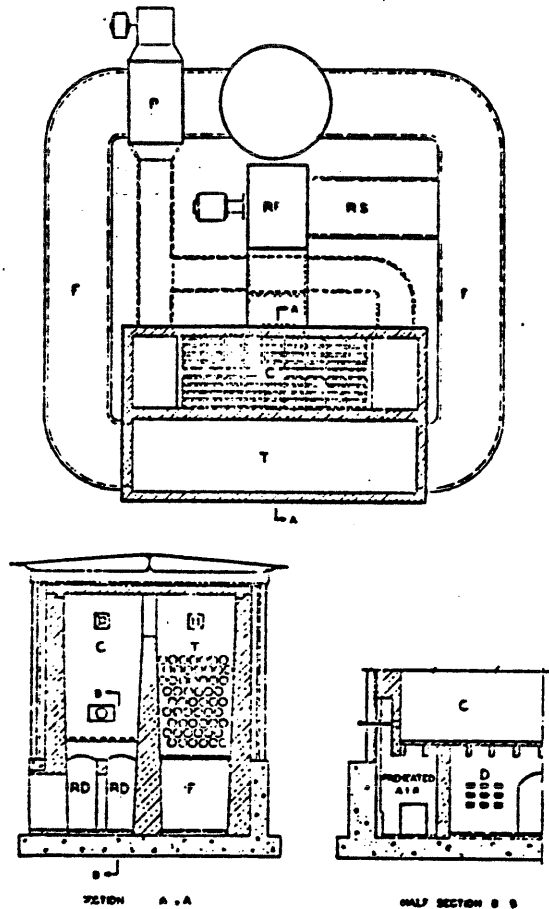
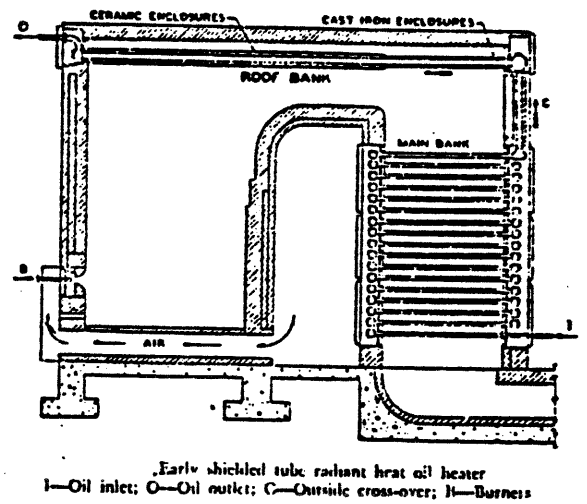


FIGURE 14a

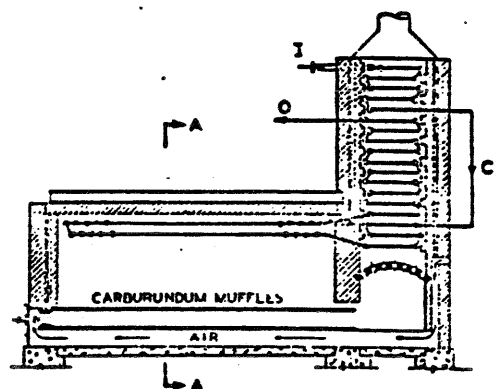
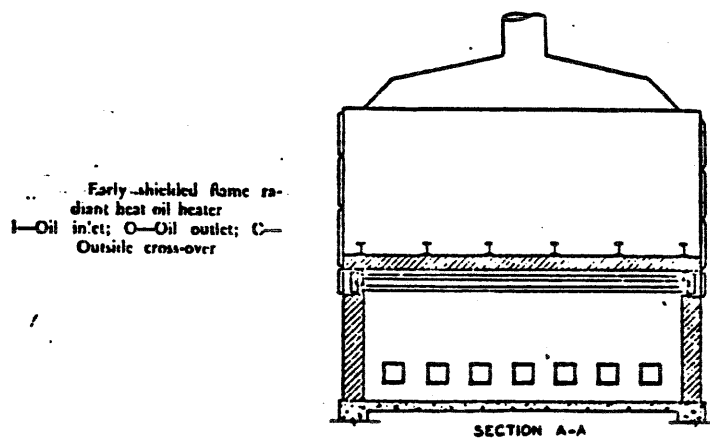


Flue Gas Recirculating Furnace with Air preheater
C—Combustion chamber; T—Tube or heating chamber; F—Flue to stack;
RF—Recirculating fan; RD—Recirculating gas distributing duct; RA—
Recirculating fan suction duct; P—Air preheater

A

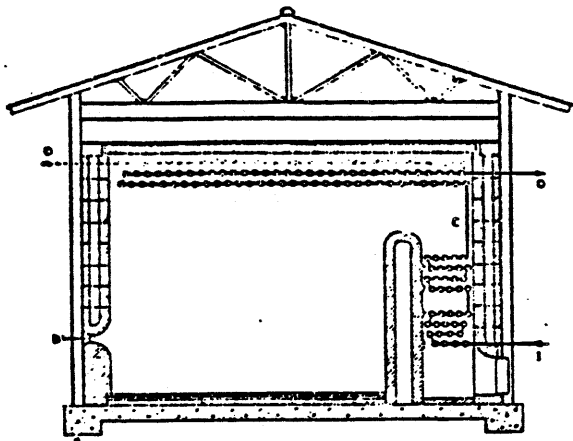


B



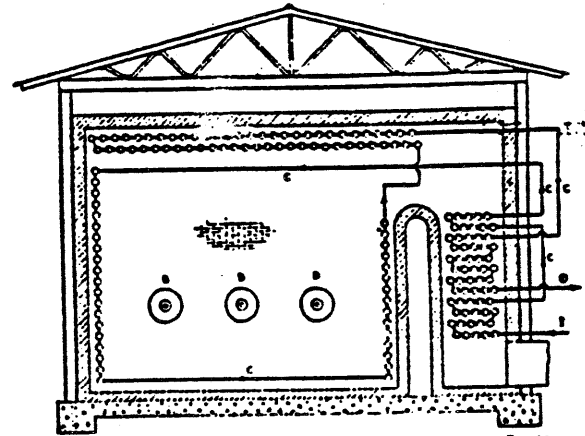
C

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

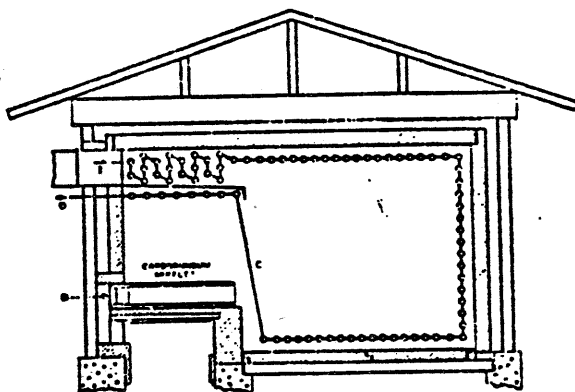
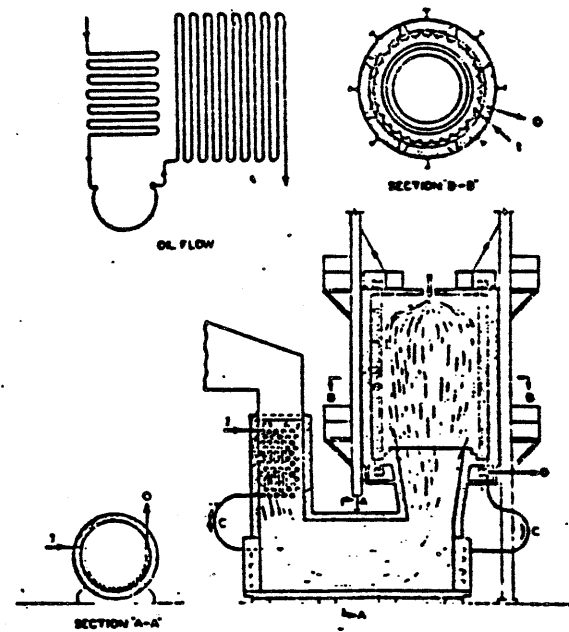


Fig. 19. Partially shielded flame heater
I—Oil inlet; O—Oil outlet; C—Outside cross-over; B—Burners

F



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

G

FIGURE 14c

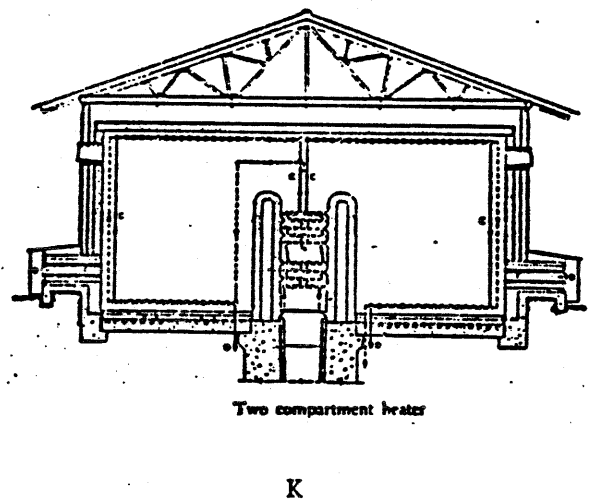
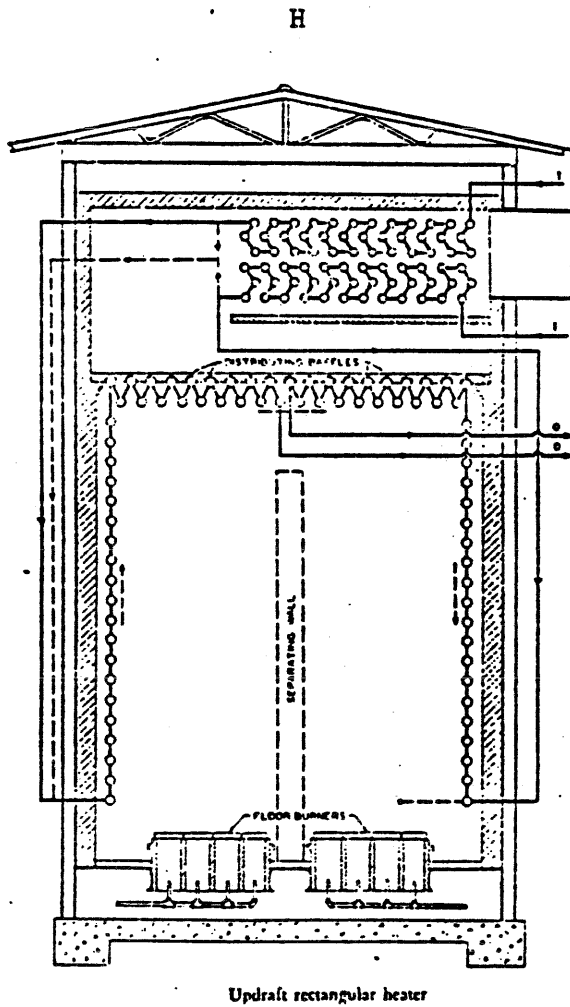
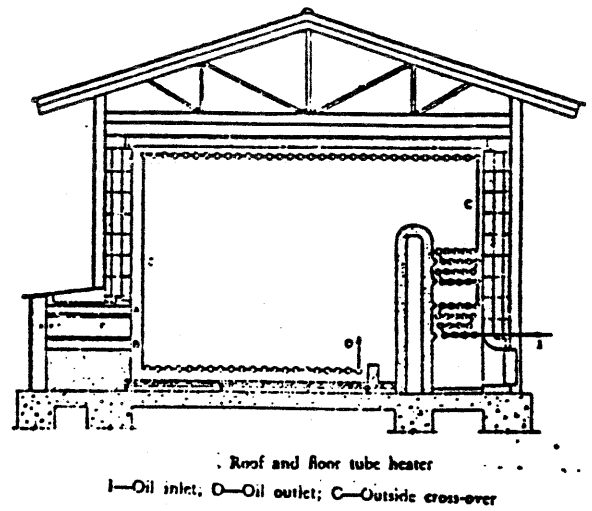
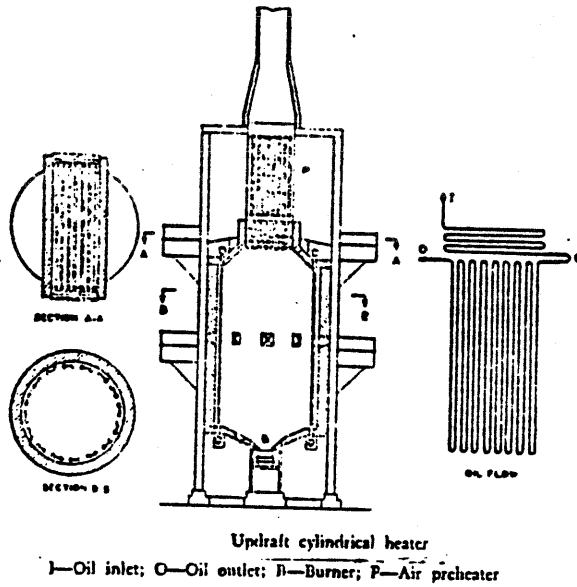
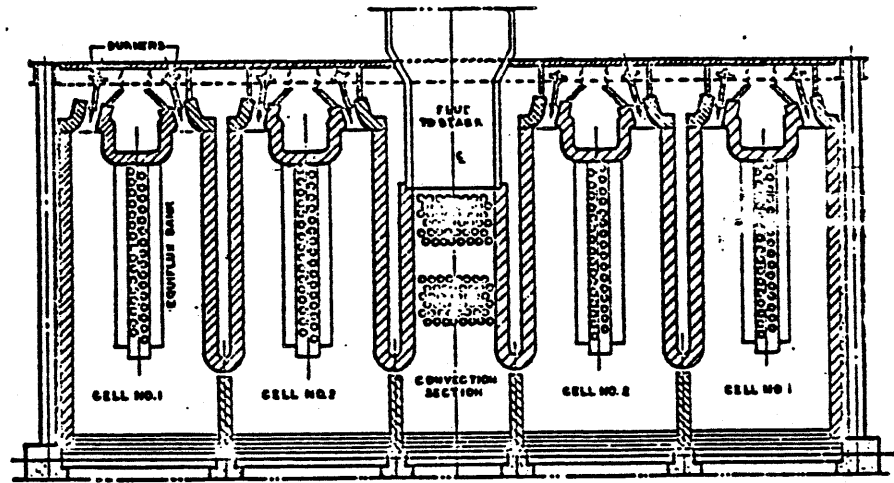
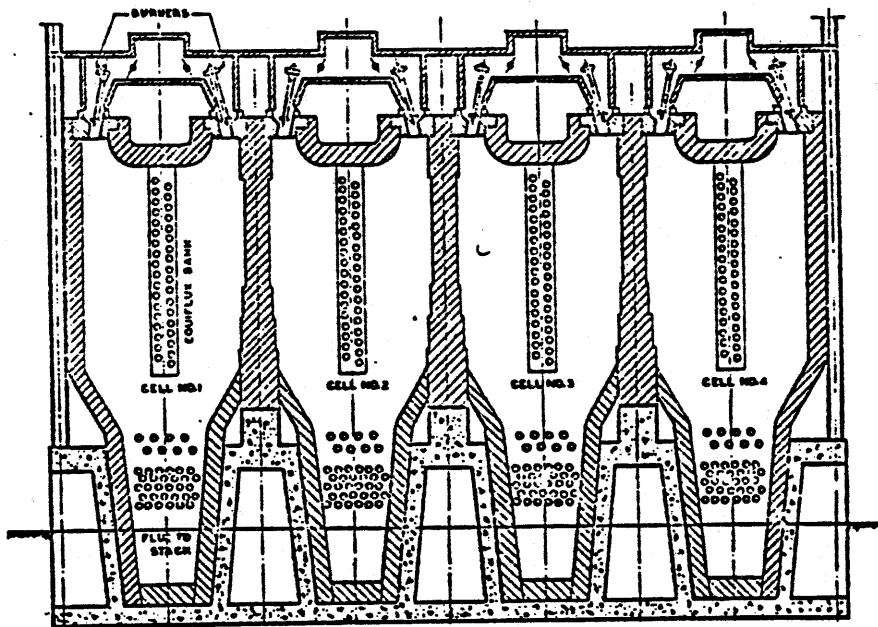


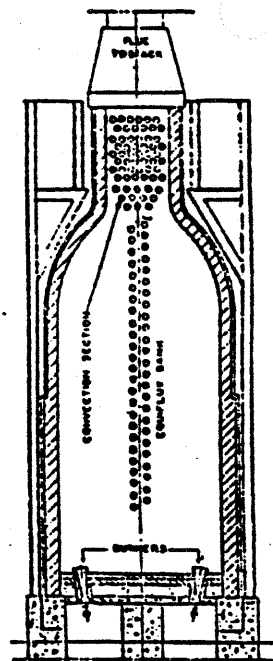
FIGURE 15



A



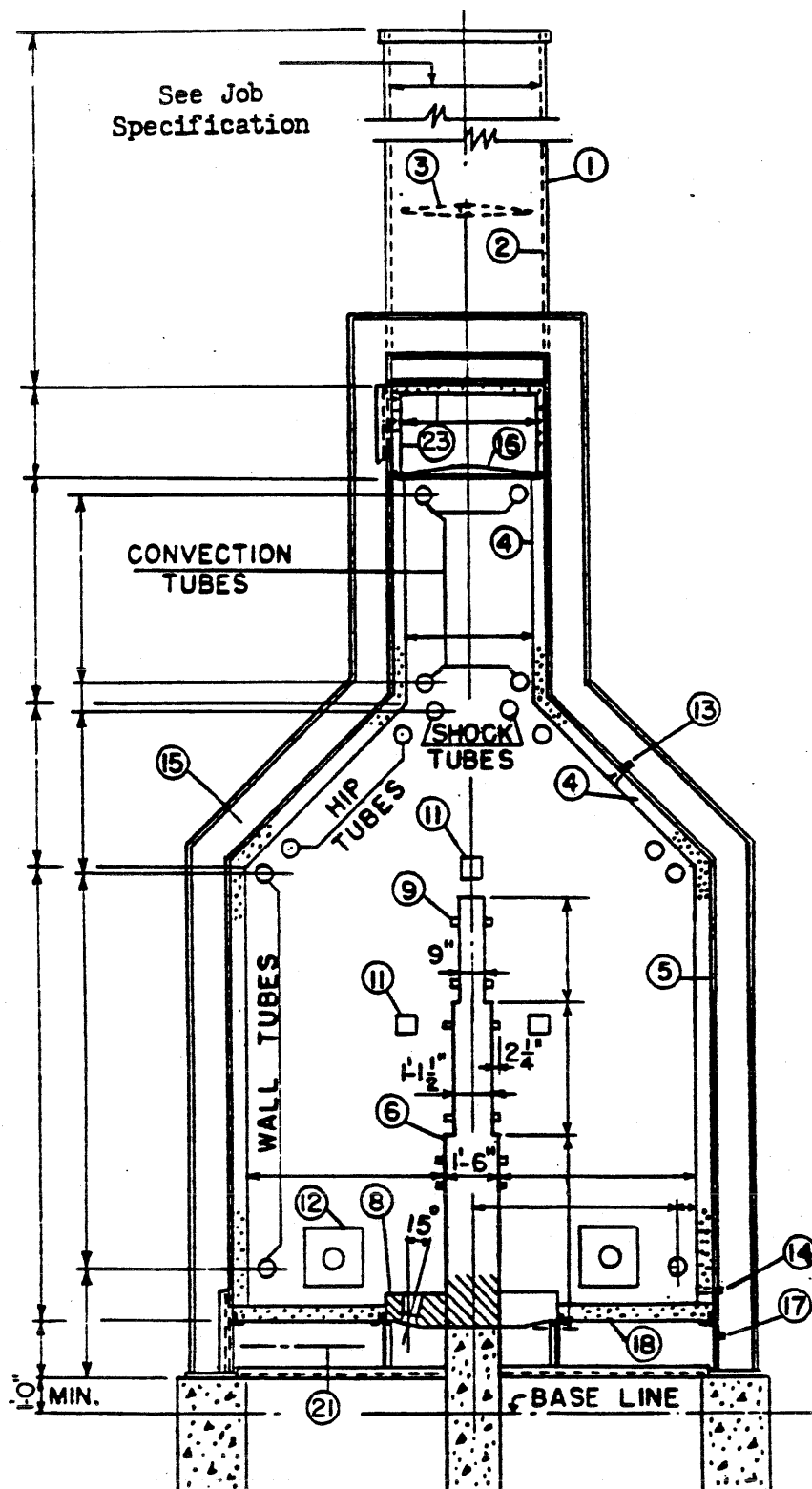
B



C

FIGURE 16

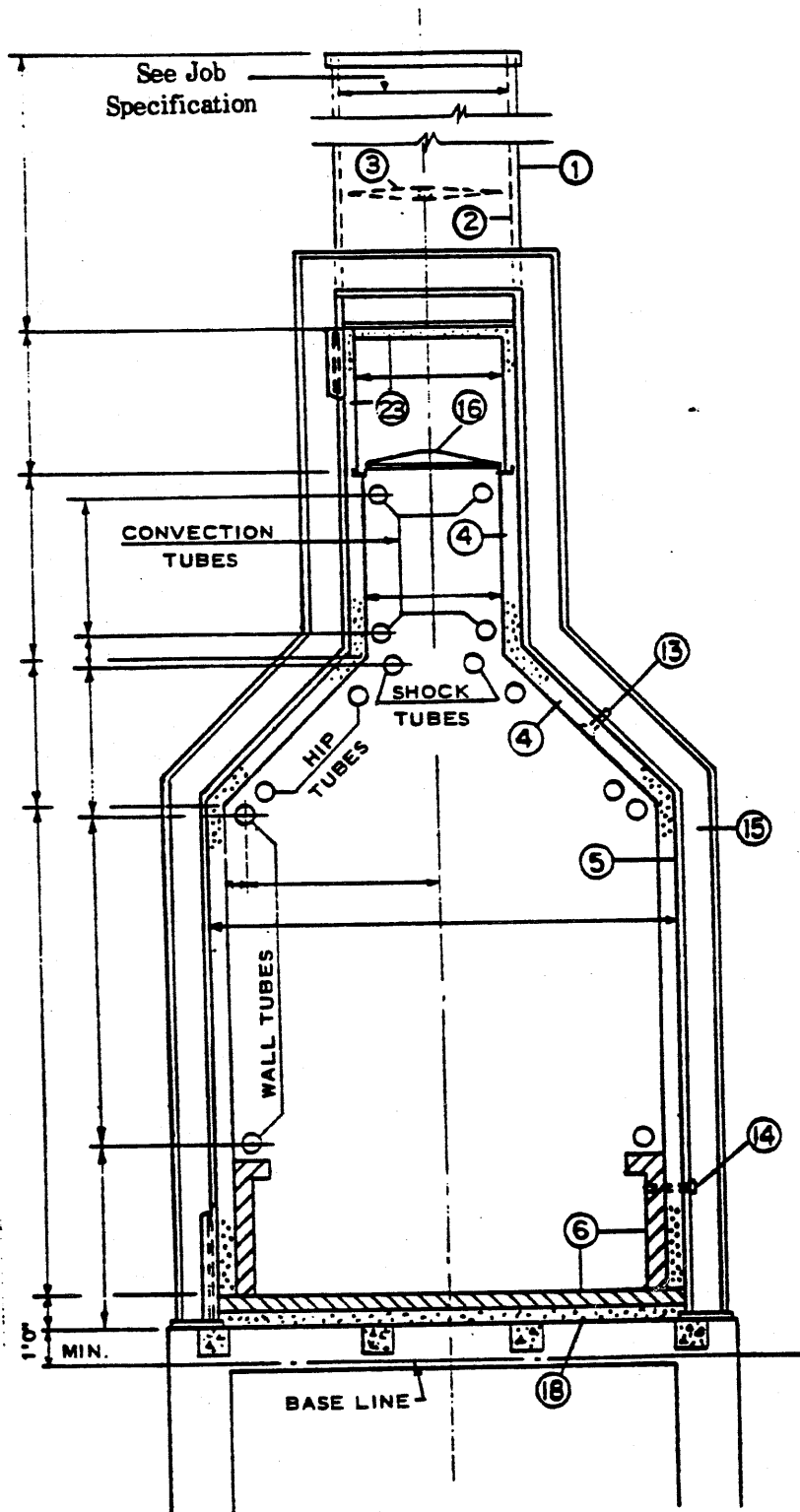
CENTER-WALL UPDRAFT
CROSS SECTION



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 grad
4. 5" insulating concrete UOP Specification No. 294 or No. 296.
5. $\frac{3}{16}$ " steel casing plat
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 end 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.

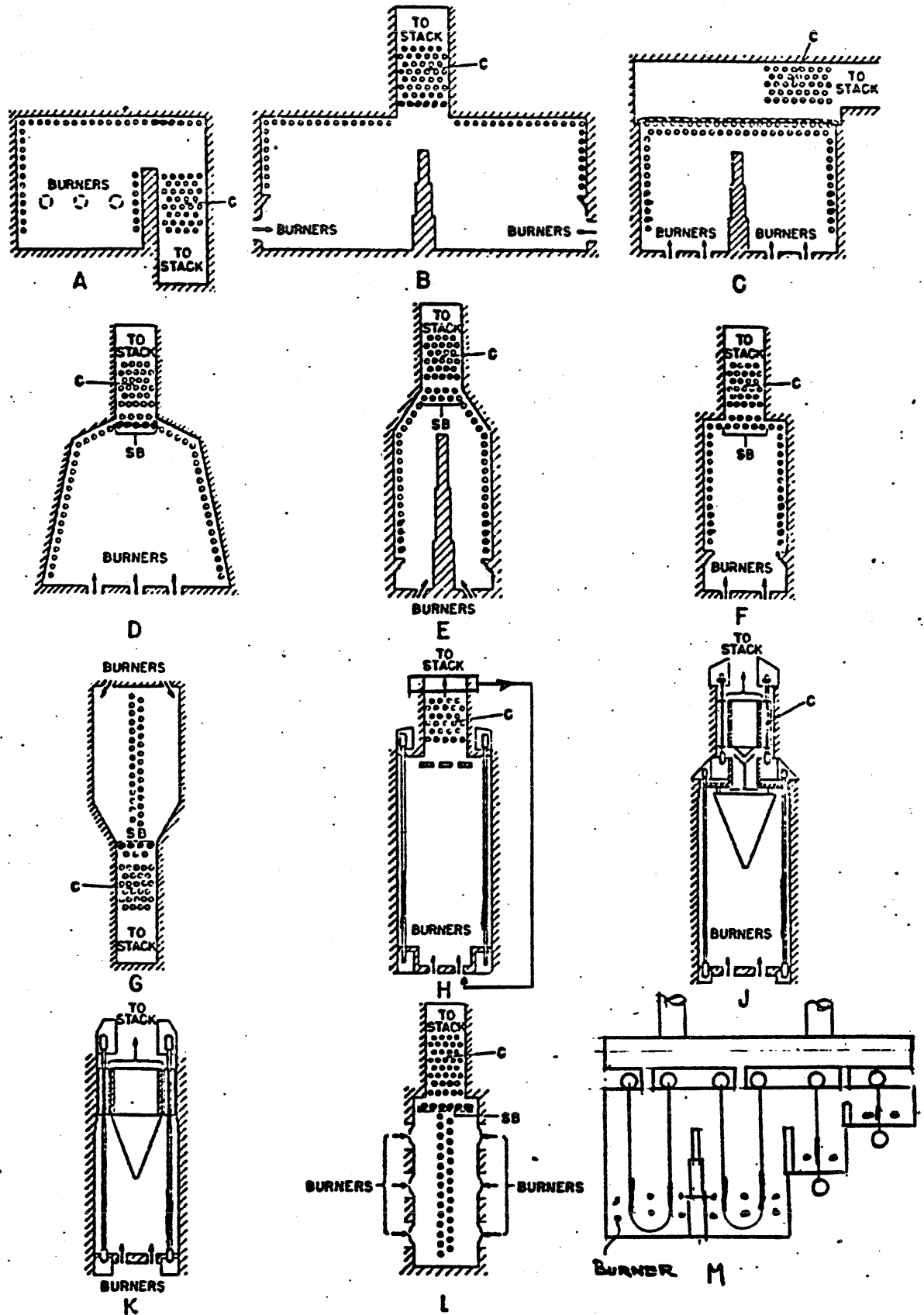
FIGURE 17

END OR BOTTOM FIRED UPDRAFT
CROSS SECTION



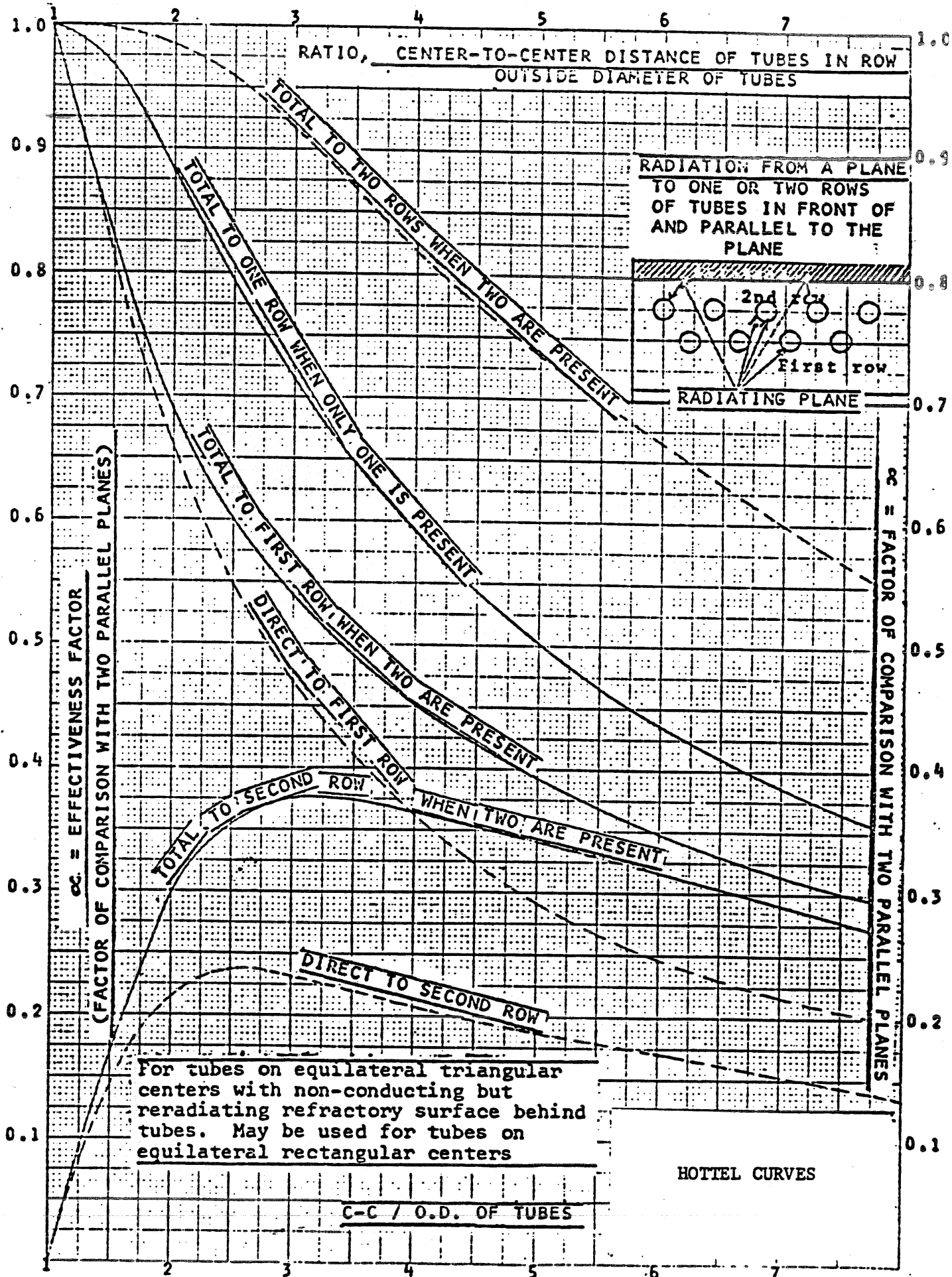
1. Stack $\frac{1}{4}$ " steel plate minimum thickness.
2. $1\frac{1}{2}$ " insulation UOP Specification No. 292.
3. 18-8 stainless steel damper parts with positive manual 2-way control from grade.
4. 5" insulating concrete UOP Specification No. 294, or No. 296.
5. $\frac{3}{16}$ " steel casing plate.
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 connect. (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



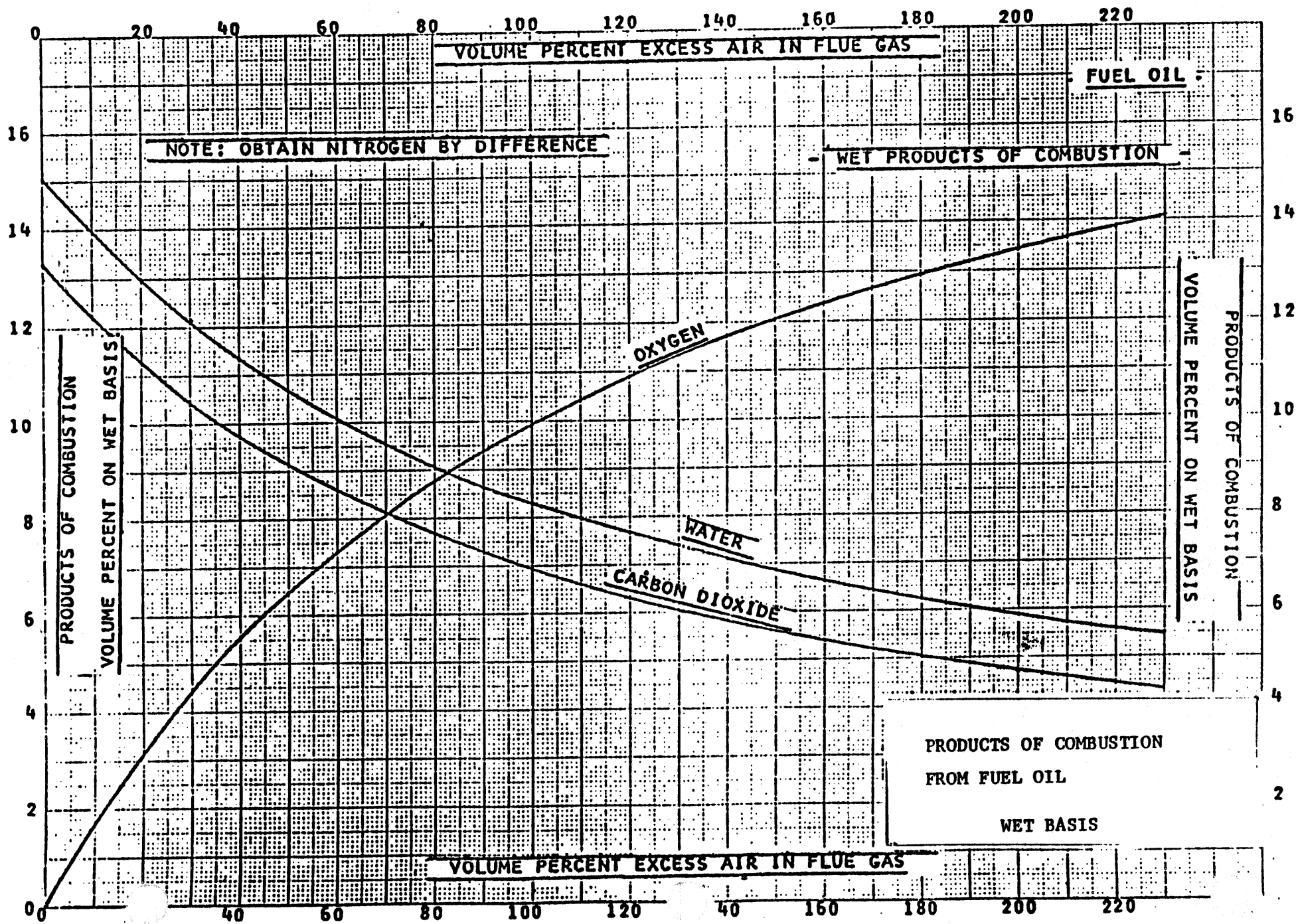


FIGURE 20

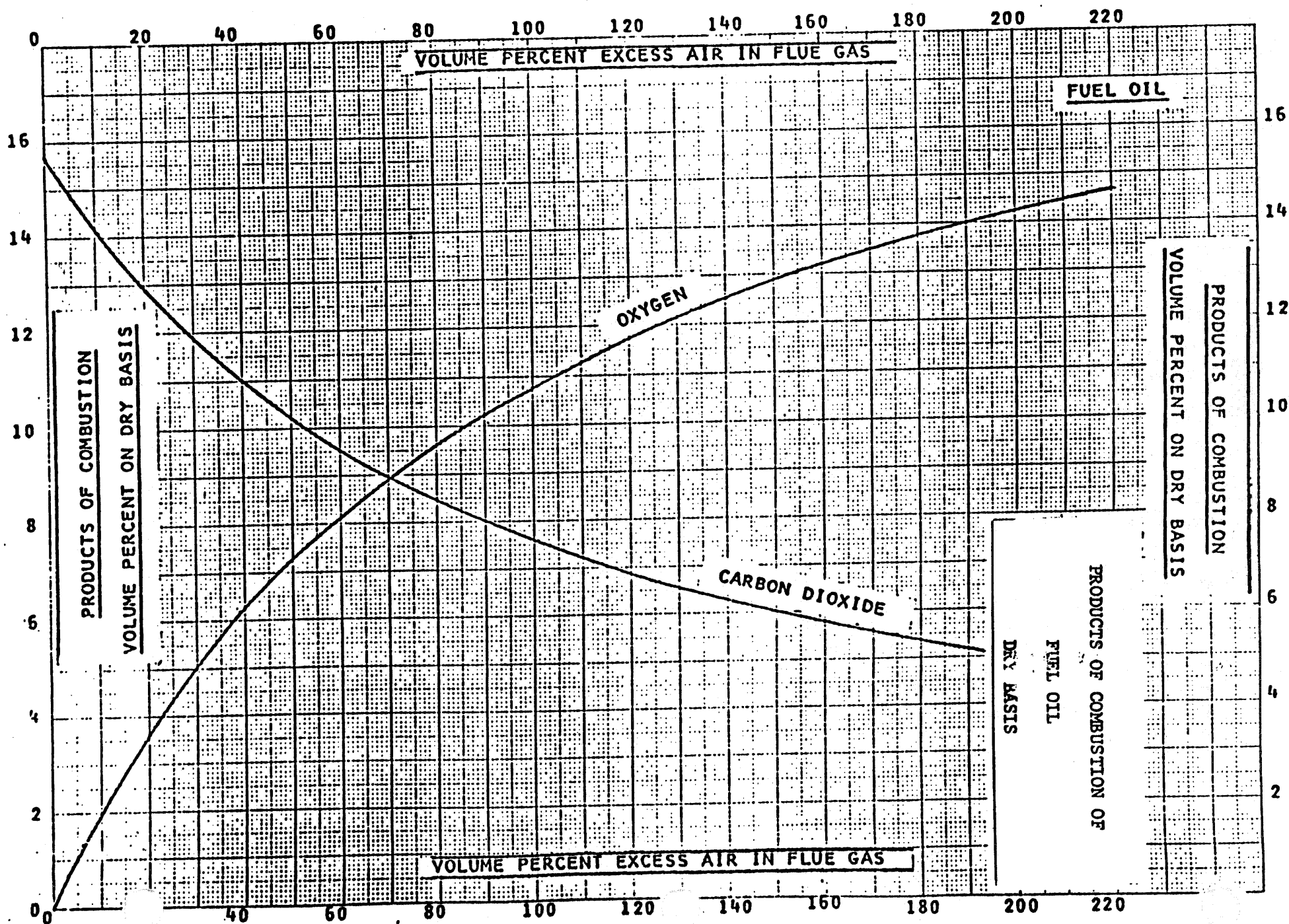


FIGURE 21

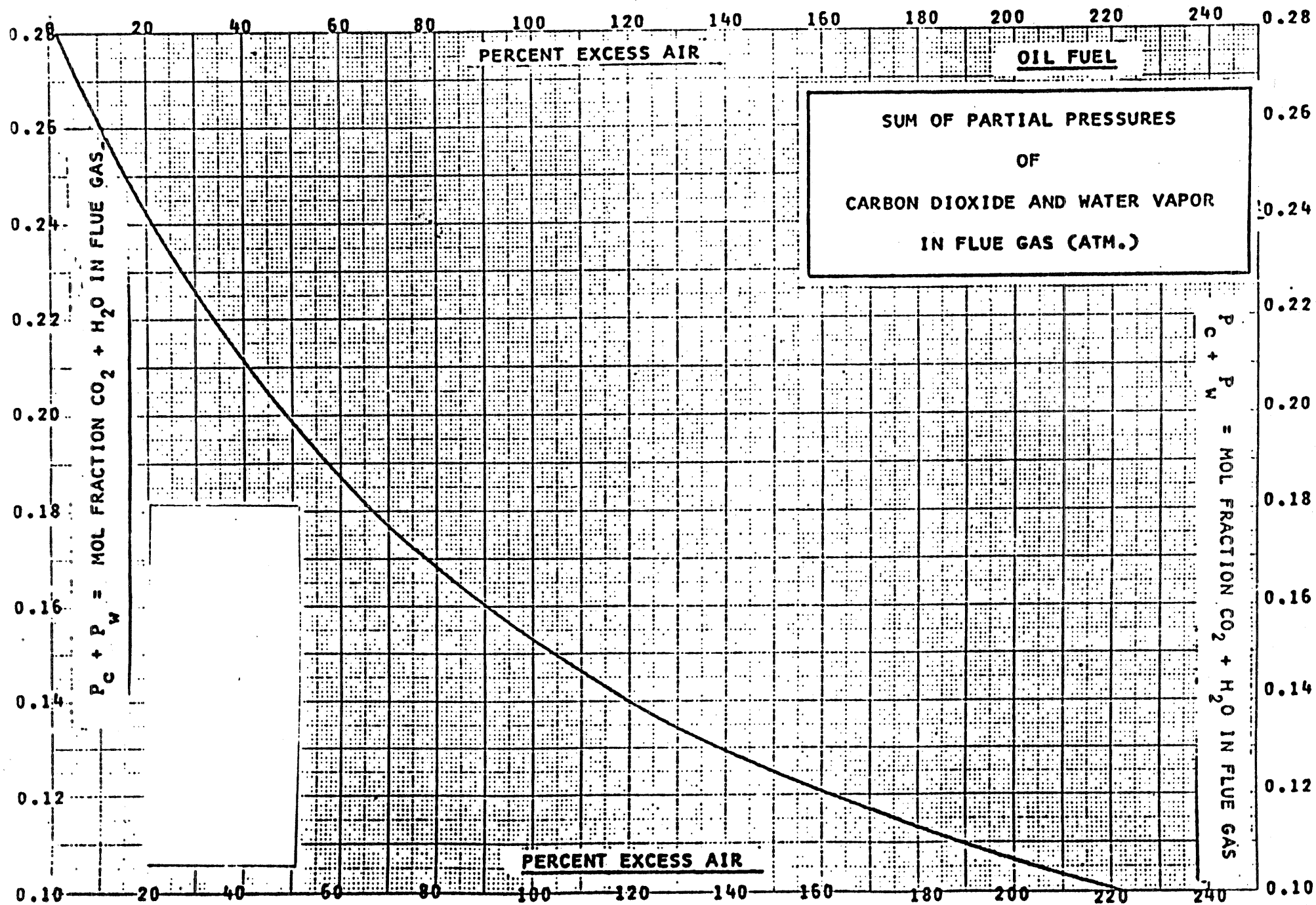


FIGURE 22

The following table gives 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	
			P_{GL}		
			0.01	0.10	1.00
Parallelepipeds			0.61	0.58	0.54
1:1:1 (cube)	edge	0.67			
1:1:4					
radiating to 1 x 4 face	shortest edge	0.90	0.82	0.77	0.73
radiating to 1 x 1 face	" "	0.86	0.78	0.74	0.70
radiating to all faces	" "	0.89	0.81	0.77	0.72
1:2:6					
radiating to 2 x 6 face	shortest edge	1.18	1.07	1.02	0.95
radiating to 1 x 6 face	" "	1.24	1.13	1.07	1.00
radiating to 1 x 2 face	" "	1.18	1.07	1.02	0.95
radiating to all faces	" "	1.20	1.09	1.03	0.97
1:∞:∞ (infinite parallel planes)	distance between planes	2.00	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 walls L may be obtained from Curve No. B-339.

TABLE 1

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)

FIGURE 23a

