Training Services

Fired Heaters

Section Outline

- Introduction
- Process Function
- Description of Heater
- Terminology of Heater
- Heat Transfer Basics

Introduction

- Safety Concerns
- Heaters are Expensive
- Lack of Understanding
- Desire to Increase Capacity

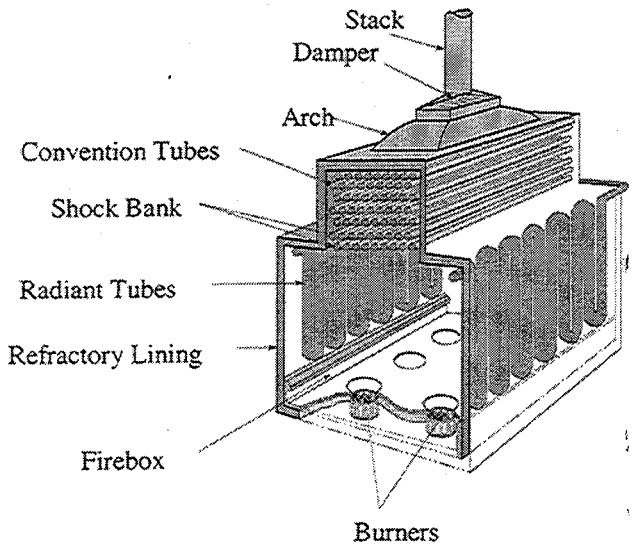
Process Function

- Heaters Upstream of Reactors
- Heaters Used for Fractionation
- Black Oil Cracking Heaters
- Miscellaneous Specialty Heaters

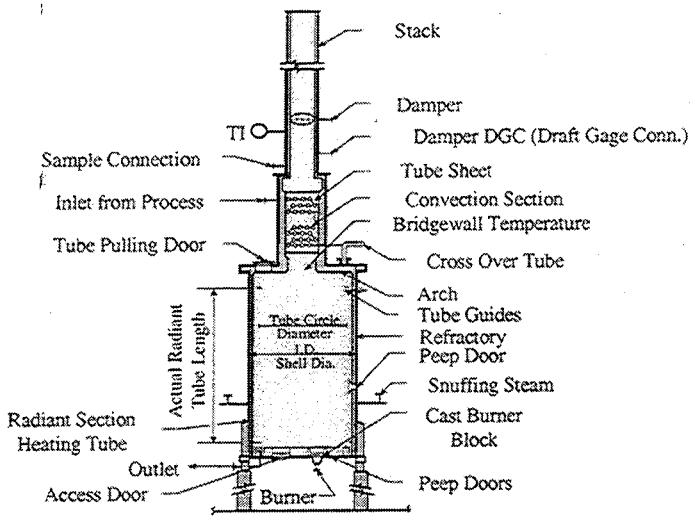
Description of Heater

- Large heat exchanger
- Energy for transfer comes from combustion of fuel
- Radiation heat transfer plays significant role
- Many variations of arrangements
  - Single and double fired
  - Top and bottom supported tubes
  - Vertical and horizontal radiant tubes
  - Vertical Cylindrical is most common

Typical Heater

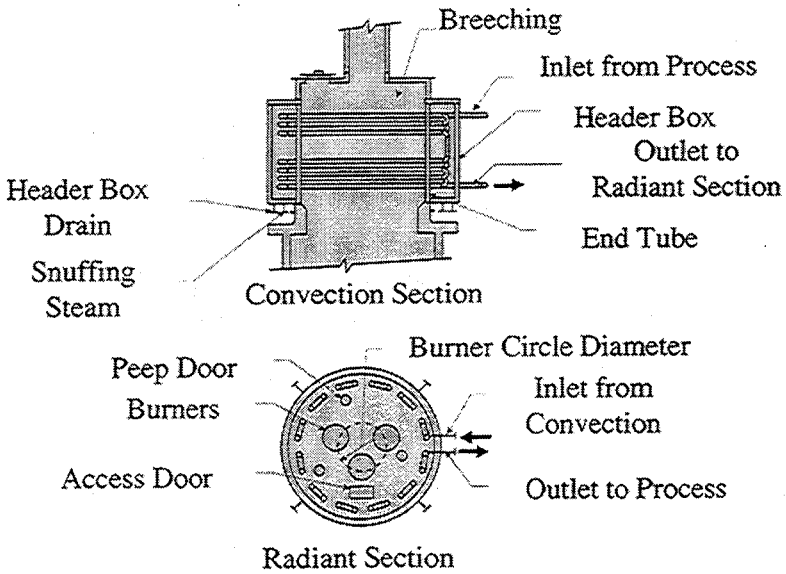


Vertical Cylindrical Heater - Side View



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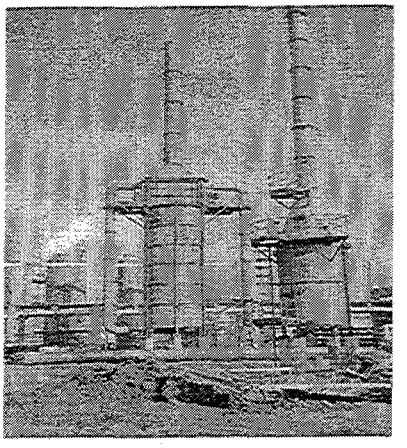
Vertical Cylindrical Heater



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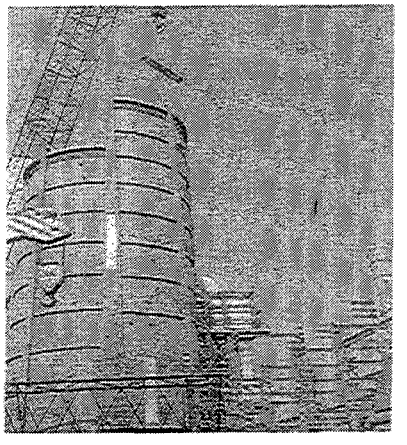
Vertical Cylindrical Heater

- Most Common Heater
- Least Cost
- Smallest Plot
- Easy to Construct
- Good for small duties
- Maximum process duty is 170 MM Btu/h



Vertical Cylindrical Heater

- Radiant Section Panels Modules Joined
- Coil lifted in place
- Convection section placed on top



Description of Heaters

- Radiant Section
- Convection Section
- Refractory
- Burners
- Stack

Heat Transfer Basics

- Radiation
- Convection
- Conduction

Heat Transfer Basics - Radiation

- Transfer of heat by direct waves
- Direct from heat source or reradiated from reflecting surface
- Driving force is difference in absolute temperature to 4th power

$$Q \approx T^4_{hot} - T^4_{cold}$$

Heat Transfer Basics - Convection

- Heat transfer by flow of liquid or vapor
- Contact required
- Driving force is temperature difference

$$Q = h\,A\,(T_{hot} - T_{cold})$$

- Depends on fluid properties and geometry

Heat Transfer Basics - Conduction

- Heat transfer within a substance
- Depends on thermal conductivity of material
- Driving force is temperature difference

$$Q = \frac{k}{L} A (T_{hot} - T_{cold})$$

API 560 - Fired Heaters for General Refinery Services

SECTION 1.4 - Definition of Terms

1.4.1 *Air heater or air preheater* is a heat transfer apparatus through which combustion air is passed and heated by a medium of higher temperature, such as the products of combustion, steam or other fluid.

1.4.1.1 *Direct air preheater is* an exchanger which transfers heat directly between the flue gas and the combustion A regenerative air preheater uses heated rotating elements and a recuperative design uses stationary tubes, plates, cast iron elements to separate the two heating media.

1.4.1.2 *Indirect air preheater is* a fluid-to-air heat transfer device. The heat transfer can be accomplished by using a heat transfer fluid, process stream, or utility stream which has been heated by the flue gas or other means. A heat pipe air preheater uses a vaporizing/condensing fluid to transfer heat between the flue gas and air.

1.4.2 *Arch* is a flat or sloped portion of the heater radiant section opposite the floor.

1.4.3 *Atomizer is* a device used to reduce a liquid fuel oil to fine mist. Atomization media are steam, air, or mechanical.

1.4.4 *Anchor or tieback* is a metallic or refractory device that holds the refractory or insulation in place.

1.4.5 *Backup layer* is any refractory layer behind the hot face layer.

1.4.6 *Balanced draft heater* uses a fan to supply combustion air and to remove flue gases.

1.4.7 *Breeching* is the heater section where flue gases are collected after the last convection coil for transmission to the stack or the outlet ductwork.

1.4.8 *Bridgewall, division, or gravity wall* separates two adjacent heater zones.

1.4.9 *Bridgewall temperature* is the flue gas temperature leaving the radiant section.

1.4.10 *Burner* introduces fuel and air into a heater at the desired velocities, turbulence, and concentration to establish and maintain proper ignition and combustion. Burners are classified by the types of fuel fired, such as: oil, gas, or combination of gas and oil. A secondary consideration in classifying burners is the means by which combustion air is mixed with the fuel.

1.4.11 *Casing is* the metal plate used to enclose the fired heater.

1.4.12 *Castable* is an insulating concrete poured or gunned in place to form a rigid refractory shape or structure.

1.4.13 *Ceramic fiber* is a fibrous refractory insulation composed primarily of silica and alumina. Applicable forms include blanket, board, module, rigidized blanket, and vacuum-formed shapes.

1.4.14 *Coil pressure drop* is the difference between the coil inlet pressure and the coil outlet pressure between terminals, excluding the effect of static head.

1.4.15 *Convection section* is the portion of the heater in which the heat is transferred to the tubes primarily by

1.4.16 *Corbel is* a projection from the refractory surface generally used to prevent flue gas bypassing the convection section tubes.

1.4.17 *Corrosion allowance is* the additional material thickness added to allow for material loss during the design life of the component. It is the corrosion rate times tube design life, expressed in inches (millimeters).

1.4.18 *Corrosion rate is* the reduction in the material thickness due to the chemical attack from the process fluid or flue gas or both, expressed in mils per year (millimeters per year).

1.4.19 *Crossover is* the interconnecting piping between any two heater coil sections.

1.4.20 *Damper is* a device for introducing a variable resistance for regulating volumetric flow of gas or air.

1.4.20.1 *Butterfly damper is* a single-blade damper pivoted about its center.

1.4.20.2 *Louver damper* is a damper consisting of several blades each pivoted about its center and linked together for I simultaneous operating system.

1.4.21 *Draft* is the negative pressure (vacuum) of the flue gas measured at any point in the heater, expressed in inches of water column (pascals).

1.4.22 *Draft Fan is* the pressure drop through duct conduits or across tubes and equipment in air and flue gas systems.

1.4.23 *Duct* is a conduit for air or flue gas flow.

1.4.24 *Efficiency, fuel* refers to the heat absorbed divided by the net heat of combustion of the fuel as heat input, expressed as a percentage.

1.4.25 *Efficiency, thermal* refers to the total heat absorbed divided by the total heat input. expressed as a percentage.

1.4.26 *Erosion* is a reduction in material thickness due to mechanical attack from a fluid, expressed in inches (millimeters).

1.4.27 *Excess air is* the amount of air above the stoichiometric requirement for complete combustion. expressed as a percentage.

1.4.28 *Extended surface* refers to the heat transfer surface in the form of fins or studs attached to the heat absorbing surface.

1.4.29 *Extension ratio* is the ratio of total outside exposed surface to the outside surface of the bare tube.

1.4.30 *Flue gas* is the gaseous product of combustion expressed as a percentage.

1.4.31 *Forced draft heater* supplies combustion air by a fan or other mechanical means.

1.4.32 *Fouling allowance is* a factor to allow for a layer of residue that increases pressure drop, usually a build up of coke or scale on the inner surface of a coil, expressed as inches (millimeters). This value shall be used in calculating the fouled pressure drop.

1.4.33 *Fouling resistance is* a factor used to calculate the overall heat transfer coefficient. The inside fouling resistance shall be used to calculate the maximum metal temperature for design. The external fouling resistance is used to compensate the loss of performance due to deposits on the external surface of the tubes or extended surface.

1.4.34 *Guillotine blind is* a single blade device that is used to isolate equipment or heaters.

1.4.35 *Header or return bend is* the common term for a 180-degree cast or wrought fitting that connects two or more tubes.

1.4.36 *Header box is* the internally insulated structural compartment, separated from the flue gas stream, which is used to enclose a number of headers or manifolds. Access is afforded by means of hinged doors or removable panels.

1.4.37 *Heat absorption is* the total heat absorbed by the coils excluding any combustion air preheat, expressed in British thermal units per hour (megawatts)

1.4.38 *Heat flux density, average is* the heat absorbed divided by the exposed heating surface of the coil section. Average flux density for an "tended surface tube shall be indicated on a bare surface basis with extension ratio noted, expressed in British thermal units per hour per square foot (kilowatts per square meter).

1.4.39 *Heat flux density, maximum* is the maximum local heat transfer rate in the coil section, expressed in British thermal units per hour per square foot (kilowatts per square meter).

1.4.40 *Heat release* is the total heat liberated from the specified fuel, using the lower heating value of the fuel, expressed in British thermal units per hour (megawatts).

1.4.41 *Heating value, higher (HHV),* is the total heat obtained from the combustion of a specified fuel at 60T, expressed in British thermal units per pound or per cubic foot (kilojoules per kilogram or per cubic meter),

1.4.42 *Heating value, lower (LHV),* is the higher heating value minus the latent heat of vaporization of the water formed by combustion of hydrogen in the fuel, also called the net heating value, expressed in British thermal units per pound or per cubic foot (kilojoules per kilogram or per cubic meter.

1.4.43 *Hot face layer* is the refractory layer exposed to the highest temperatures in a multilayer or multicomponent lining.

1.4.44 *Hot face temperature* is the temperature of the refractory surface in contact with the flue gas or heated combustion air. The hot face temperature is used to determine refractory or insulation thickness and heat transmitted. The design temperature is used to specify the service temperature limit of the refractory materials.

1.4.45 *Induced draft heater* uses a fan to remove flue gases and maintain a negative pressure in the heater to induce combustion air without a forced draft fan.

1.4.46 *Manifold* is a chamber for the collection and distribution of fluid to or from multiple parallel flow paths.

1.4.47 *Metal fiber* reinforcement is stainless steel needles added to castable for improved toughness and durability.

1.4.48 *Monolithic lining* is a single-component lining system.

1.4.49 *Mortar* is a refractory material preparation used for laying and bonding refractory bricks, for example, castable and ceramic fiber.

1.4.50 *Multicomponent lining* is a refractory system consisting of two or more layers of different refractory types.

1.4.51 *Multilayer lining* is a refractory system consisting of two or more layers of the same refractory.

1.4.52 *Natural draft heater* is a unit in which a stack effect inducts the combustion air and removes the flue gases.

1.4.53 *Normal heat release* is the design heat absorption of the heater divided by the calculated fuel efficiency expressed in British thermal units per hour (megawatt).

1.4.54 *Pass or stream* is a flow circuit consisting of one or more tubes in series.

1.4.55 *Pilot* is a. smaller burner that provides ignition energy to light the main burner.

1.4.56 *Plenum or windbox* is a chamber surrounding the burners that is used to distribute air to the burners or reduce combustion noise.

1.4.57 *Plug header* is a cast return bend provided with one or more openings for the purpose of inspection, mechanical tube cleaning, or draining.

1.4.58 *Primary air* is that portion of the total combustion air that first mixes with the fuel.

1.4.59 *Protective coating* is a corrosion-resistant material applied to a metal surface; for example, on casing plates behind porous refractory materials to protect against sulfur in the flue gases.

1.4.60 *Radiant section* is that portion of the heater in which bee is transferred to the tubes, primarily by radiation.

1.4.61 *Radiation or setting loss* is the heat lost to the surroundings from the casing of the heater and the ducts and auxiliary equipment (when heat recovery systems are used) expressed as percent of heat release.

1.4.62 *Secondary air* is the air supplied to the fuel to supplement primary air.

1.4.63 *Setting* is the heater casing, brickwork, refractory, and insulation, including the tiebacks or anchors.

1.4.64 *Shield* section contains those tubes that shield the remaining convection section tubes from direct radiation.

1.4.65 *Sootblower* is a mechanical device for discharging steam or air to clean heat-absorbing surfaces.

1.4.66 *Stack* is a vertical conduit used to discharge flue gas to the atmosphere.

1.4.67 *Strakes or spoilers* are metal stack attachments that prevent wind-induced vibration.

1.4.68 *Target wall* is a vertical refractory firebrick wall which is exposed to direct flame impingement on one or both sides.

1.4.69 *Temperature allowance* is the number of degrees Fahrenheit (Celsius) to be added to the process fluid temperature to account for flow maldistribution and operating unknowns. The temperature allowance is added to the calculated maximum tube metal temperature or the equivalent tube metal temperature to obtain the design metal temperature.

1.4.70 *Terminal* is a flanged or welded projection from the coil providing for inlet and outlet of fluids.

1.4.71 *Tube guide* is used with vertical tubes to restrict horizontal movement-while allowing the tube to expand axially.

1.4.72 *Tube retainer* is used to restrain horizontal radiant tubes from lifting off the intermediate tube supports during operation.

1.4.73 *Tube support or tube sheet* is any device used to support tubes

1.4.74 *Vapor barrier* is a metallic foil placed between layers of refractory as a barrier to flue gas flow.

1.4.75 *Volumetric heat release* is the beat released divided by the net volume of the radiant section, excluding the coils and refractory dividing walls, expressed in British thermal units per hour per cubic foot (kilowatts per cubic meter).

Heat Transfer Basics

Heat is a form of energy. Like other forms of energy, heat can be moved from one place to another. In a fired heater, heat is produced by burning fuel. This heat energy is then transferred to the oil that is being passed through the heater tubes. There are three different methods of heat transfer: convection, conduction, and radiation.

Radiant heat transfer is the transfer of heat by direct heat or light waves The transfer of the sun's-energy to the earth is an example of radiant heat transfer. The earth's source of heat and light comes from the radiant energy of the sun. The source and receiver must "see each other either directly or via an intermediate reradiating or reflecting surface. The driving force for radiant heat transfer is $T^4(hot) - T^4(cold)$.

Conduction is the way that heat travels through a solid substance. For example, if you heat one end of a metal bar, this heat will gradually be conducted throughout the bar. Some materials are better conductors of heat than others. Metal, for example is a better conductor of heat than ceramic or glass. So when maximum heat transfer is desired, we use a material that is a good conductor of heat.

Convection is the transfer of heat by the movement or flow of a liquid or gas. For example, if warm air is introduced into a cold room, the temperature difference produces air currents that will gradually flow throughout the room. The heat transfer occurs as the warm and cool air mix together. This mixing effect within a liquid or gas is heat transfer by convection.

The tubes in a furnace that are in direct line of sight with the burners receive rays of radiant heat. Since this section of tubes receives most of its heat by radiation, it is called the **radiant section**. The tubes in the upper section of the furnace are heated by the flow of hot gases that pass over them. This type of heat transfer is called convection. So the tubes in the upper part of the furnace make up what is called the **convection section**. The **shock bank** is located in the direct line of sight to the burners and in the flow steam of hot gases. So the tubes in the shock bank receive both radiant and convection heat.

The process fluid is heated as it flows through the tubes. First, the heat must pass through the wall of the metal tube. Heat is transferred through a solid substance like metal by conduction. When the heat reaches the process flow at the inside of the tube, a convection current is set-up that transfers heat throughout the process fluid.

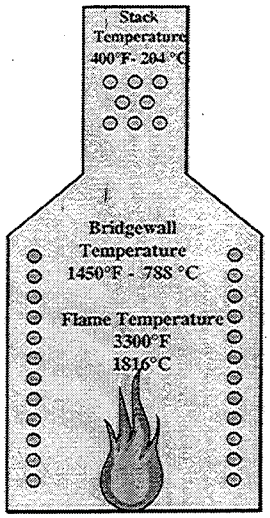
Training Services

Radiant Section

Section Outline

- Typical Furnace Temperatures
- Heat Distribution in Heaters
- Flux Distribution Around Tubes
- Peak Heat Flux
- Radiant Heat Balance
- Tube Wall Temperature (TWT) Calculations

Typical Furnace Temperatures



- Flame Temperature
  - Depends on Fuel Composition
- Radiant exit or Bridgewall Temperature
- Depends on Tube Temperatures and Heat Flux

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API Fired Heater Data Sheets

	COMBUSTION DESIGN CONDITIONS	
* 1. Type of fuel	fuel oil	fuel gas
* 2. Excess air, percent	25%	15%
3. Calculated heat release, Gcal/h (LHV)		
4. Calculated fuel efficiency, percent (LHV)	93.8	
5. Calculated thermal efficiency, percent (LHV)		
6. Guaranteed fuel efficiency, percent (LHV)	93.8	
7. Radiation loss, percent of heat release (LHV)	2.5	
8. Flue gas temperature leaving radiant section, °C	867	789
9. Flue gas temperature leaving convection section, °C	315	315
10. Flue gas temperature leaving air heater, (uncorrected), °C	187	
11. Flue gas temperature leaving air heater, (corrected), °C		
12. Maximum flue gas mass velocity through convection section, kg/m²s	1.95	
13. Volumetric heat release, kcal/m³ (LHV)		
* 14. Ambient air temperature, °C	10 / 41	
* 15. Air temperature, stack design, °C	41	
16. Air temperature leaving air heater, °C	113 / 251	
* 17. Altitude above sea level, m	0	
18. Minimum furnace draft/location, mm H2O	3mm / below convection tubes	

* 19. Note: A fuel saving of _____ Gcal/h will offset a $10000 increase in furnace cost (erected)

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Heat Flux

- **Heat Flux=** $\frac{HeatAbsorbed(Btu)}{OutsideSurfaceArea(ft2)}$
- **Outside Surface Area=(Π*D)**
- **Btu/h-ft2, W/m2, kcal/h-m2**

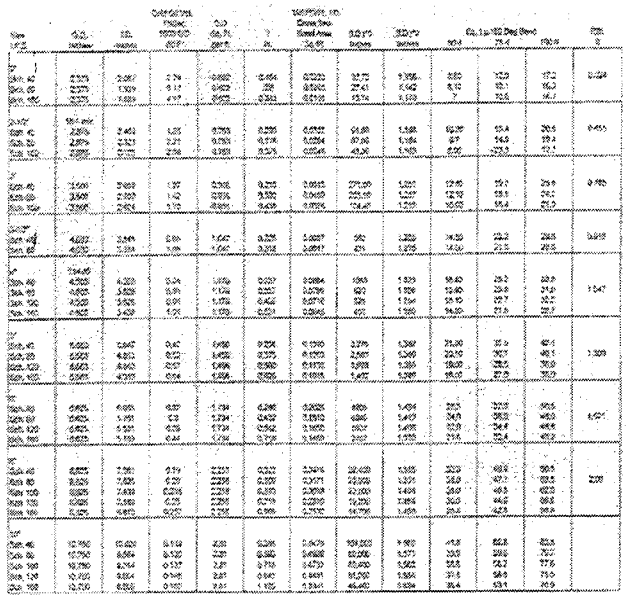
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Mass Velocity (In-tube)

- **Mass Velocity(G)=** $\frac{Proces,lb/\sec}{InsidePipeArea(ft2)}$
- **Inside Pipe Area=(0.7854*D^2)**
- **Lb/sec-ft^2, Kg/sec-m^2**

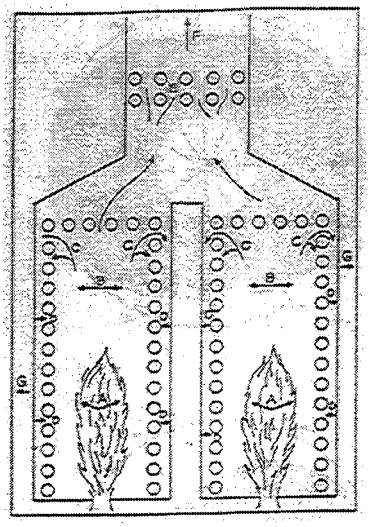
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Pipe Factors



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Heat Distribution in Heater



flame height ⅓ of radiant section height

Typical Heat Distribution In a Heater

A	Direct heat transfer through flame radiation	14%
B	Direct heat transfer through gas radiation	28%
C	Direct heat transfer through gas convection	6%
D	Indirect heat transfer through refractory reflection	12%
	Total heat transfer in radiant section	**60%**
E	Direct heat transfer through convection in convection bank	25%
	Total usefiul heat	**85%**
F	Flue gas loss	13%
G	Redintion loss	2%
	Total	**100%**

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Heat Distribution in Heaters

- **Radiant Section - 70%-75%**
- **Convection Section - 30%-25%**

Heat Distribution in Radiant Section

- **Flame Radiation - Visible Flame Light Waves**
- **Hot Gas Radiation - Combustion Products**
- **Convection - Flue Gas Flow Over Tubes**
- **Refractory Reflection - Mirror Effect**

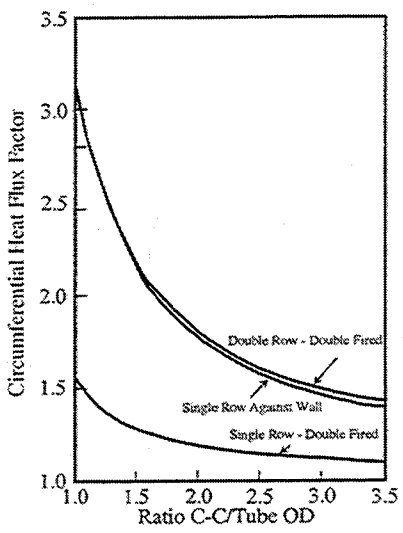
Hot Gas Radiation

- Combustion products H_2O and CO_2 are polar molecules that emit radiation at elevated temperatures
- N_2 and O_2 do not emit radiation

Flux Distribution Around Tubes

- Circumferential Flux Distribution
- Longitudinal Flux Distribution

Heat Flux Factor - Circumferential



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Peak Heat Flux

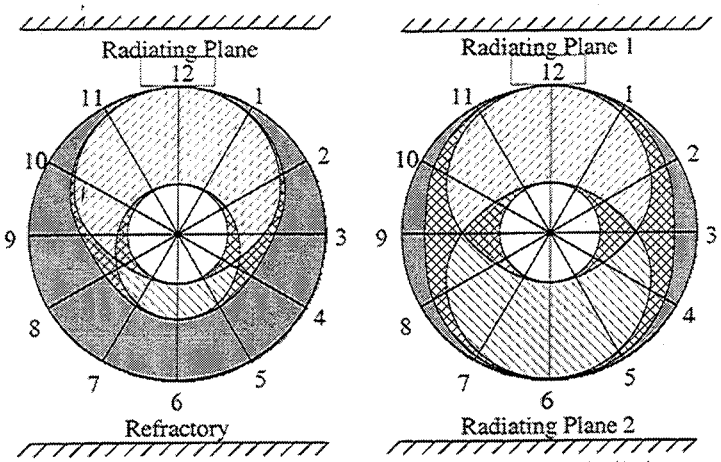
- **Peak Heat Flux**
  Average Flux x Circumferential Factor

- **Circumferential Flux Factors**
  Single Fired (2 diameters C to C) = 1.9
  Single Fired (3 diameters C to C) = 1.5
  Double Fired = 1.2

- **Peak Heat Flux = Maximum TWT**

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Flux Distribution Around Tubes



Primary Radiation Plane from One Side
Refractory from the Other

Direct Radiation
Reradiation

Primary Radiation Both Sides

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API Fired Heater Data Sheets

PROCESS DESIGN CONDITIONS

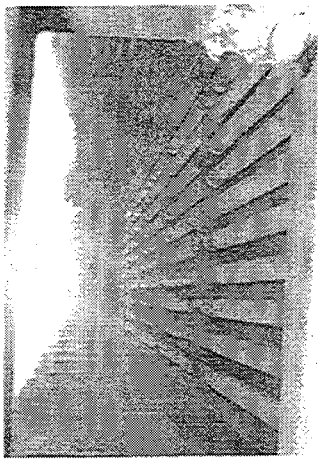
		Rad./Conv.	Convection	Rad./Conv.	Convection
1. Total duty per heater, Gcal/h					
2. Heater service					
* 3. Service					
4. Heat absorption, Gcal/h		78.99	69.[illegible]	83.55	0.[illegible]
5. Fluid name		HC	steam	HC	steam
* 6. Flow rate, kg/h		659,610	21,000	774,515	1,000
7. Flow rate, cm³/s					
* 8. Pressure drop (allowable, clean/fouled), kg/cm²		14.25	0.95	17.25	0.95
9. Pressure drop (calculated, clean/fouled), kg/cm²	clean	9.4	0.2	10.7	0.0
* 10. Fouling allowance, m²·°C·h/kcal					
* 11. Thermal conductivity of refractories, kcal/h·m·°C					
* 12. Average radiant flux density (allowable), kcal/h·m²				27125	
13. Average radiant flux density (calculated), kcal/h·m²				27125	
* 14. Average convection flux density (allowable), kcal/h·m²					
15. Average convection flux density (calculated), kcal/h·m²					
16. Maximum radiant flux density, kcal/h·m²				40875	
17. Maximum convection flux density, kcal/h·m²					
* 18. Velocity, ft/second					
* 19. Maximum inside film temperature (allowed/calc), °C					

Inlet Conditions

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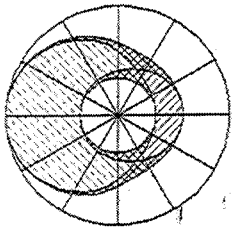
Single Fired Heater

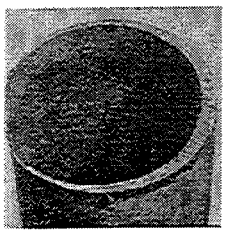
- Flame radiation from one side
- Uneven heating
- Front Face receives 6 times as much heat as back side of tube



Single Fired Heater

- Corrosion and Tube Thinning Follows Heater Flux Pattern





Maximum Average Radiant Heat Flux

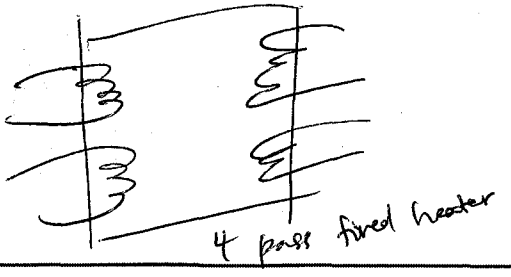
- Vertical cylindrical with tube length 20-30 feet
  - 12,000 Btu/h-ft^2
- Vertical cylindrical with tube length > 30 feet
  - 13,000 Btu/h-ft^2
- Cabin
  - 14,000 Btu/h-ft^2
- Double Fired U-tube
  - 22,000 Btu/h-ft^2

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Radiant Heat Balance

- Energy of flue gases at flame temperature minus energy at bridgewall temperature = energy absorbed by process minus radiant heat loss (~2%)

$$H_{T_{flame}} - H_{T_{BWT}} = H_{Process} + H_{Loss}$$



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Tube Wall Temperature (TWT)

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

where:

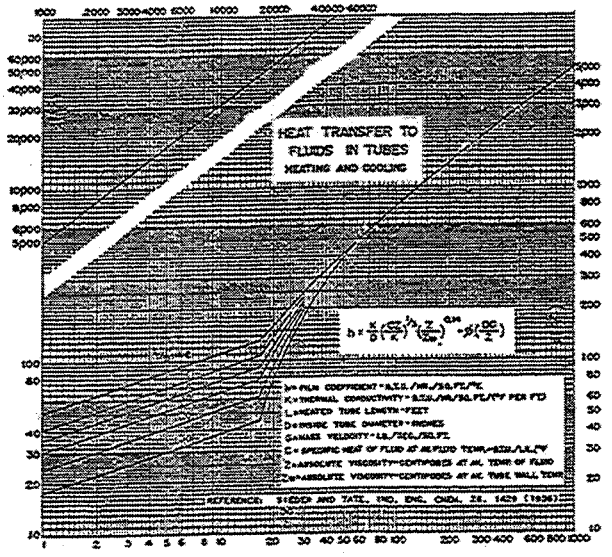
T_f = the bulk fluid temperature

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

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

PH 2004/2-21

TWT Calculations
(continued)



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Heat Transfer Coefficient(Inside Tube)

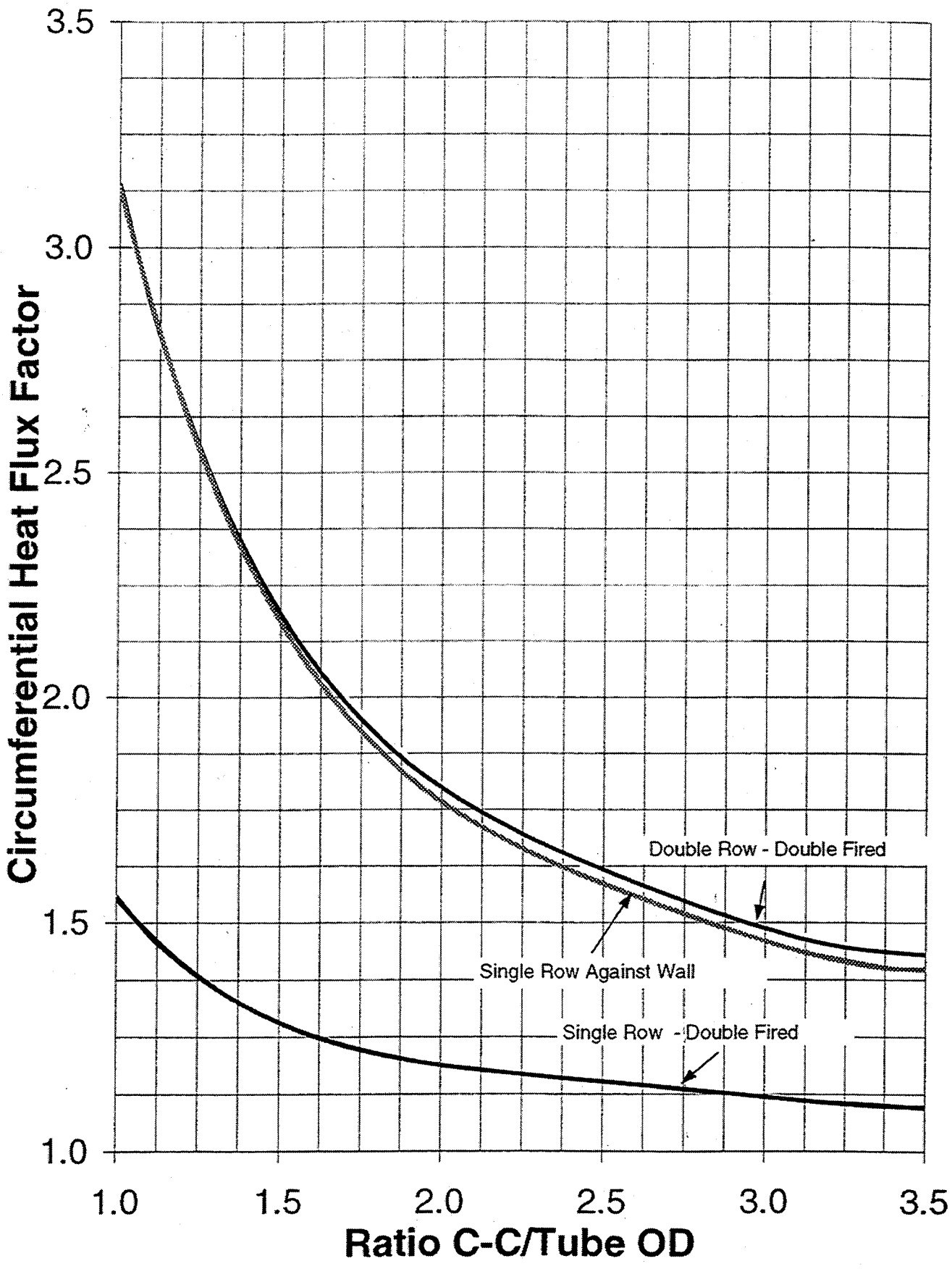
- $h_i = G^{.8}$
- Low ΔP Heaters ~170 Btu/h-ft2-°F
- High ΔP Heaters ~300-500 Btu/h-ft2-°F
- Boiling Water ~1000-1500 Btu/h-ft2-°F

Estimated Tube Wall Temperature (TWT)

TWT= Process Outlet + Factor(F)

F = 150°F for Platforming Heaters and Hydrocrackers
F = 60°F for Oleflex Heaters
F = 200°F for Vacuum Heater Outlet Tube
F = 60-100°F for all other heaters

Flux Distribution Around Tubes



Tube Wall Temperature (TWT)

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

where: **T_f = the bulk fluid temperature**

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

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

where: TWT = Tube Wall Temperature, °F

T_f = Average Fluid Temperature, °F

ΔT_f = Temperature Differential Across the Inside Film, °F

ΔT_m = Temperature Difference Across the Metal, °F

1. Temperature Difference Across the Film

It is common to express heat flux rates on the basis of the outside circumferential tube area. Expressing the flux rate on an outside circumferential tube area basis and correcting the equation since the flux rate specified occurs on the inside of the tube, we have:

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

where: A_O = Outside Circumferential Tube Area, ft^2

D_O = Outside Diameter of Tube, inches

D_i = Inside Diameter of Tube, inches

2. Temperature Difference Across the Metal and Deposits

The temperature differential across deposits results form conductive heat transfer. The determination of this differential is handled in an identical manner as the above. Most heater calculations are handled on a clean basis; deposits are not considered.

$$\Delta T_f = \left(\frac{q}{A_O}\right)\left(\frac{D_o}{2k}\right)\ln\left(\frac{D_O}{D_i}\right)$$

3. Heat Transfer Coefficient

For our purposes, 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:

$$h = \frac{k}{D}\left(\frac{CZ}{k}\right)^{.333}\left(\frac{Z}{Z_w}\right)^{.14}\Phi\left(\frac{DG}{Z}\right)$$

where: h = Inside Film Coefficient

k = Fluid Thermal Conductivity

D = Inside Tube Diameter

C = Specific Heat

Z = Fluid Viscosity at Bulk Fluid Temperature

G = Tube Side Mass Velocity

The individual heat transfer coefficient for hydrocarbon vapor phase systems may be expressed in the following correlation by McAdams:

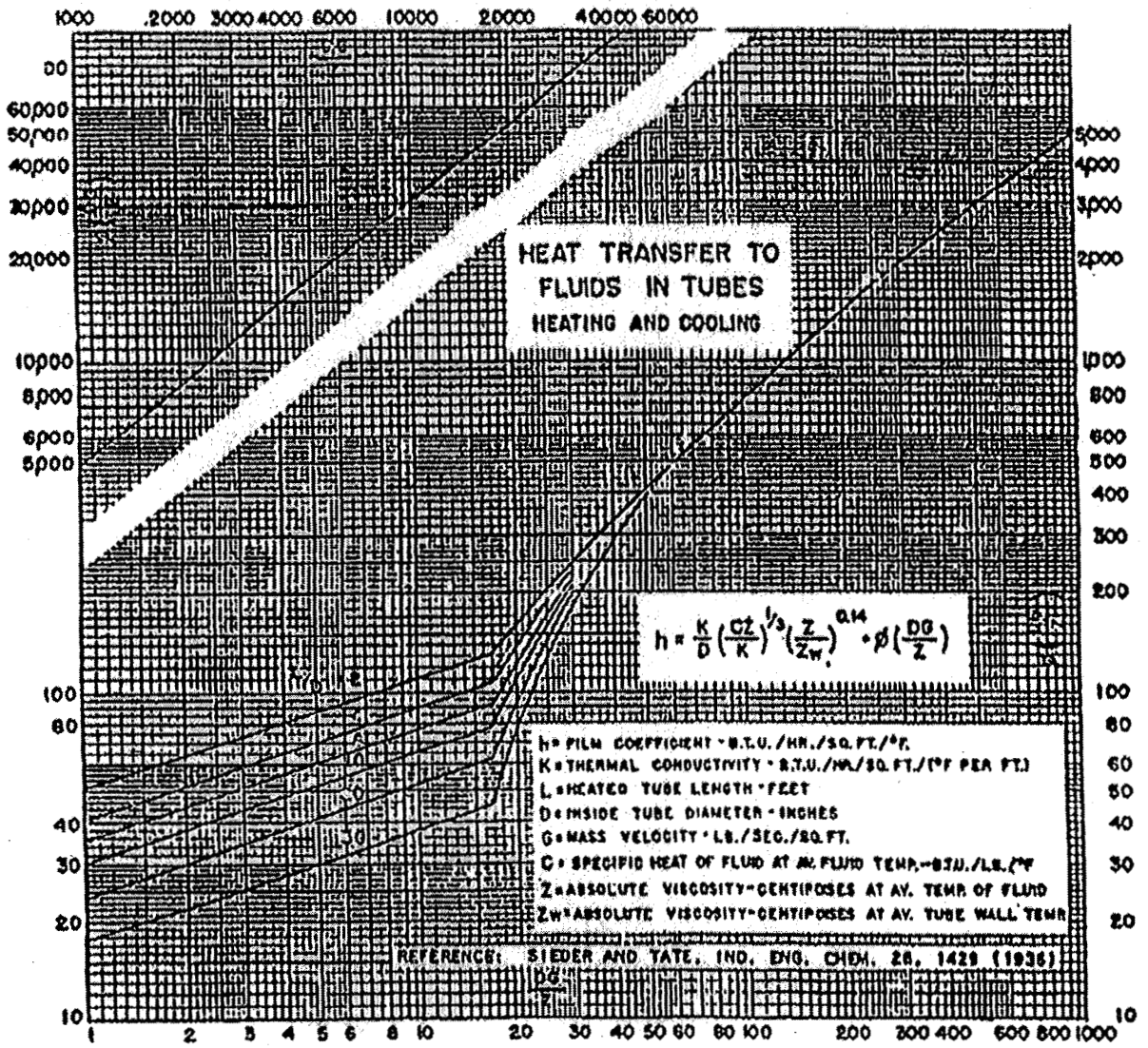
$$h = \frac{16.6\, C\, G^{0.8}}{D^{0.2}}$$

where: C = Specific Heat, Btu/lb °F

G = Mass Velocity, lb/ft^2 Sec

D = Inside Diameter of Tube, in.

TWT Calculations



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Pipe Fa...rs for Heaters

Size I.P.S.	O.D. Inches	I.D. Inches	Cold Oil Vel. Ft/Sec 1000 B/D 60 F	O.D. Sq. Ft. per ft.	t in.	Vol.Ft3/Ft. I.D. Cross Sectional Area Sq. Ft.	(I.D.)^5 Inches	(I.D.)^2 Inches	Eq. Lg.180 Deg Bend 50 d	75 d	100 d	RBL ft
2"												
Sch. 40	2.375	2.067	2.79	0.622	0.154	0.0233	37.72	1.156	8.60	12.9	17.2	0.524
Bch. 80	2.375	1.939	3.17	0.622	.218	0.0205	27.41	1.142	8.10	12.1	16.2	
Bch. 160	2.375	1.689	4.17	0.622	0.343	0.0156	13.74	1.110	7	10.6	14.1	
2-1/2"	58.1 min											
Bch. 40	2.875	2.469	1.96	0.753	0.203	0.0332	91.80	1.198	10.30	15.4	20.6	0.655
Sch. 80	2.875	2.323	2.21	0.753	0.276	0.0294	67.60	1.184	9.7	14.5	19.4	
Sch. 160	2.875	2.125	2.64	0.753	0.375	0.0246	43.30	1.163	8.90	23.3	17.7	
3"												
Sch. 40	3.500	3.068	1.27	0.916	0.216	0.0513	271.80	1.251	12.80	19.2	25.6	0.785
Sch. 80	3.500	2.900	1.42	0.916	0.300	0.0459	205.10	1.237	12.10	18.1	24.2	
Sch. 160	3.500	2.624	1.73	0.916	0.438	0.0376	124.40	1.213	10.90	16.4	21.9	
3-1/2"												
Sch. 40	4.000	3.548	0.95	1.047	0.226	0.0687	562	1.288	14.80	22.2	29.6	0.916
Sch. 80	4.000	3.364	1.05	1.047	0.318	0.0617	431	1.275	14.00	21.0	28.0	
4"	114.30											
Sch. 40	4.500	4.026	0.74	1.178	0.237	0.0884	1058	1.321	16.80	25.2	33.6	
Sch. 80	4.500	3.826	0.81	1.178	0.337	0.0799	820	1.308	15.90	23.9	31.9	1.047
Sch. 120	4.500	3.626	0.91	1.178	0.438	0.0716	625	1.294	15.10	22.7	30.2	
Sch. 160	4.500	3.438	1.01	1.178	0.531	0.0645	480	1.280	14.30	21.5	28.7	
5"												
Sch. 40	5.563	5.047	0.47	1.456	0.258	0.1390	3,275	1.382	21.00	31.5	42.1	
Sch. 80	5.563	4.813	0.52	1.456	0.375	0.1263	2,583	1.369	20.10	30.1	40.1	1.309
Sch. 120	5.563	4.563	0.57	1.456	0.500	0.1136	1,978	1.355	19.00	28.5	38.0	
Sch. 160	5.563	4.313	0.64	1.456	0.625	0.1015	1,492	1.340	18.00	27.0	35.9	
6"												
Sch. 40	6.625	6.065	0.32	1.734	0.280	0.2006	8206	1.434	25.3	37.9	50.5	
Sch. 80	6.625	5.761	0.3	1.734	0.432	0.1810	6346	1.419	24.0	36.0	48.0	1.571
Sch. 120	6.625	5.501	0.39	1.734	0.562	0.1650	5037	1.406	22.9	34.4	48.8	
Sch. 160	6.625	5.189	0.44	1.734	0.718	0.1469	3762	1.390	21.6	32.4	43.2	
8"												
Sch. 40	8.625	7.981	0.19	2.258	0.322	0.3474	32,400	1.515	33.3	49.9	66.5	
Sch. 80	8.625	7.625	0.20	2.258	0.500	0.3171	25,800	1.501	31.8	47.7	63.5	2.09
Sch. 100	8.625	7.439	0.215	2.258	0.593	0.3018	22,800	1.494	31.0	46.5	62.0	
Sch. 120	8.625	7.189	0.23	2.256	0.718	0.2819	19,200	1.484	30.0	44.9	59.9	
Sch. 160	8.625	6.813	0.257	2.258	0.906	0.2532	14,700	1.468	28.4	42.6	56.8	
10"												
Sch. 40	10.750	10.020	0.119	2.81	0.365	0.5475	101,000	1.586	41.8	62.6	83.5	
Sch. 80	10.750	9.564	0.130	2.81	0.593	0.4989	80,000	1.571	39.9	59.8	79.7	
Sch. 100	10.750	9.314	0.137	2.81	0.718	0.4732	69,400	1.563	38.8	58.2	77.6	
Sch. 120	10.750	9.064	0.145	2.81	0.843	0.4481	61,200	1.564	37.8	56.6	75.5	
Sch. 160	10.750	8.500	0.165	2.81	1.125	0.3941	44,400	1.534	35.4	53.1	70.9	

Training Services

Bridgewall Temperature Evaluation

Section Outline

- Stefan-Boltzman Equation
- UOP Method
- Lobo Evans Method

Stefan-Boltzman Equation

$$W_B = \sigma T^4$$

where:

W_B = Radiant energy rate

σ = Stefan-Boltzman constant (0.1713×10^{-8} Btu/h/ft^2/°R)

T = Absolute temperature, °R

FH 2004/3-3

Radiation Heat Transfer

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

where:

Q_r = Radiant heat transfer

A = Area (one of the surfaces)

F = Exchange factor

T_a = Absolute temperature of one surface

T_b = Absolute temperature of other surface

FH 2004/3-4

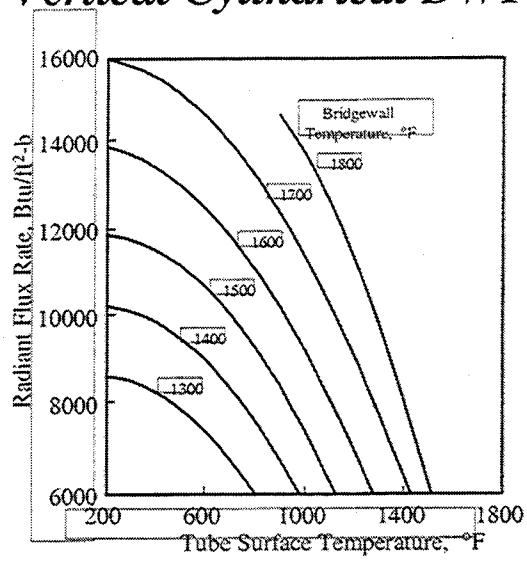
Bridgewall Temperature Evaluation

- How hard is the heater operating?
  - 1400-1500°F (760-815°C) is normal
  - 1600-1650°F (871-900°C) is approaching maximum limits
- Fouling of tubes
  - Every 100° fouling raises the BWT 50°

Lobo Evans Method

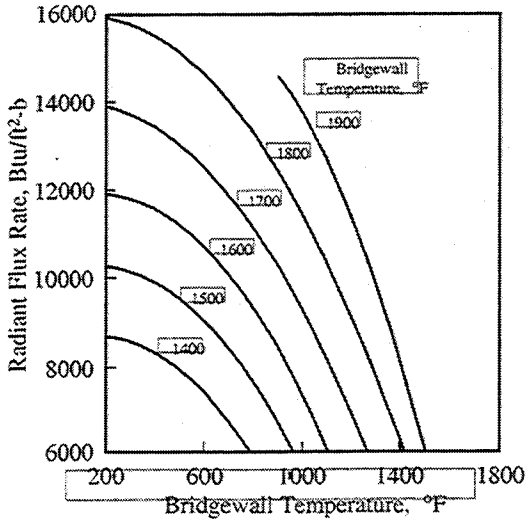
- Effective Surface
- Mean Length of Radiant Beam
- Partial Pressure of CO_2 and H_2O
- Flame Emissivity
- Effective Refractory Area
- Exchange Factor
- Rate of Heat Absorption Factor
- Bridgewall Temperature

Lobo Evans
Vertical Cylindrical BWT



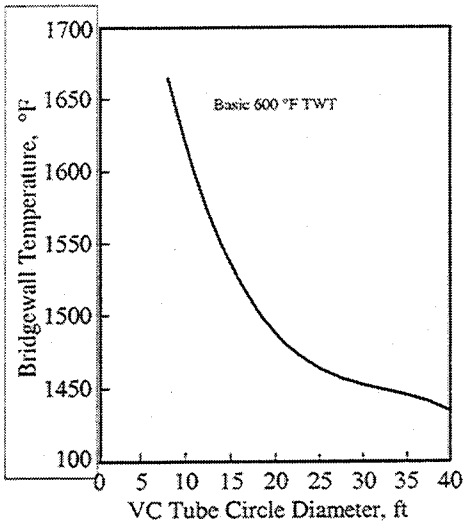
FH 2004/3-7

Lobo Evans
Horizontal Tubes BWT



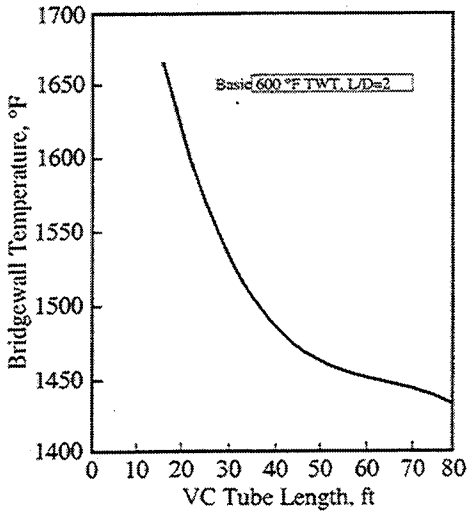
FH 2004/3-8

Lobo Evans - Vertical Cylindrical Tube Circle vs. BWT



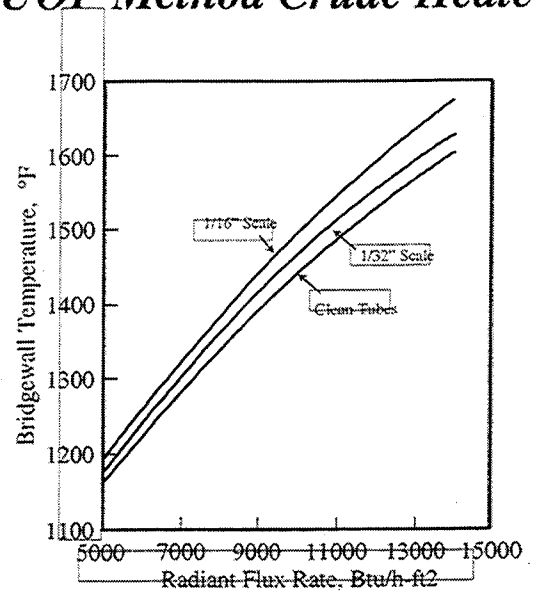
FH 2004/3-9

Lobo Evans - Vertical Cylindrical Tube Length vs. BWT



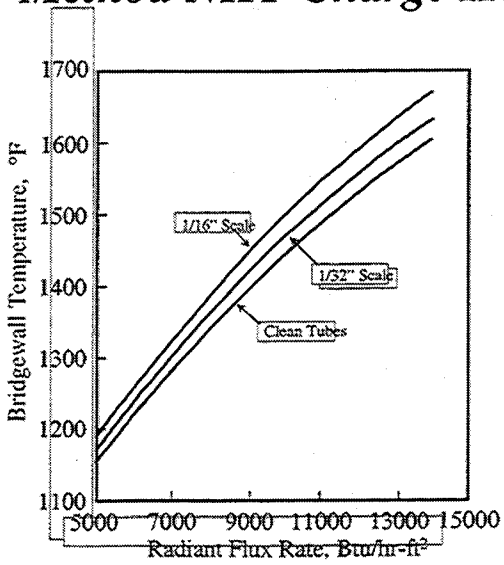
FH 2004/3-10

UOP Method Crude Heater



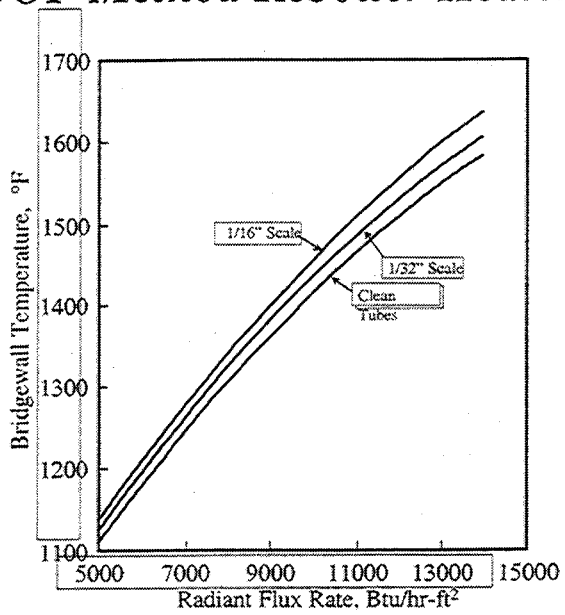
FH 2004/3-11

UOP Method NHT Charge Heater



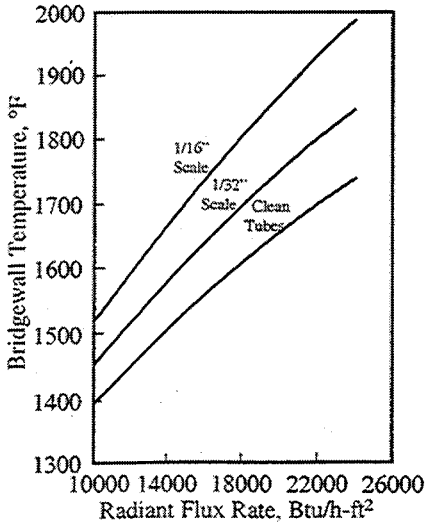
FH 2004/3-12

UOP Method Reboiler Heater



FH 2004/3-13

UOP Method Platforming Double Fired U Tube Heater



FH 2004/3-14

BWT

The Bridgewall temperature is the flue gas radiant exit temperature. It is an excellent indication of how the radiant section is performing. Radiant heat transfer is the T^4*hot gas*- T^4*tube*. The BWT is the solution to this equation.

The BWT is primarily affected by the radiant flux rate and the tubewall temperature. The radiant flux rate has a greater effect than TWT. Flame temperature does not affect BWT that much because the hot gas temperature quickly drops down to the BWT range.

The BWT is the most important item in evaluating an operating heater. The BWT is an indirect measure of the radiant flux rate.

BWT evaluation is extremely important, both in new heater design and field troubleshooting. For most process heaters, the tube metal temperature (TWT) is estimated at 100°F above the process temperature. Calculate the TWT, read up to a flux rate, and determine the BWT.

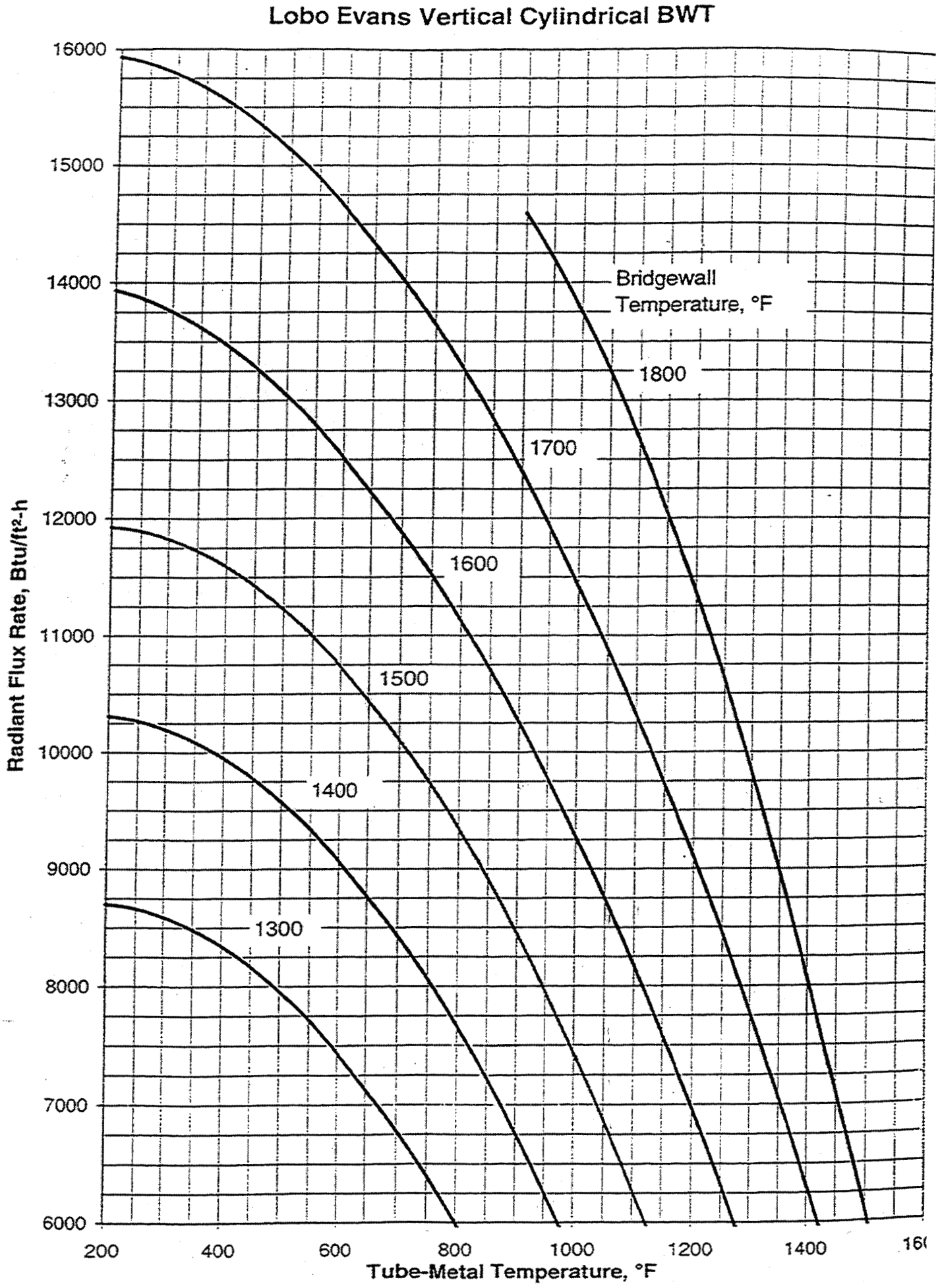
For coking or fouling services, measure the TWT with a pyrometer. Use this TWT and flux rate to determine BWT.

The charts labeled *Lobo-Evans Method, Vertical Tube Heaters and Horizontal Tube Heaters* were obtained from the Herb Berman article, part 3, enclosed in Section 14 of this manual.

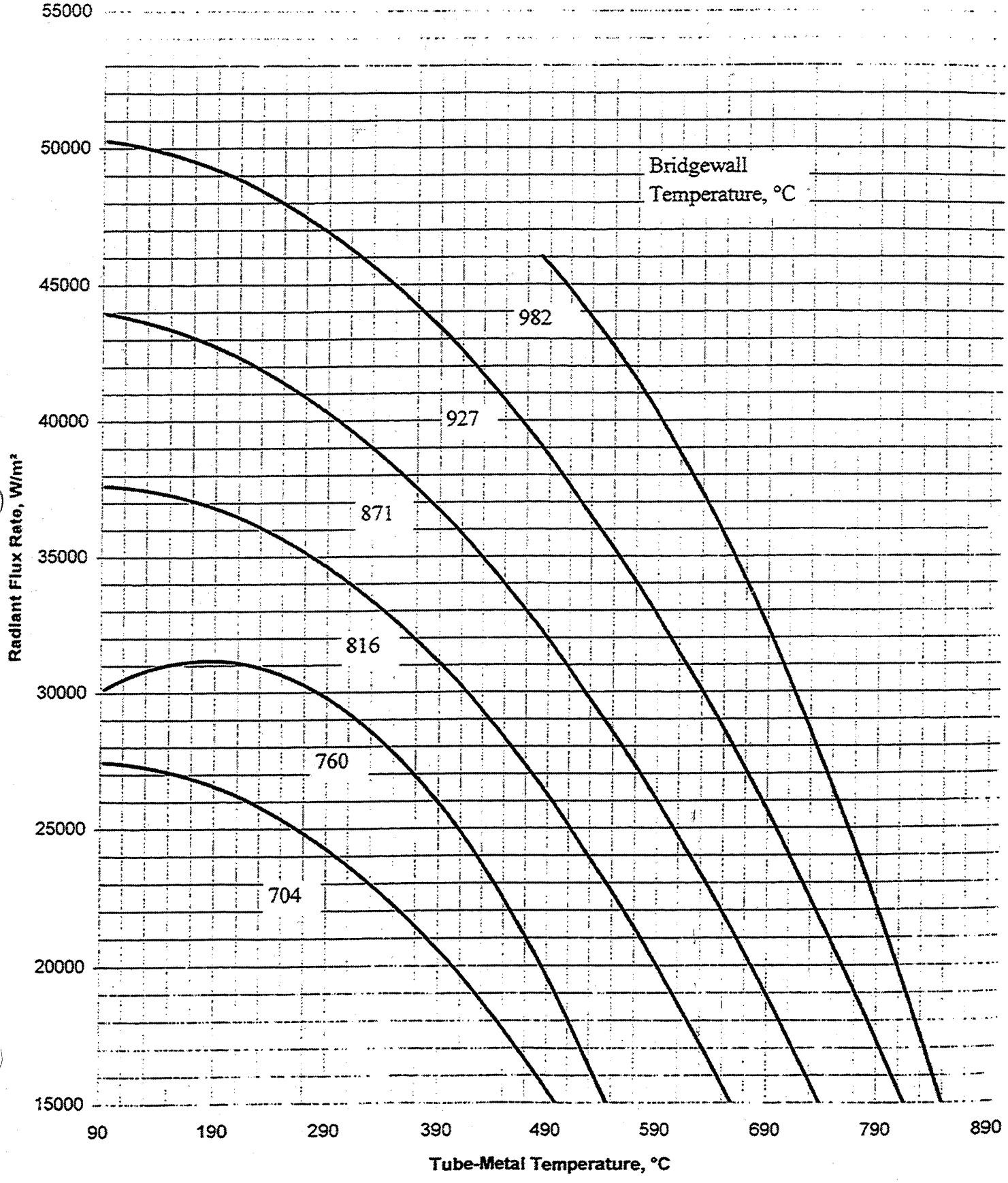
The charts *Lobo-Evans Method, Vertical Cylindrical Heater, Tube Length versus BWT, and Tube Circle Diameter versus BWT* show the effect of geometry on heater performance (BWT).

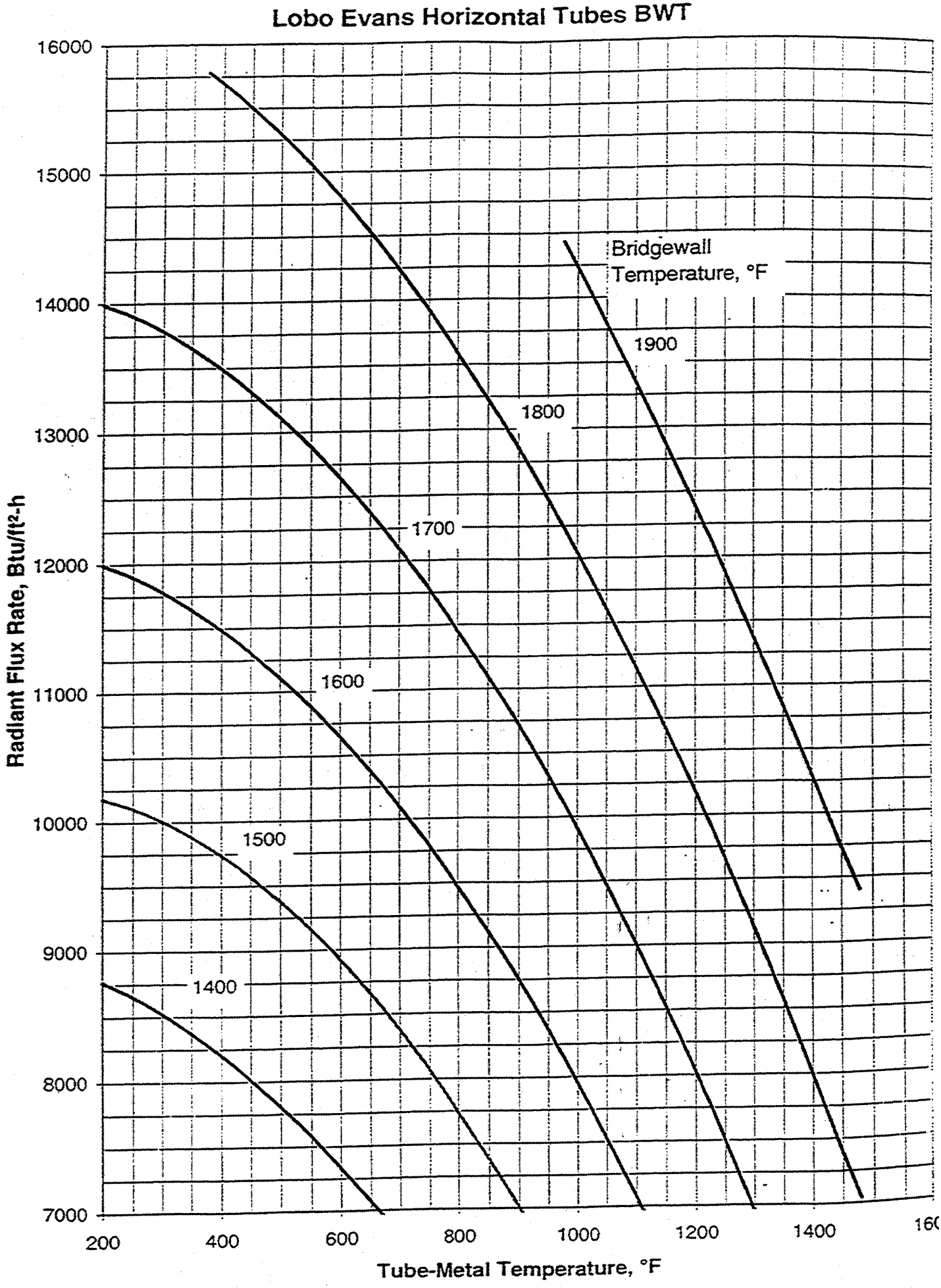
Short heater tubes often result in high BWTs. Large heaters have lower BWTs than small heaters.

A more detailed description of the Lobo- Evans method in listed in the Wimpress article *Rating Fired Heaters* enclosed in Section 14.

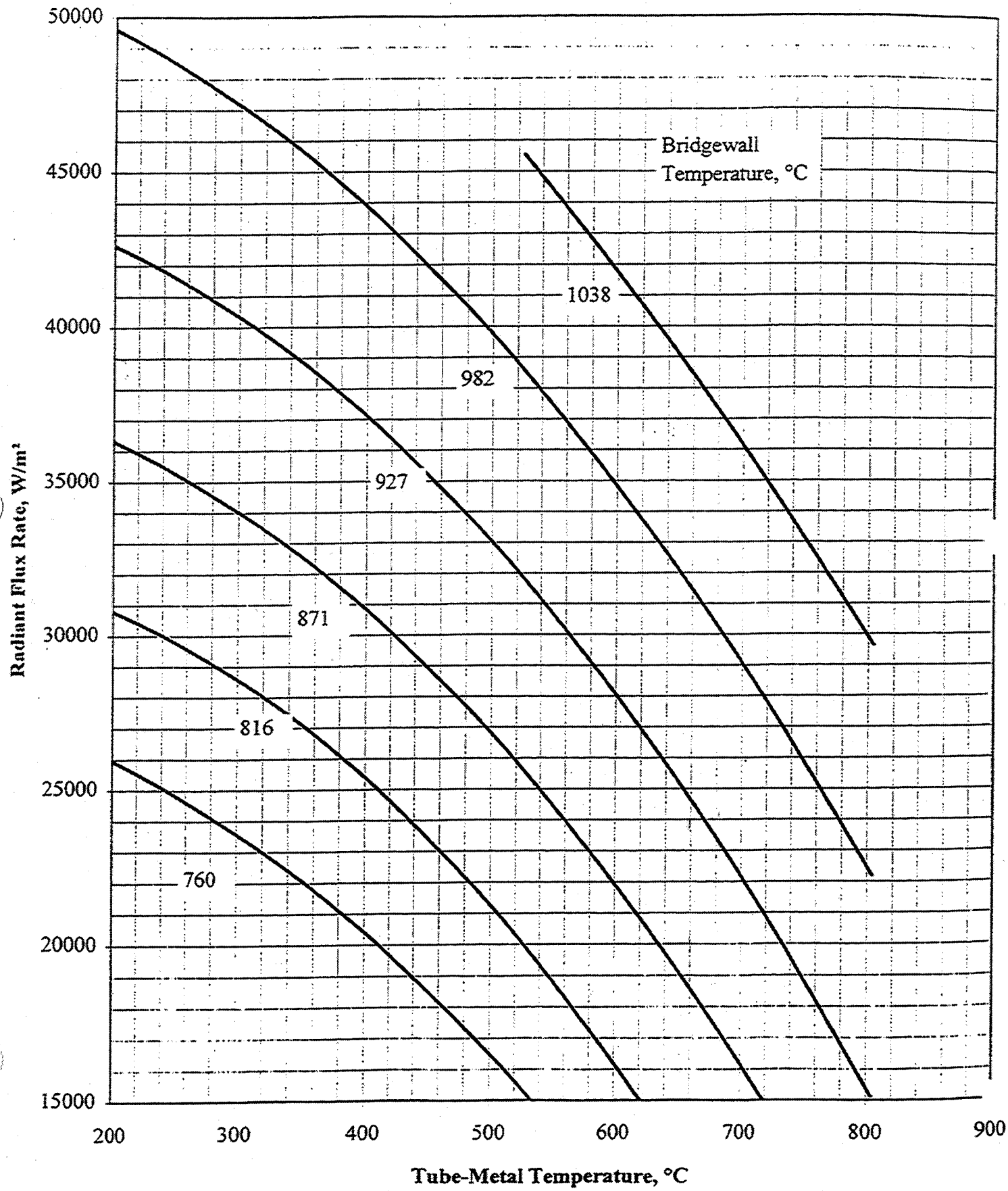

Lobo Evans Vertical Cylindrical BWT
16000
15000
14000
13000
12000
11000
10000
9000
8000
7000
6000
Radiant Flux Rate, Btu/ft²-h
Bridgewall
Temperature, °F
1800
1700
1600
1500
1400
1300
200
400
600
800
1000
1200
1400
Tube-Metal Temperature, °F

Lobo-Evans Method
Vertical Cylindrical Heaters
Bridgewall Temperature Profile

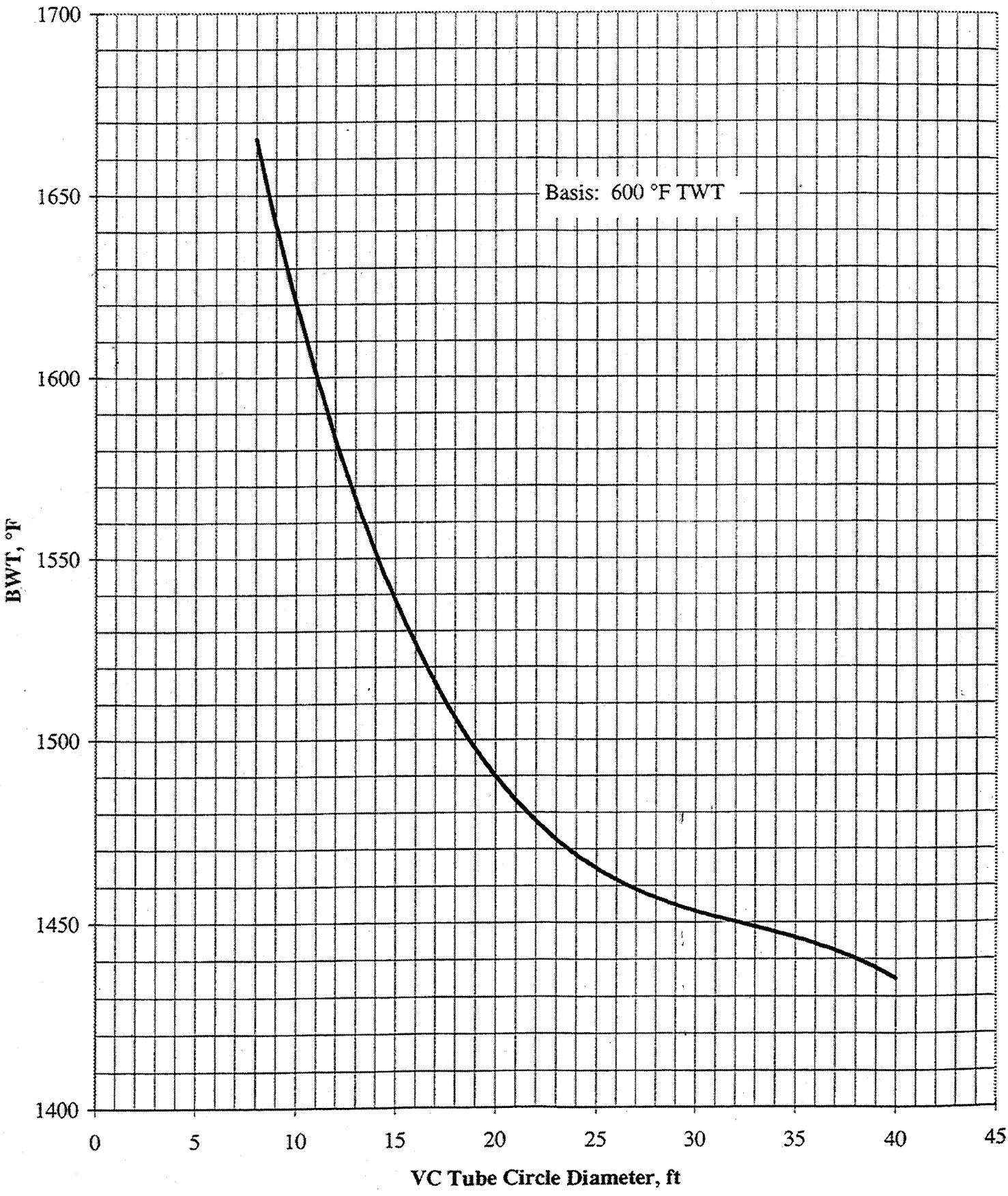



Lobo Evans Horizontal Tubes BWT
Bridgewall Temperature, °F
1900
1800
1700
1600
1500
1400
Radiant Flux Rate, Btu/ft²-h
16000
15000
14000
13000
12000
11000
10000
9000
8000
7000
200
400
600
800
1000
1200
1400
Tube-Metal Temperature, °F

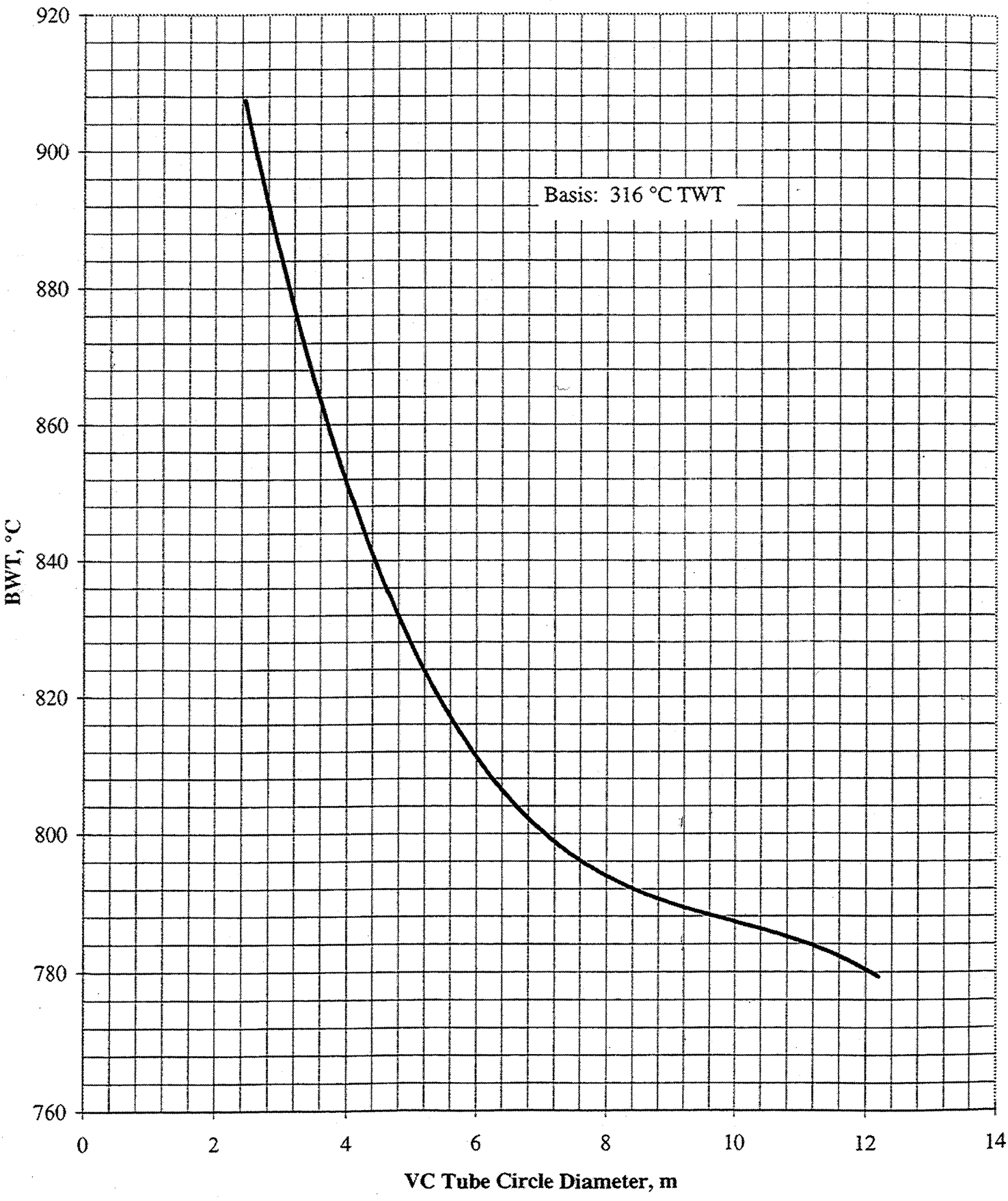
Lobo-Evans Method
Horizontal Tube Heaters
Bridgewall Temperature



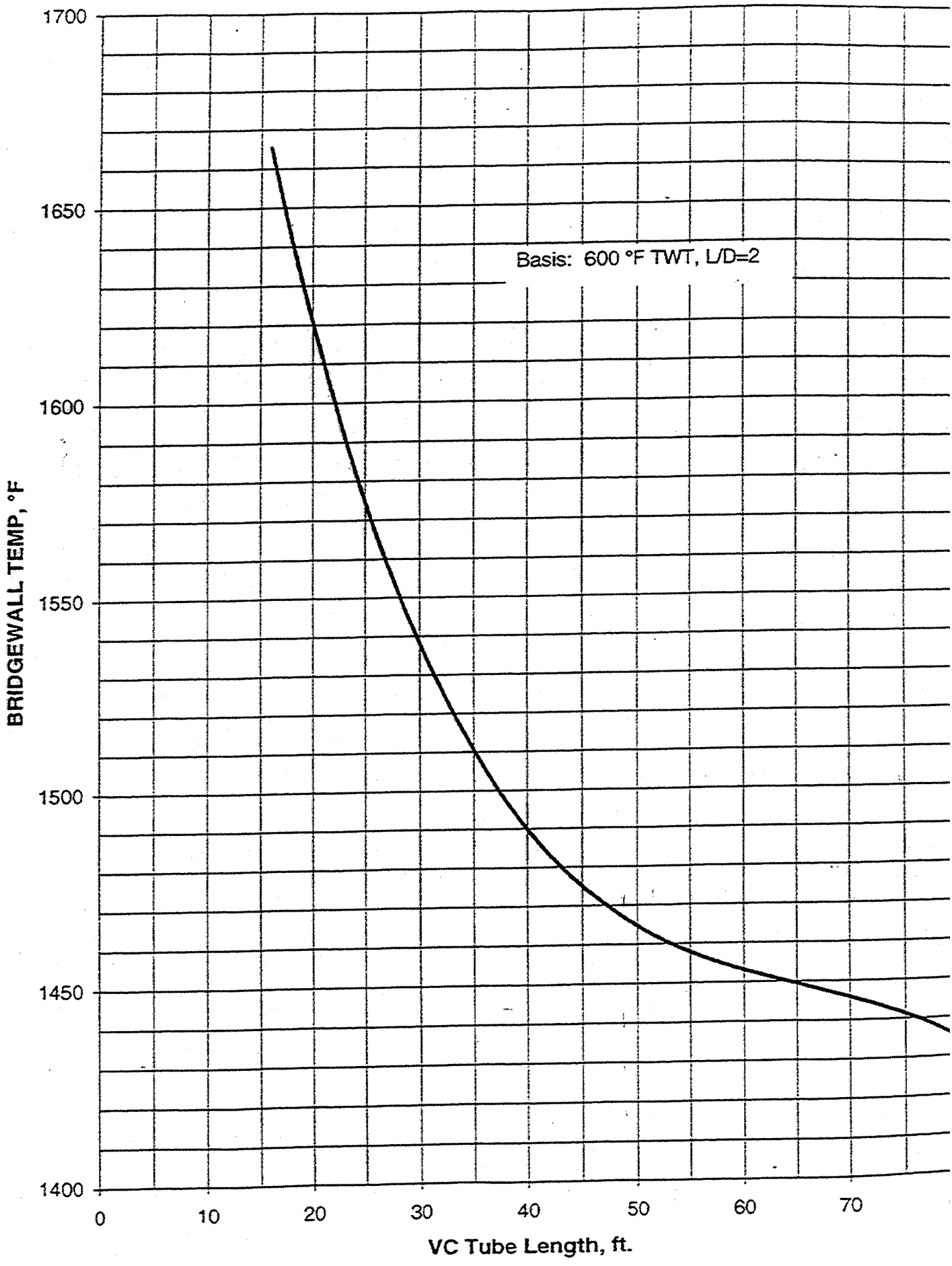
Lobo-Evans Method
Vertical Cylindrical Heater Tube Circle Diameter vs. BWT



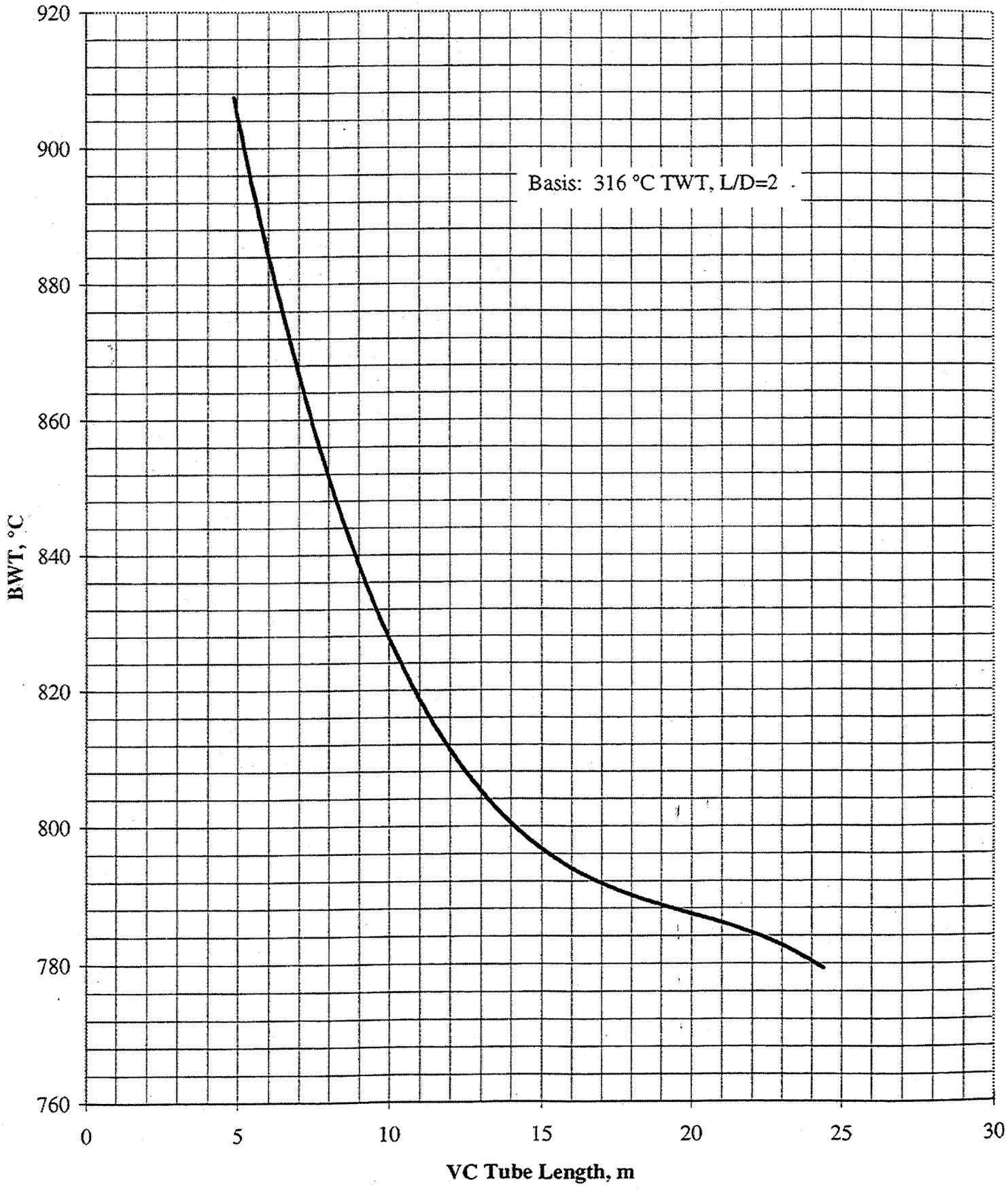
Lobo-Evans Method
Vertical Cylindrical Heater Tube Circle Diameter vs. BWT



Lobo Evans Vertical Cylindrical Tube Length vs BWT



Lobo-Evans Method
Vertical Cylindrical Heater Tube Length vs. BWT

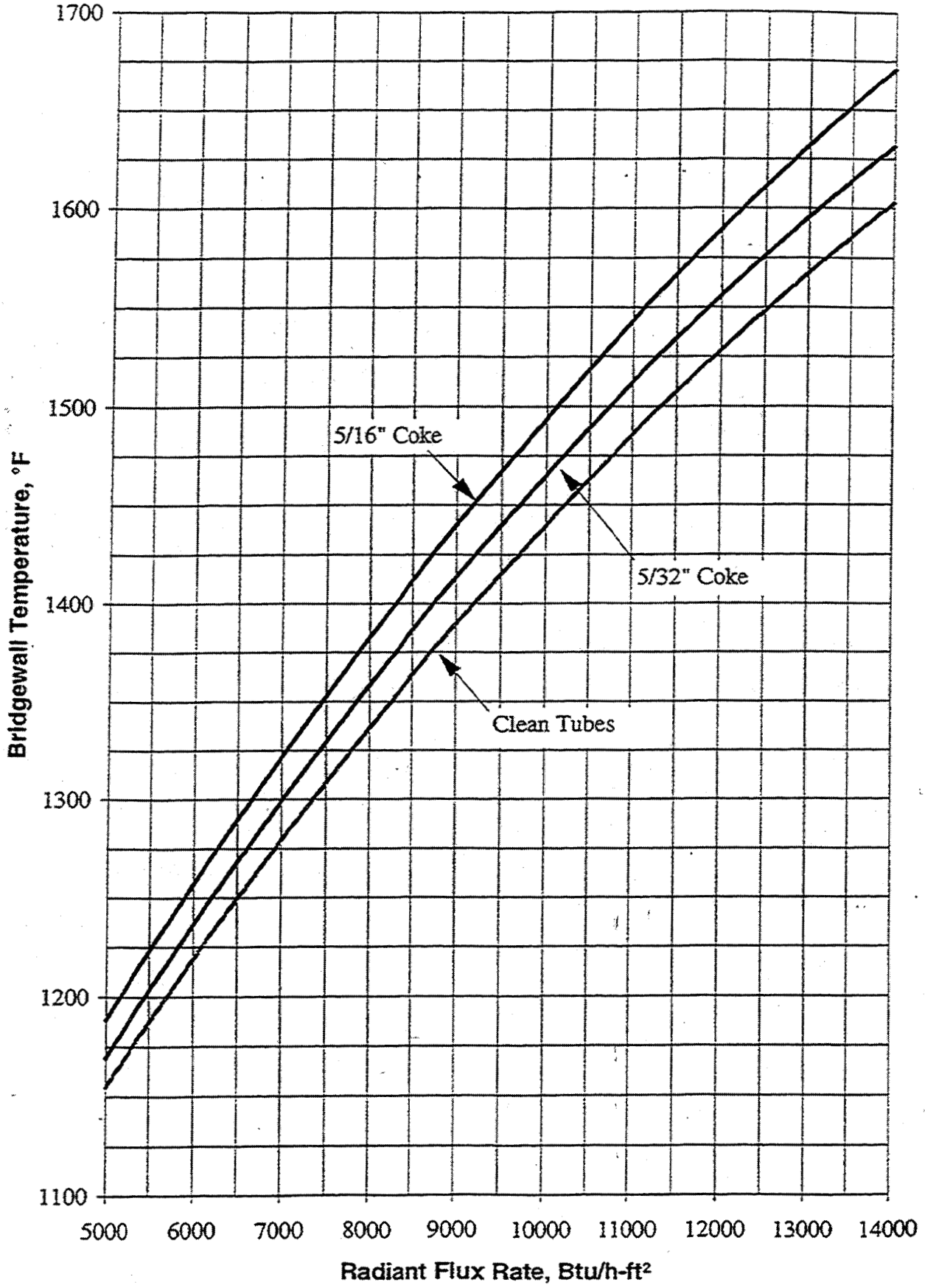


BWT Evaluation- UOP Method

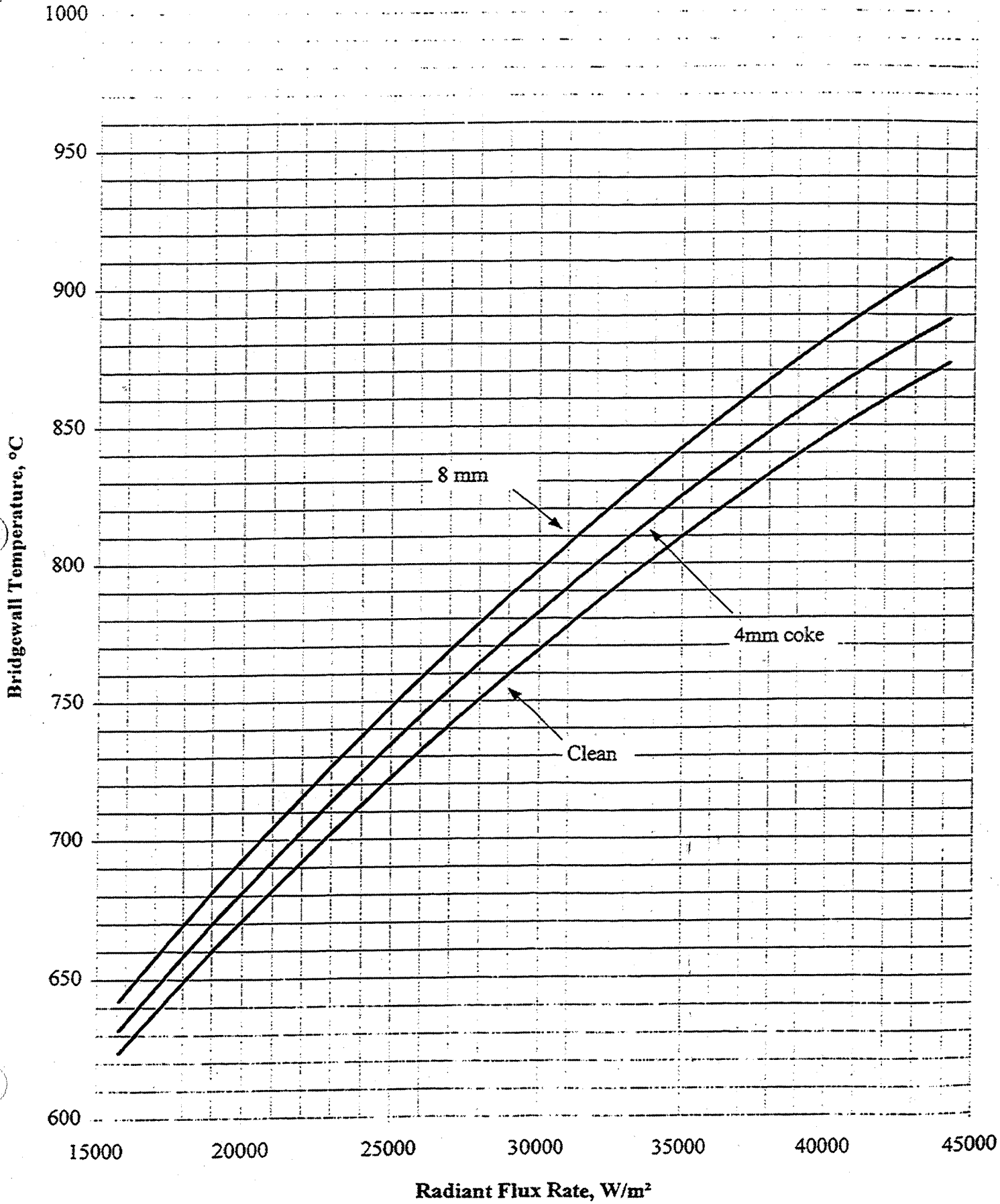
The Lobo-Evans Method often gives a BWT that is too high. Many companies modify the method to more closely match field data.

The UOP simulator was used to produce a series of charts for common refinery services. These charts are easy to use. These charts show the effect of tube fouling on BWT.

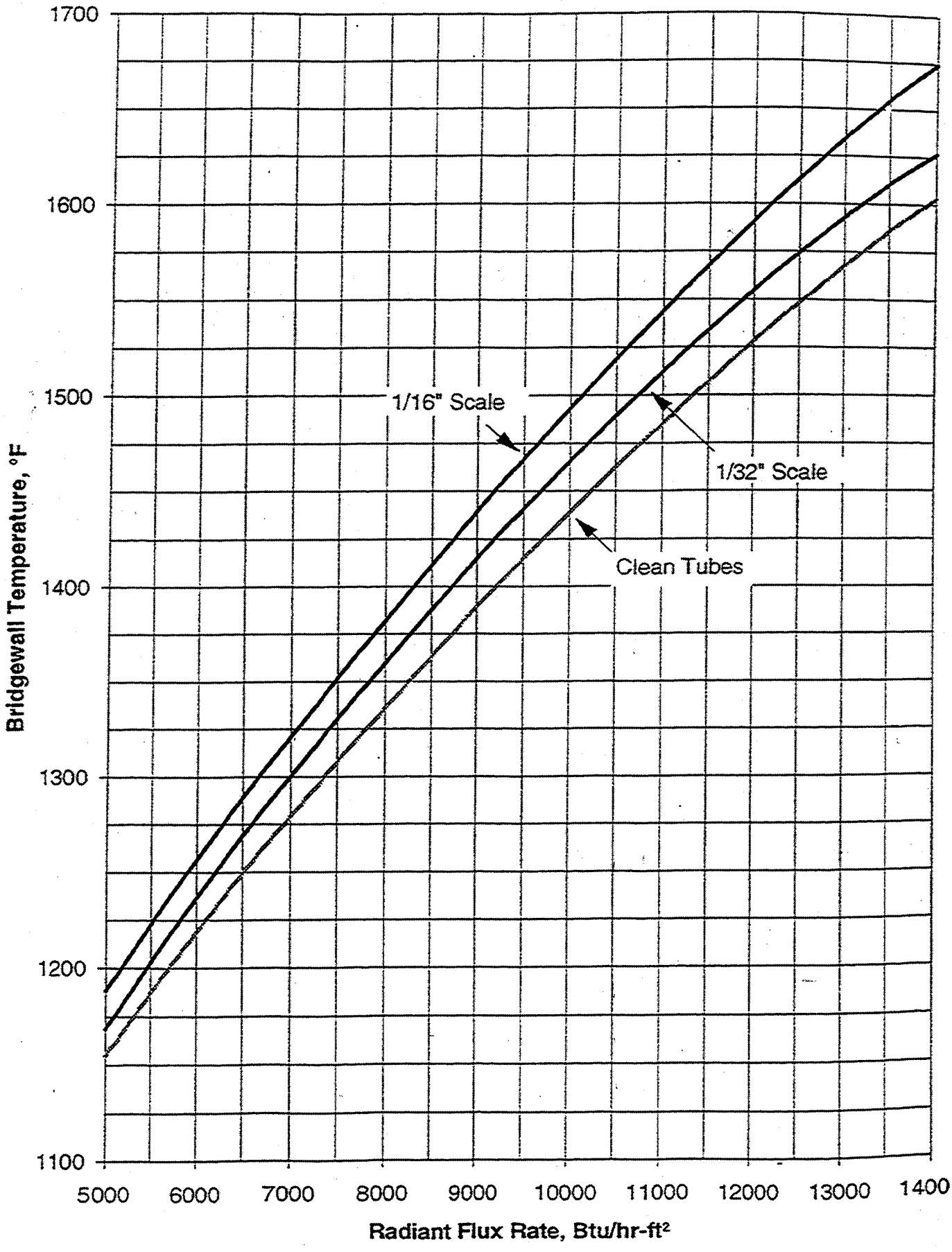
UOP Method Crude Heater



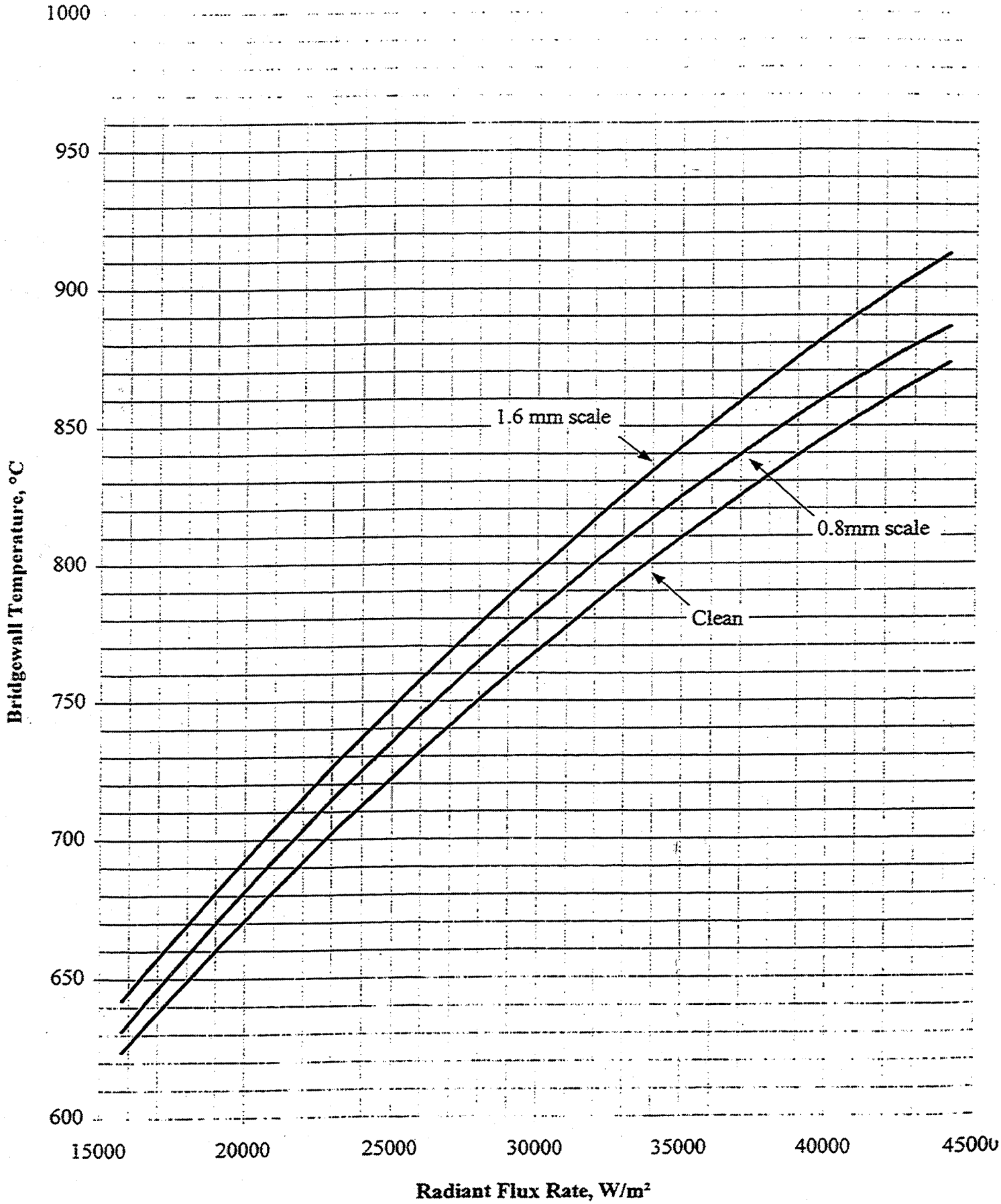
UOP Method
Crude Unit Heater



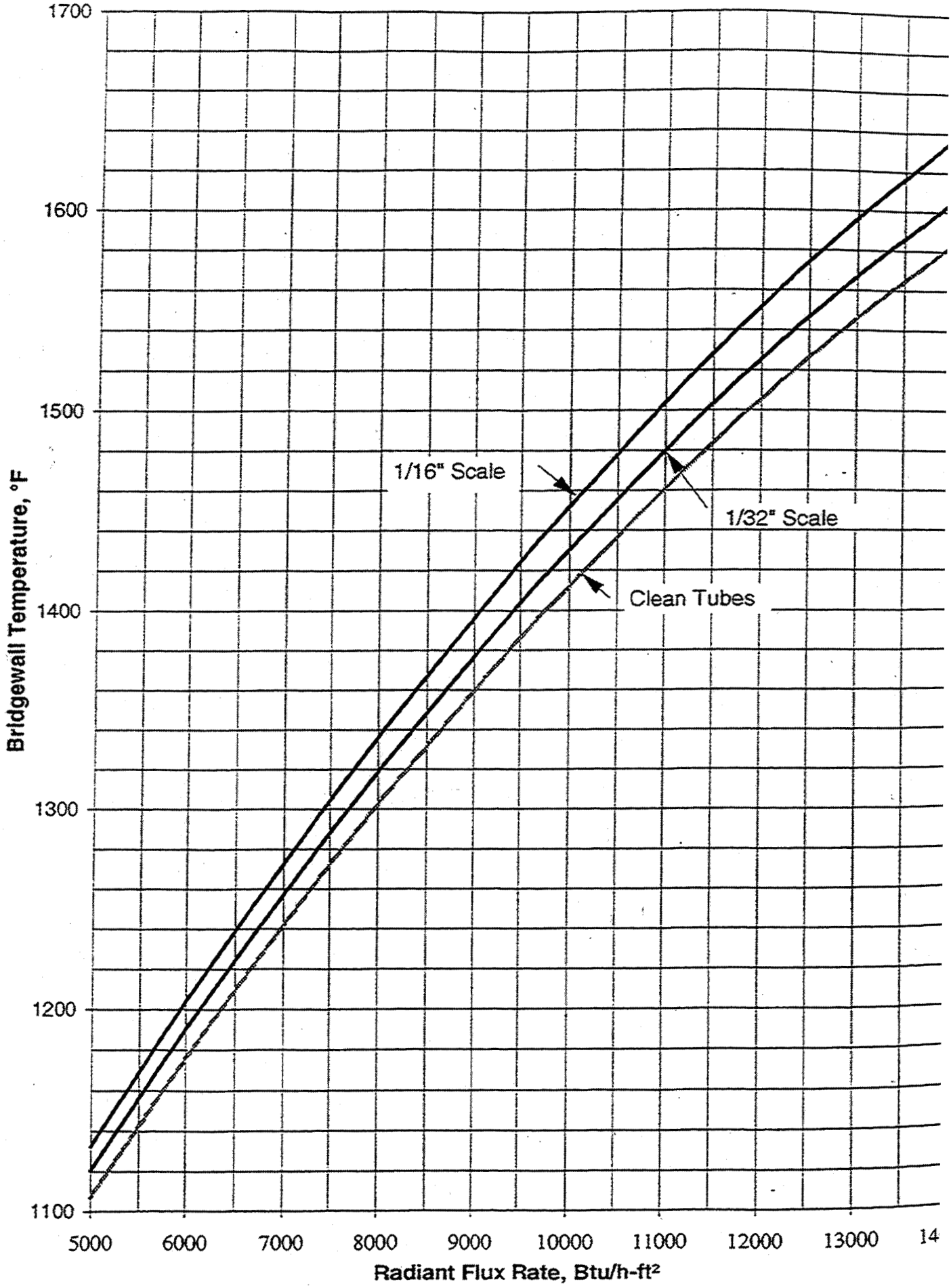
UOP Method NHT Charge Heater



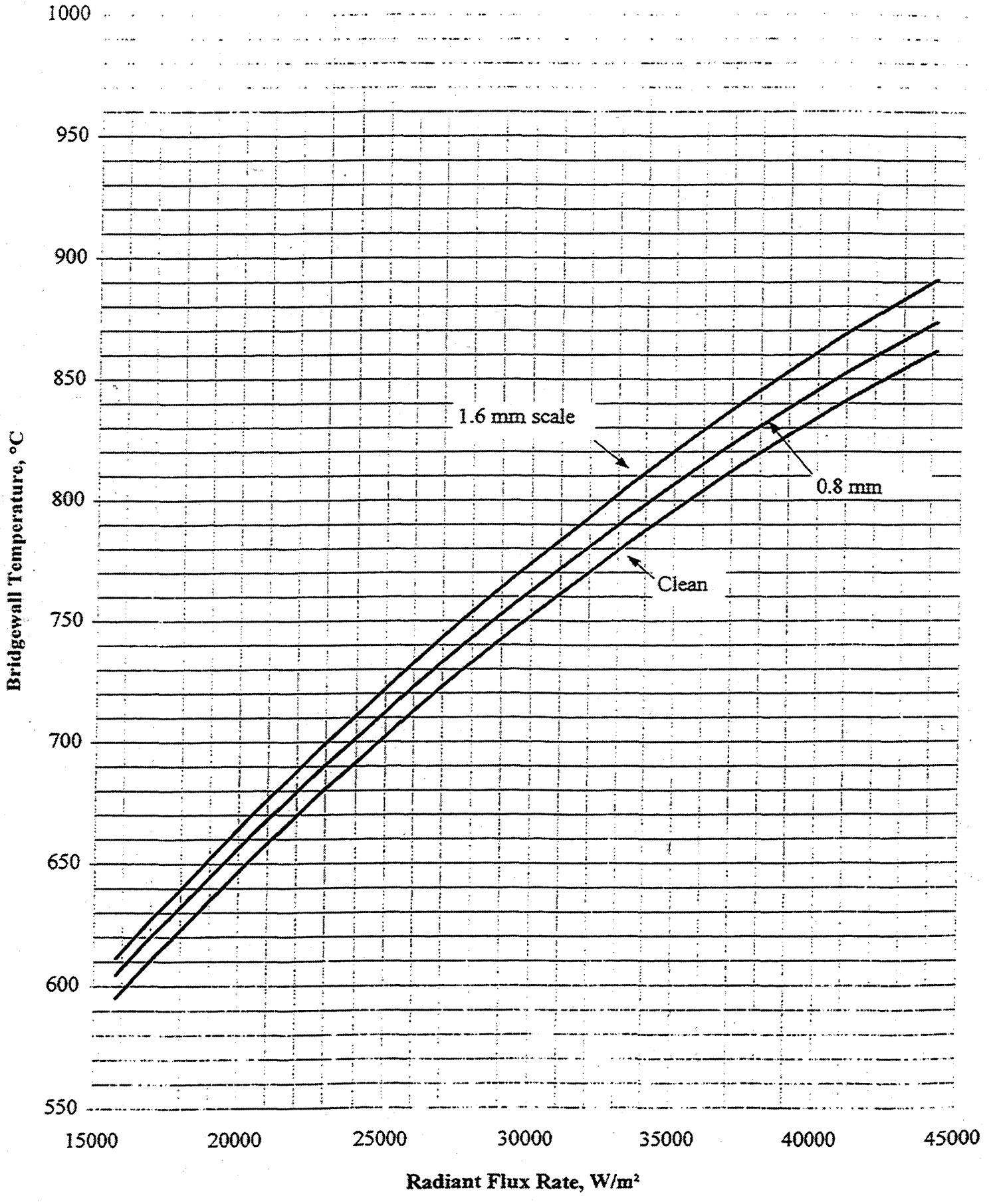
UOP Method
NHT Charge Heater



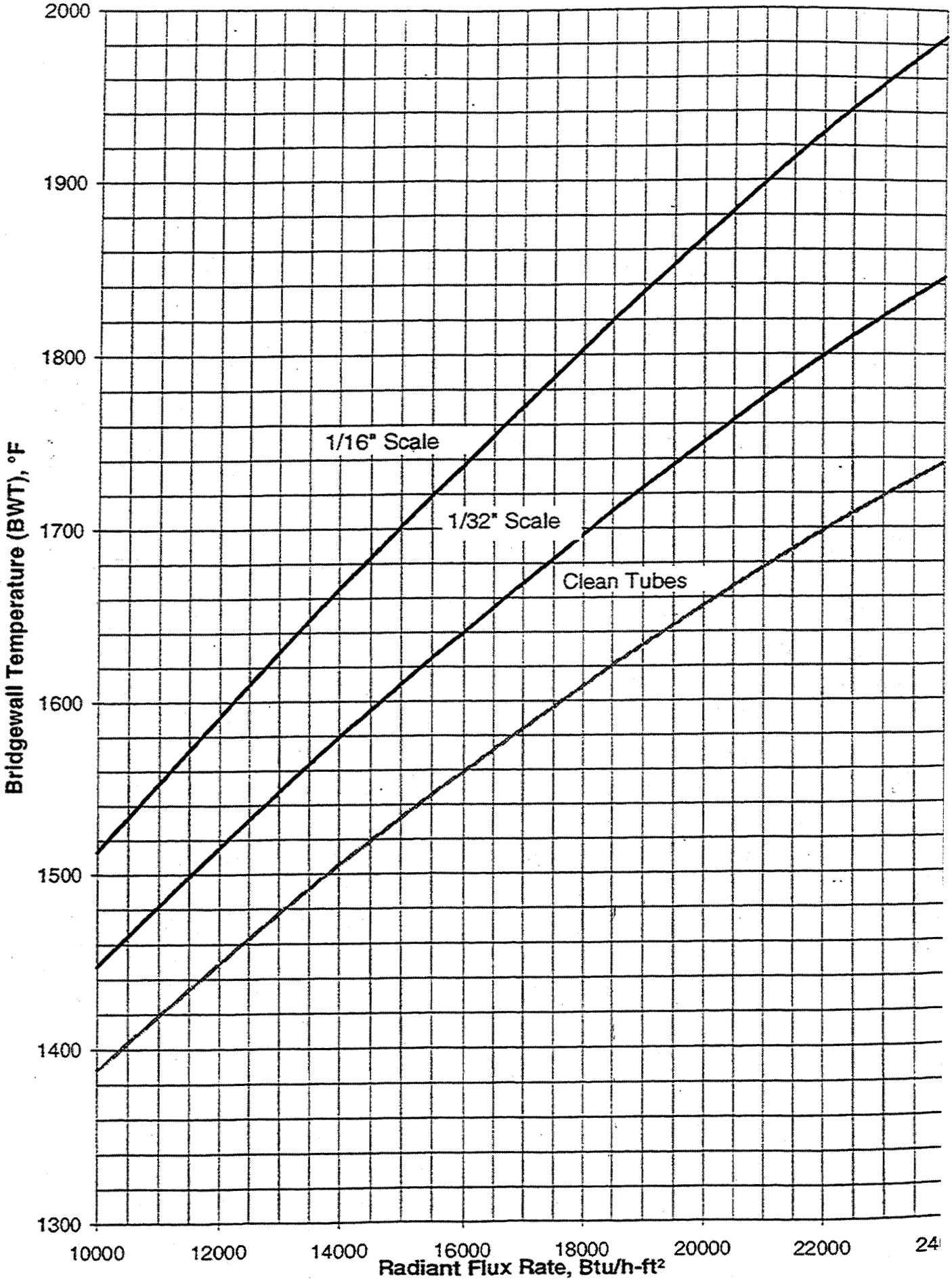
UOP Method Reboiler Heater



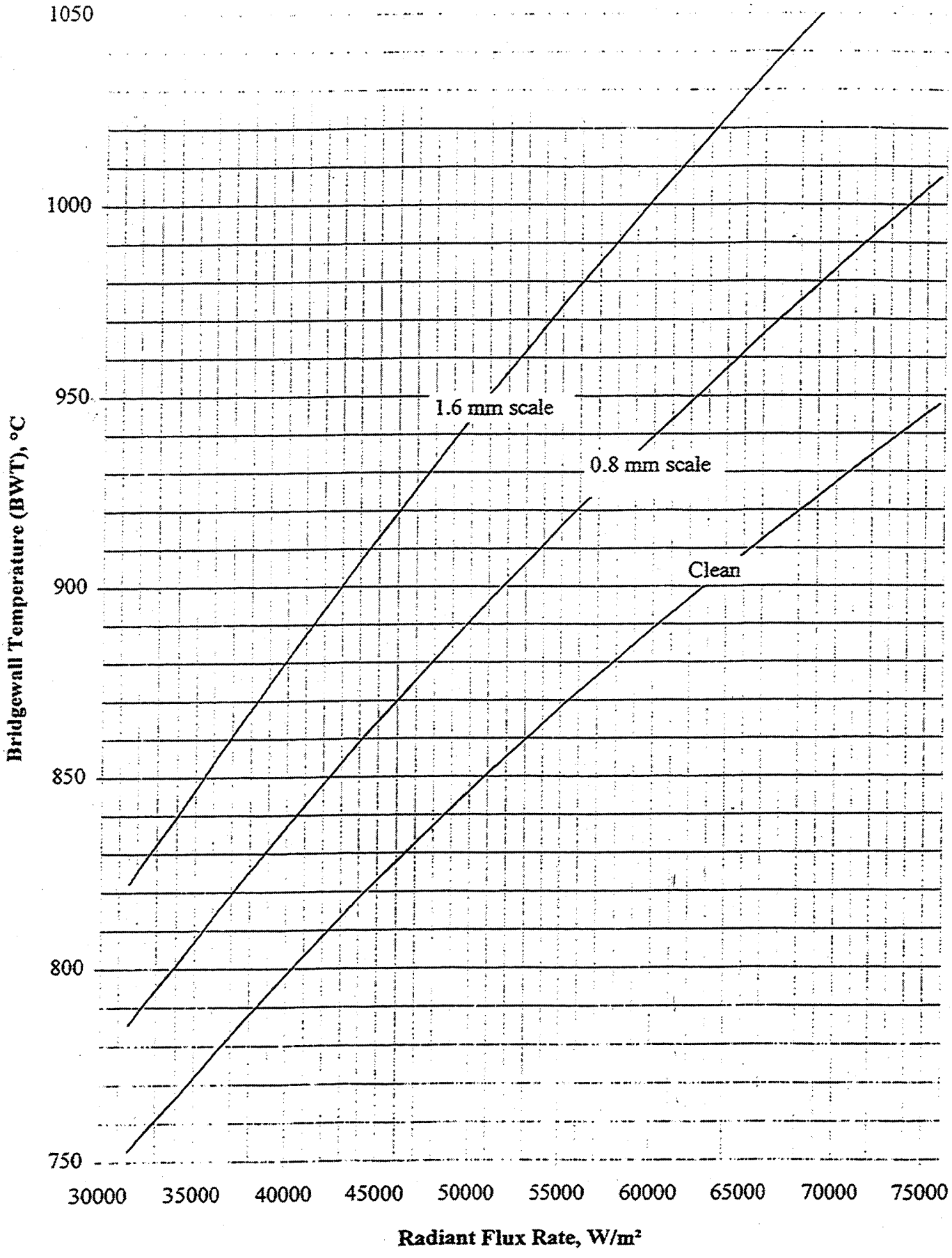
UOP Method
Reboiler Heater



UOP Method Platforming Double Fired U Tube Heater



UOP Method
Platforming Double Fired U-Tube



Radiant Section - Lobo-Evans Method

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

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

$$W_B = \sigma T^4$$

where: W_B = Radiant Energy Rate

σ = Stefan-Boltzman Constant = 0.1713×10^{-8} Btu/Hr, Sq.Ft., $°R^4$

T = Absolute Temperature, °R

The relationship for radiant heat transfer between two real surfaces at temperatures Ta and T_b, may be expressed as follows:

$$q_R = \sigma AF(T_a^4 - T_b^4)$$

where: q_R = Radiant Heat Transfer, Btu/Hr

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

F = Exchange Factor

T_a = Absolute Temperature of One Surface, °R

T_b = Absolute Temperature of Other Surface, °R

The area, A, is the area of either of the surfaces. The value of the exchange factor depends on which surface is used. The exchange factor is complex. It will be covered in more detail later.

A. Equivalent Cold Plane Area

The heat-absorbing surface (the heater tubes) is usually well defined within a furnace as opposed to the heat emitting surface (the firebox gases). Furthermore, the rate of heat transfer per unit of absorbing area serves as a basis of design. Therefore, it has become accepted practice to use the heat-absorbing, or "cold", surface as the area used in computing the rate of radiant heat transfer.

The usual radiant section, heat-absorbing surface consists of a number of parallel cylindrical tubes in front of a refractory wall. Radiation from the hot gases strike the tubes directly and is absorbed. Some of the radiant beams miss the tubes and contact the refractory. The heated refractory reradiates heat both to the back end of the tubes and back into the furnace. The tubes accept direct radiant heat from the firebox gases and reradiated heat from the refractory.

This complicated situation is handled in the calculations by replacing the tube bank with an equivalent plane area. It is the area of an ideal black plane having the same absorbing capacity as the actual tube bank. The equivalent area is the plane surrounding the tube bank, the cold plane area (A_{cp}), multiplied by a correction factor, α. The correction factor, α, is known as the effectiveness factor. The effectiveness factor places the cold plane area in terms of an equivalently sized black body. It considers the tube bank will not absorb all of the radiant energy beamed in its direction. Hottel has published values for the effectiveness factor, α,

B. Exchange Factor

The remaining term to be evaluated is the exchange factor, F. The exchange factor depends on the relative area of the tubes and refractory, the emissivity and absorptivity of the heat absorbing surface, the tubes, and the heat emitting "surfaces", the firebox gases. The gas in the firebox is far from an ideal black body. The only constituents in normal flue gas that contribute significantly to the radiant emission are the polar molecules, carbon dioxide, water, sulfur dioxide, etc. The non-polar molecules, nitrogen, oxygen, etc., are transparent to radiation.

The emissivity of a body of gas depends on the concentration of the polar molecules, the furnace dimensions, and the temperatures of the gas and of the absorbing surface. Lobo and Evans have shown that the composition and dimensional effects can be accounted for by a single term, the partial pressure of carbon dioxide plus water multiplied by the mean beam length. The partial pressure of other polar molecules is usually small, so it is ignored. One chart shows the partial pressure, P, of the radiating components as a function of excess air for the usual hydrocarbon fuels.

The mean beam length is the average depth of the gas envelope within the geometrical boundaries of the volume encasing the radiant heat transfer area. The table gives the mean beam length, L, for various furnace shapes. It has been found that the tube wall temperature has only a minor effect. Therefore, the emissivity can be correlated as a function of the PL product and gas temperature, as shown in preceding charts. as a function of tube arrangement and spacing.

The cold plane area may be determined from the following equation:

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

where: A_{cp} = Cold Plane Area, ft^2

N = Number of tubes

L = Length of tubes, effective, ft.

S = Center-to-center spacing, ft.

The equivalent cold plane area, EPA, is the cold plane area multiplied by the effectiveness factor:

$$EPA = \alpha A_{cp}$$

where: EPA = Equivalent Cold Plane Area, ft^2

α = Effectiveness Factor, dimensionless (From Chart)

The shield tubes require special handling. These tubes are not backed by a reradiating refractory wall. Energy that is not directly absorbed by the shield tubes passes beyond them to the tubes in the convection section. This may lead us to believe the effectiveness factor for the shield tubes should be lower than that for those backed by refractory since there is no reradiation. However, all the flue gases leaving the radiant section flow over the shield tubes (also known as the shock bank). Considerable heat convective heat transfer occurs. To correct for this additional heat transfer in the shield tubes, Lobo-Evans considers the effectiveness factor of these tubes to be unity (1.00).

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

The preceding charts consider the tubes do not completely absorb all the radiant energy that strikes them. The curves are based on a tube-surface absorptivity of 0.9, a commonly accepted value for oxidized metal surfaces.

The working equation for calculating radiant heat transfer in the firebox thus becomes:

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

where: q_{Rc} = Radiant heat transfer by radiation in Radiant Section.

F = Exchange Factor.

α= Efficiency Factor (Hottel)

T_g = Flue Gas Temperature, °R

T_t = Tube Wall Temperature, °R

C . Convective Transfer in Radiant Section

Although radiation accounts for most of the heat transfer in the radiant section, convective heat transfer 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 and the choice of burners. The relation for convective heat transfer is shown below:

$$q_{Rc} = h_{Rc} A_{Rt} (Tg - Tt)$$

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

T_g = Firebox Temperature, °R

T_t = Tube Wall Temperature, °R

Since convective heat transfer is not the major contributor, we make some simplifying approximations. For the usual furnace, h_{Rc} is about 2.0 Btu/hrft^2°F, and A_{Rt} is about two times the effective plane area, αA_{Cp}. Refer to Section 14, page 322 for detailed equations.

D. Total Heat Transfer Rate

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

E. Radiant Heat Balance

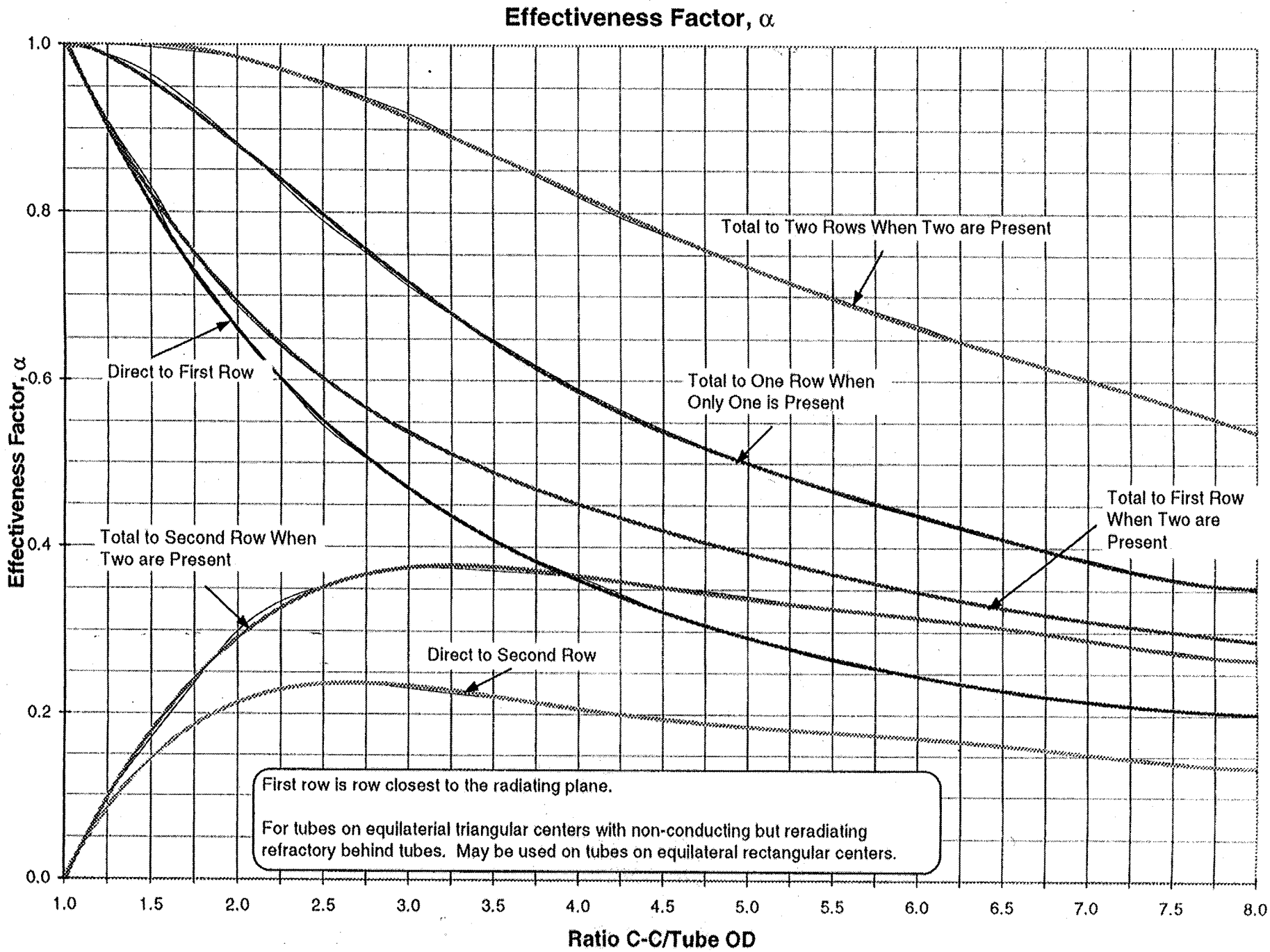
A heat balance is made around the furnace. Heat is put into the radiant section from three sources: the net heat of combustion, q_n; the sensible heat of the combustion air, q_a; and the sensible heat of the fuel, q_f. Heat is removed in three ways: that absorbed by the process fluid, q_R; the heat losses through the refractory/insulation, q_L; and the sensible heat of the exit flue gas, q_g . The heat balance is:

$$q_n + q_a + q_f = q_R + q_L + q_g$$

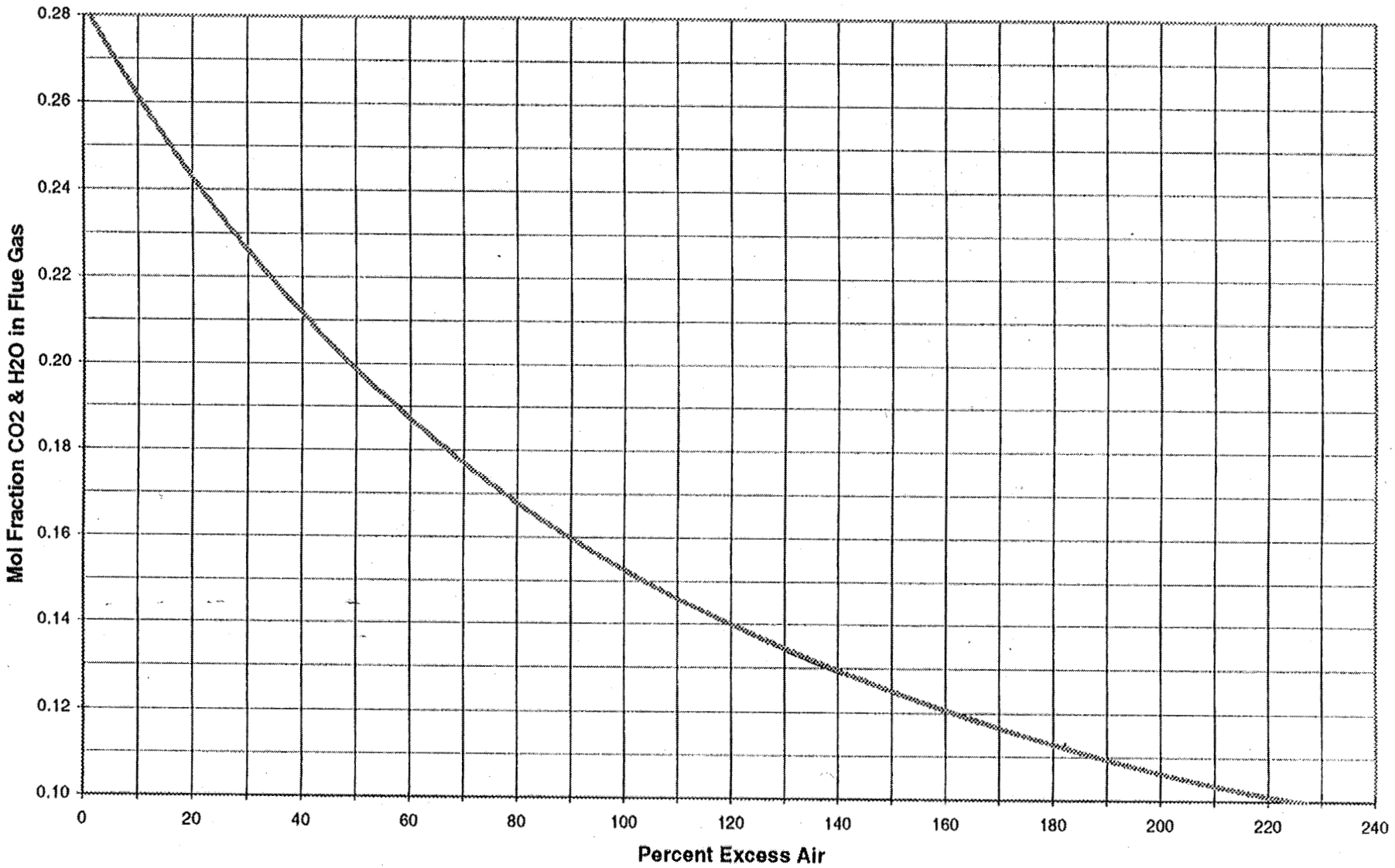
F. Firebox and Exit Gas Temperatures

The preceding charts show the rate of heat transfer between a mass of gas at a uniform temperature and a tube surface at another uniform temperature. In most actual furnaces, however, neither of these temperatures is constant. Average effective, or firebox, temperatures must be selected in order to use these figures.

The average effective gas temperature depends on the general shape of the furnace the firing conditions and the style and type of burner. Heaters with an approximately square cross-section and no large areas of refractory with direct flame impingement will have an average effective gas temperature very close to that of the exit gas. Cylindrical furnace with a high length/diameter ratio will have an average effective firebox gas temperature appreciably greater than the exit gas. The magnitude of this difference must be determined empirically from experience with similar designs.

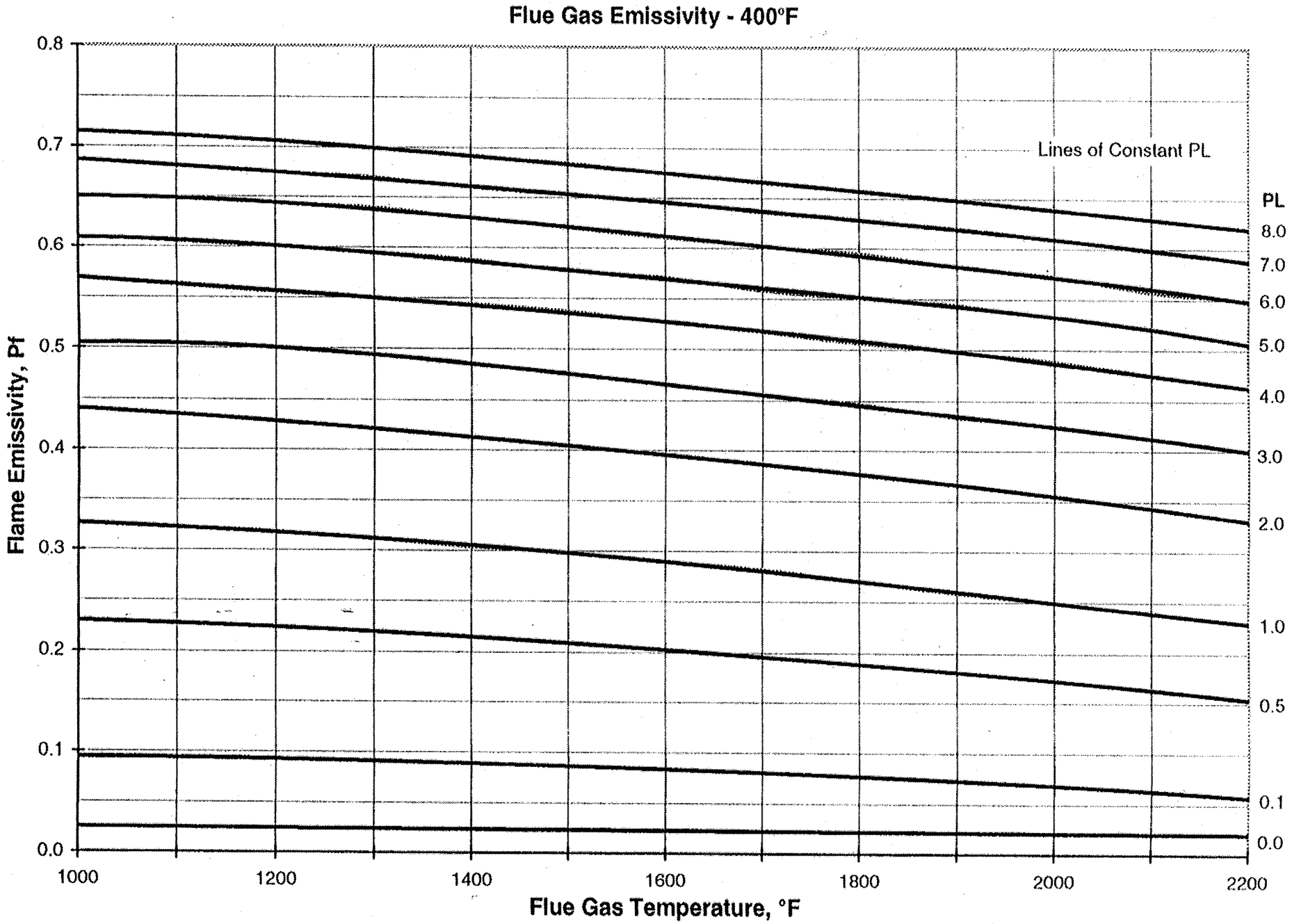

Effectiveness Factor, α
Effectiveness Factor, α
1.0
0.8
0.6
0.4
0.2
0.0
1.0
1.5
2.0
2.5
3.0
3.5
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
Ratio C-C/Tube OD
Total to Two Rows When Two are Present
Direct to First Row
Total to One Row When Only One is Present
Total to First Row When Two are Present
Total to Second Row When Two are Present
Direct to Second Row
First row is row closest to the radiating plane.
For tubes on equilaterial triangular centers with non-conducting but reradiating refractory behind tubes. May be used on tubes on equilateral rectangular centers.

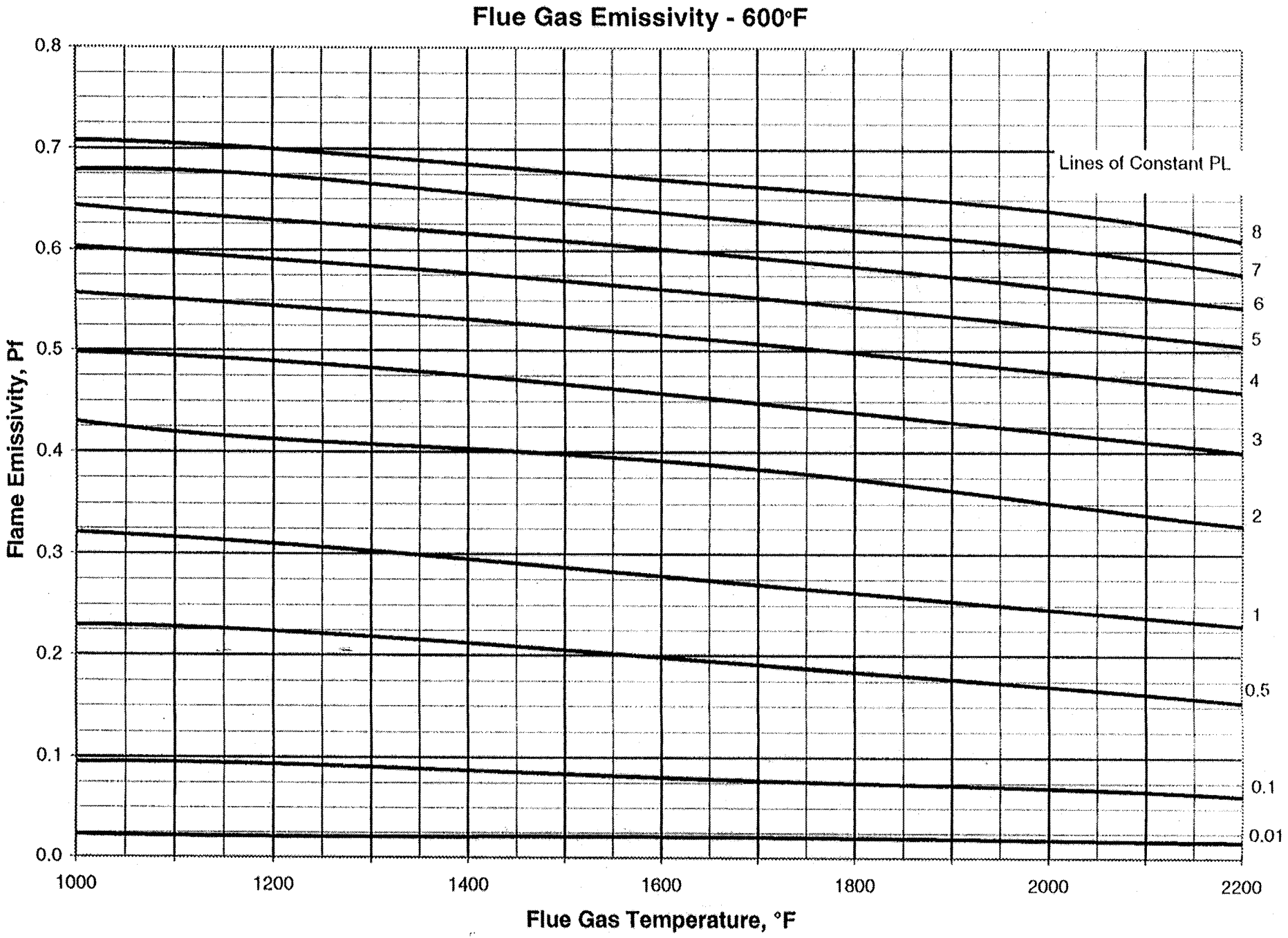
Partial Pressure of CO_2 and H_2O

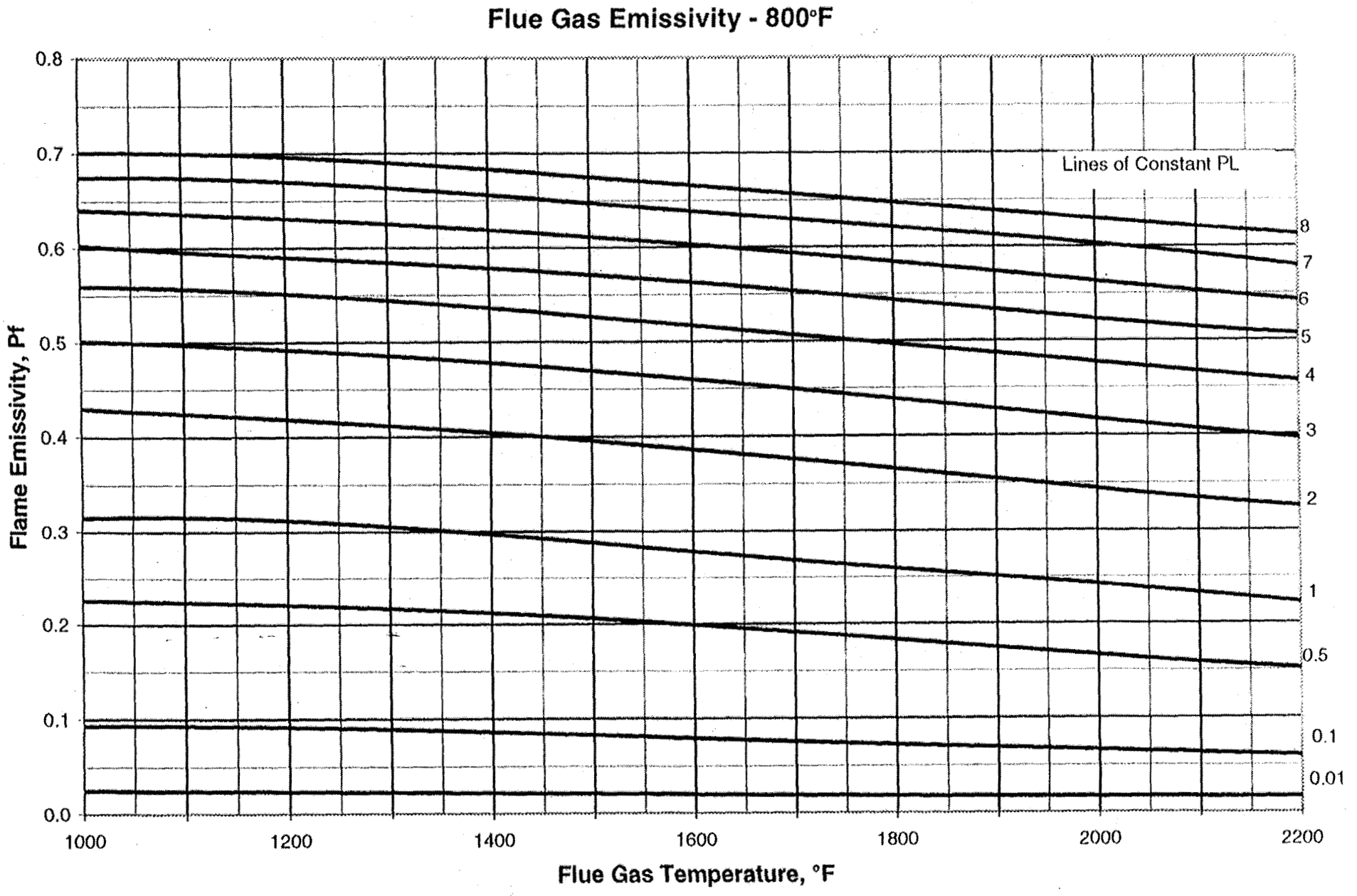


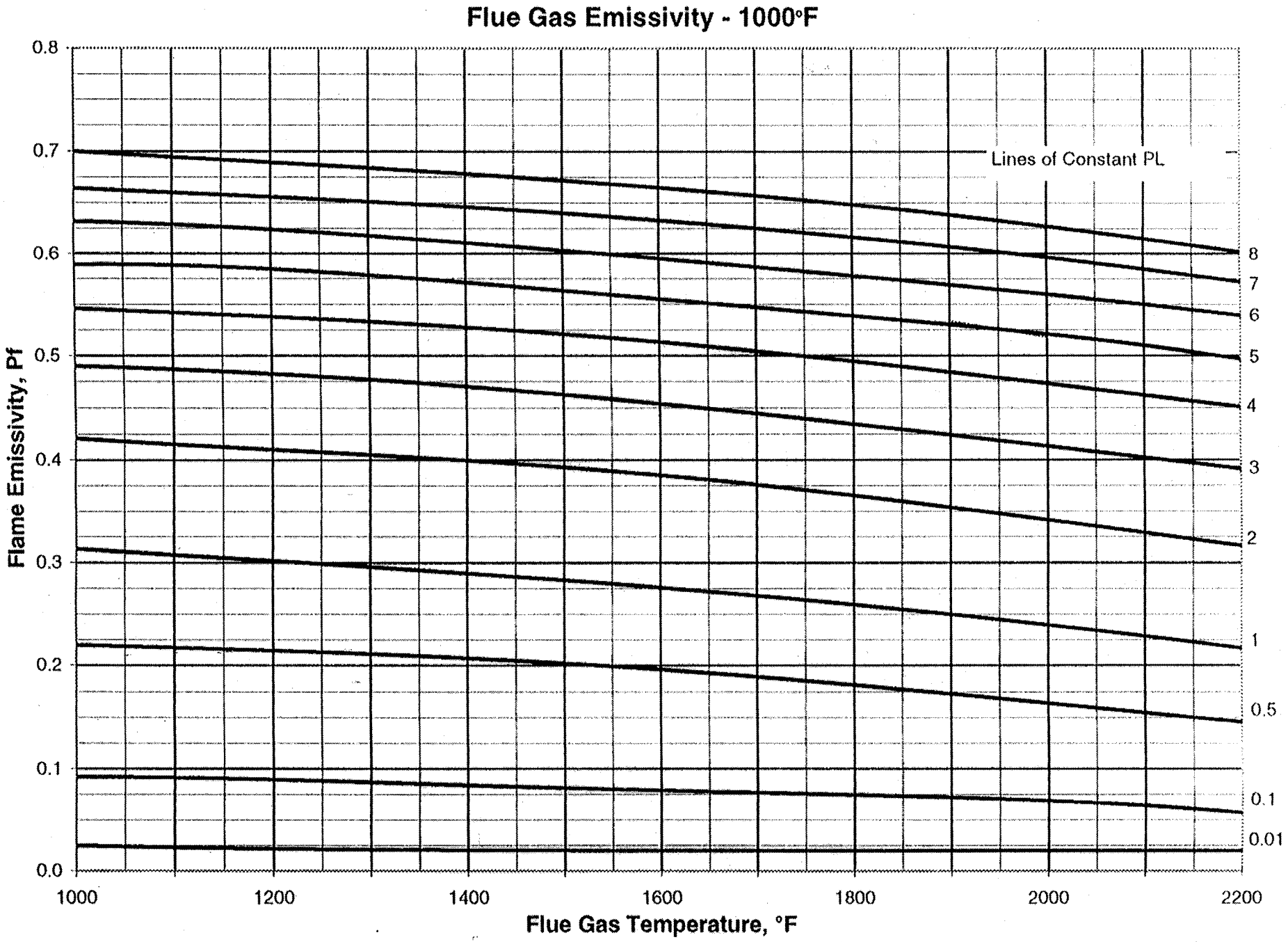
Mean Beam Length

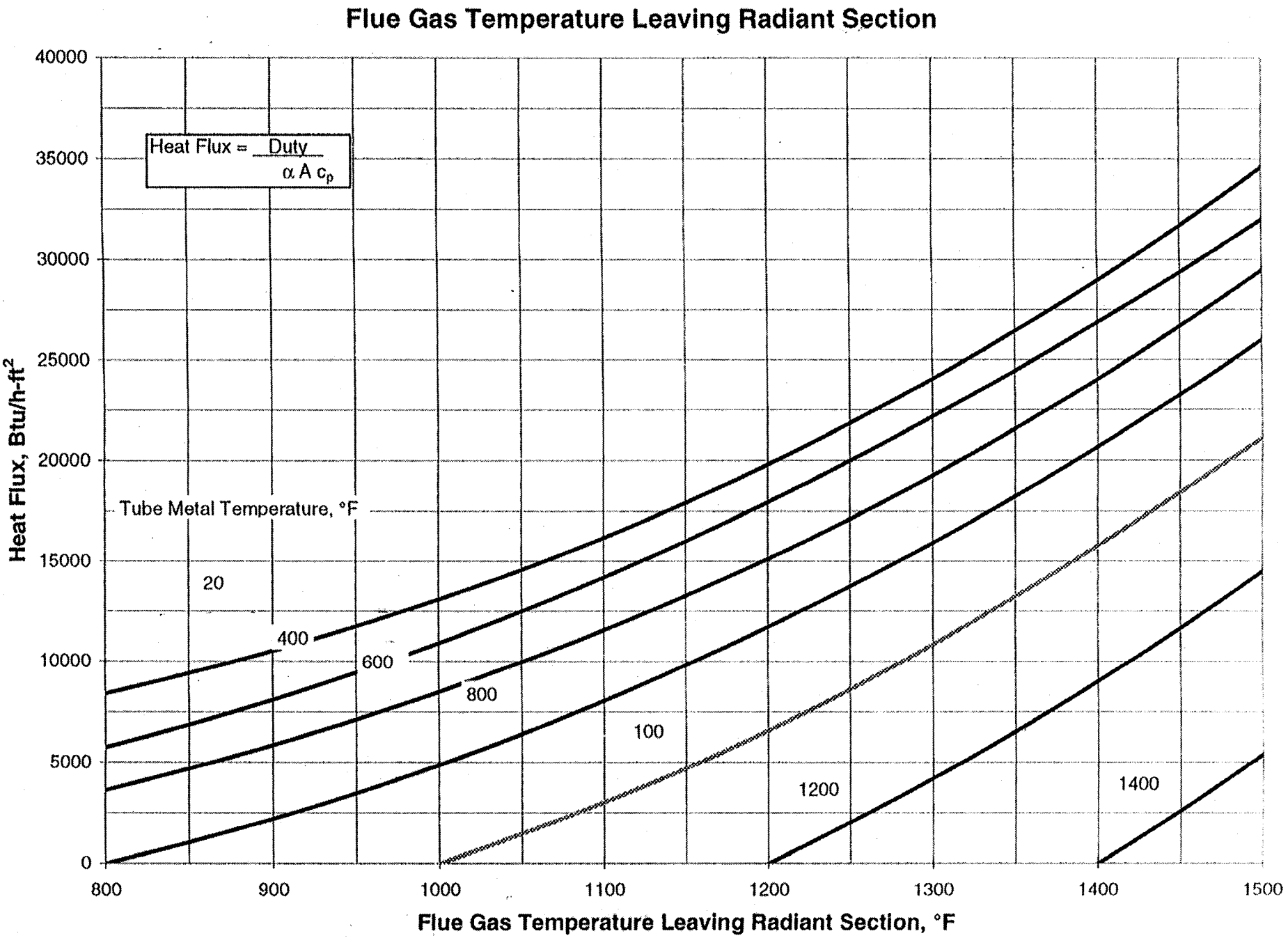
DIMENSIONAL RATIO	MEAN BEAM LENGTH (ft)
Rectangular Furnaces (length, width, height in any order)	
1 : 1 : 1 to 1 : 1 : 3	2/3 (furnace volume)$^{1/3}$
1 : 2 : 1 to 1 : 2 : 3	2/3 (furnace volume)$^{1/3}$
1 : 1 : 4 to 1 : 1 : ∞	smallest dimension
1 : 2 : 5 to 1 : 2 : 8	1.3 smallest dimension
1 : 3 : 3 to 1 : ∞ : ∞	1.8 smallest dimension
Vertical Cylindrical Updraft Furnaces (d = furnace diameter)	
d x d	d
d x d to d x ∞	d

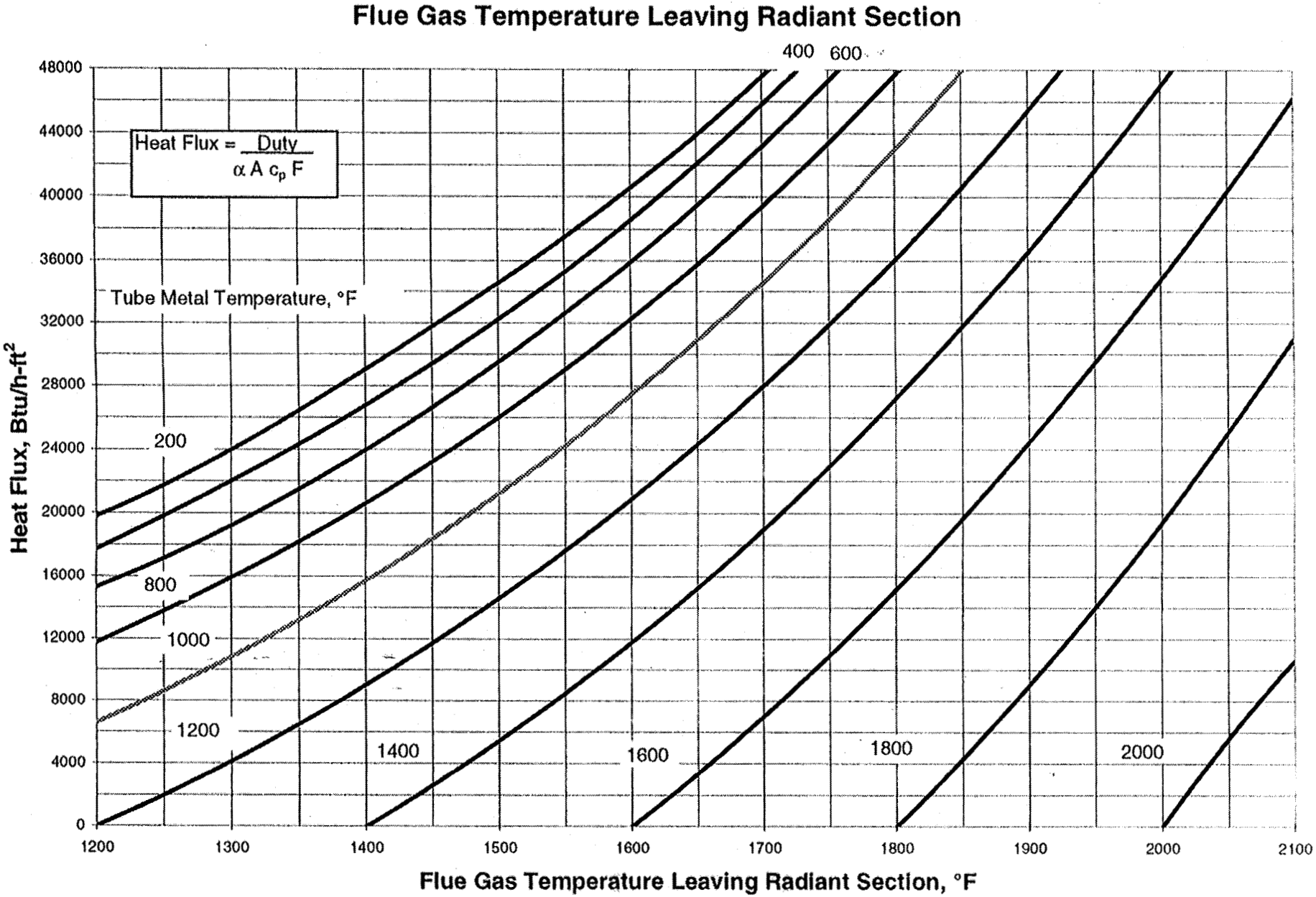

Flue Gas Emissivity - 400°F
Lines of Constant PL
PL
8.0
7.0
6.0
5.0
4.0
3.0
2.0
1.0
0.5
0.1
0.0
Flame Emissivity, Pf
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
1000
1200
1400
1600
1800
2000
2200
Flue Gas Temperature, °F


Flue Gas Emissivity - 600°F
Flame Emissivity, Pf
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
1000
1200
1400
1600
1800
2000
2200
Flue Gas Temperature, °F
Lines of Constant PL
8
7
6
5
4
3
2
1
0.5
0.1
0.01


Flue Gas Emissivity - 800°F
Flame Emissivity, Pf
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
1000
1200
1400
1600
1800
2000
2200
Flue Gas Temperature, °F
Lines of Constant PL
8
7
6
5
4
3
2
1
0.5
0.1
0.01


Flue Gas Emissivity - 1000°F
Flame Emissivity, Pf
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
1000
1200
1400
1600
1800
2000
2200
Flue Gas Temperature, °F
Lines of Constant PL
8
7
6
5
4
3
2
1
0.5
0.1
0.01


Flue Gas Temperature Leaving Radiant Section
Heat Flux = Duty / α A c_p
Tube Metal Temperature, °F
20
400
600
800
100
1200
1400
Heat Flux, Btu/h-ft²
40000
35000
30000
25000
20000
15000
10000
5000
0
800
900
1000
1100
1200
1300
1400
1500
Flue Gas Temperature Leaving Radiant Section, °F


Flue Gas Temperature Leaving Radiant Section
Heat Flux = Duty / α A c_p F
Tube Metal Temperature, °F
400
600
200
800
1000
1200
1400
1600
1800
2000
Heat Flux, Btu/h-ft²
48000
44000
40000
36000
32000
28000
24000
20000
16000
12000
8000
4000
0
1200
1300
1400
1500
1600
1700
1800
1900
2000
2100
Flue Gas Temperature Leaving Radiant Section, °F

Training Services

Convection Section

Section Outline

- Convection Heat Balance
- Monrad Equation
- Cold End Approach Charts
- ESCOA Method

Convection Heat Balance

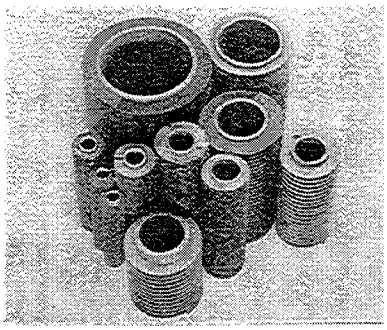
- Process Absorbed= BWT-Stack
- Q= U*A*LMTD

Convection Section

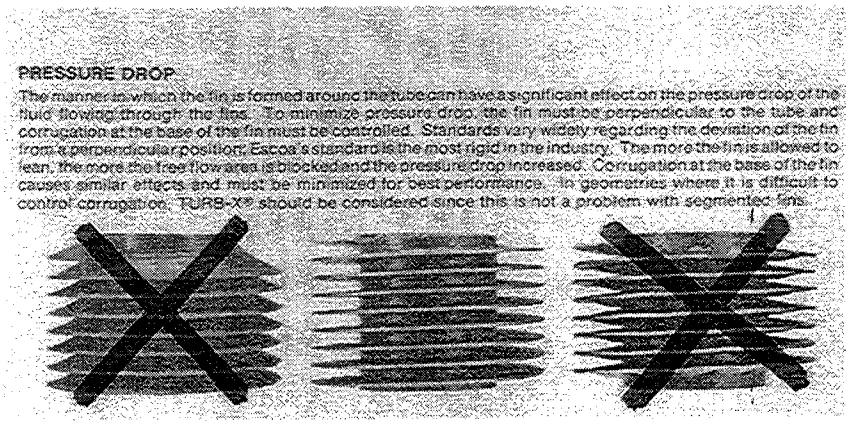
- Three Bare Rows (Shock/Shield Tubes)
- Extended Surface Rows
  - Solid fins
  - Segmental fins
  - Studs

Solid fins

- **Stronger Construction**
- **Less prone to damage than segmented fins**
- **Higher hoop stresses can cause fins to roll**
- **10 times more area than bare tube**

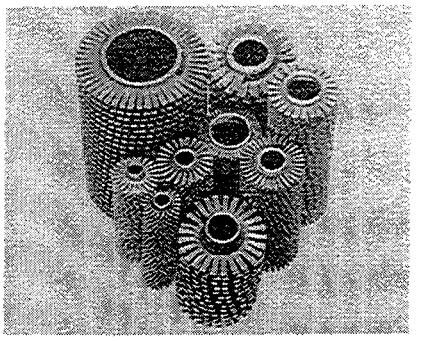


Hoop Stress on Solid fins



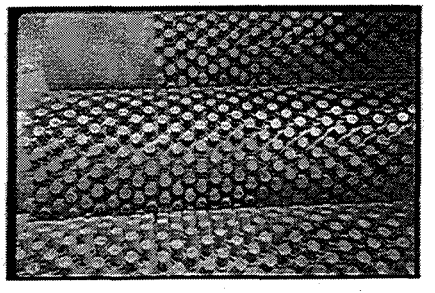
Segmented fins

- Stronger Construction
- More prone to damage than solid fins
- Segments relieve hoop stressesless bending of fin
- 7% less expensive than solid fin
- Used in gas firing



Studs

- Used for oil firing
- Clean better than fins
- Very strong construction
- Studs are 1" long by 0.5" diameter
- Cost 7 % more than solid fins



Convection Section Layout
Typical Fin Density - Gas Firing

Nominal Tube Size	Fin Length (inches)	Fin/inch	Fin Thickness (inches)
3	0.50	5	0.05
4	0.75	5	0.05
5	1.0	5	0.05
6	1.0	5	0.05

Convection Section Layout
Typical Fin Density - Oil Firing

Nominal Tube Size	Fin Length (inches)	Fin/inch	Fin Thickness (inches)
3	0.50	3	0.10
4	0.75	3	0.10
5	0.75	3	0.10
6	0.75	3	0.10

Convection Section Calculations

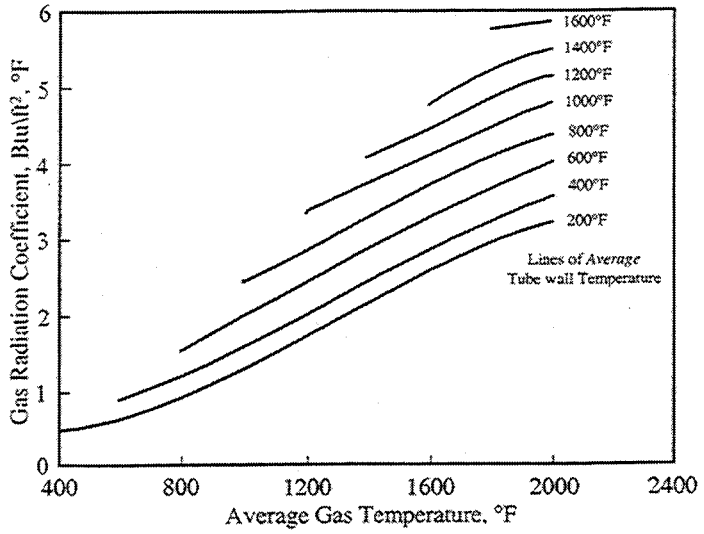
- Convection Coefficient
- Radiation Coefficient
- Wall Refractory Effect
- Fouling Coefficient
- Inside Coefficient

Monrad Equation
Outside Heat Transfer Coefficient

$$h_c = \frac{2.14(T_f)^{0.28}(G)^{0.6}}{d^{0.4}}$$

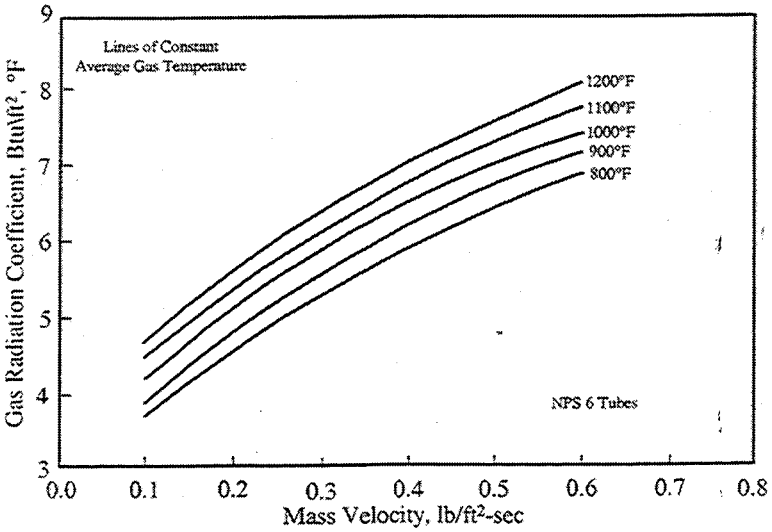
where: h_c = Convective coefficient, Btu/ft^2-°F
T_f = Outside film temperature, °R
G = Mass velocity of the flue gas, lb/ft^2-sec
d = Outside diameter of tube, inch

Gas Radiation Coefficients



FH 2004/4-13

Overall Heat Transfer Coefficient
Convection Section - NPS 6 Tubes



FH 2004/4-14

Effect of Flue Gas Mass Velocity

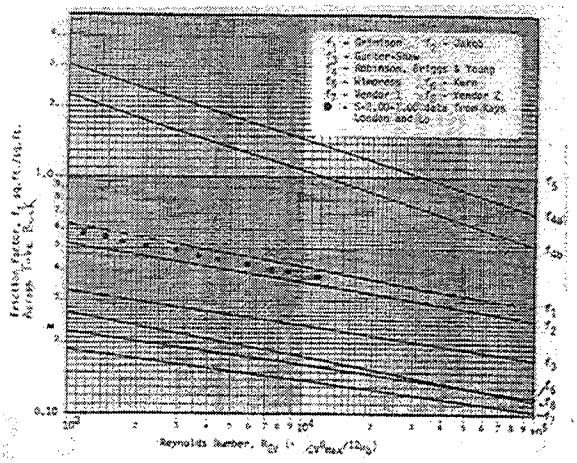
Change of mass velocity from:	**0.30 to 0.44 (45% increase)**
Heat Transfer Coefficient	**Increases 25%**
Flue Gas Pressure Drop	**Increases 210%**

Stack Height is twice as high

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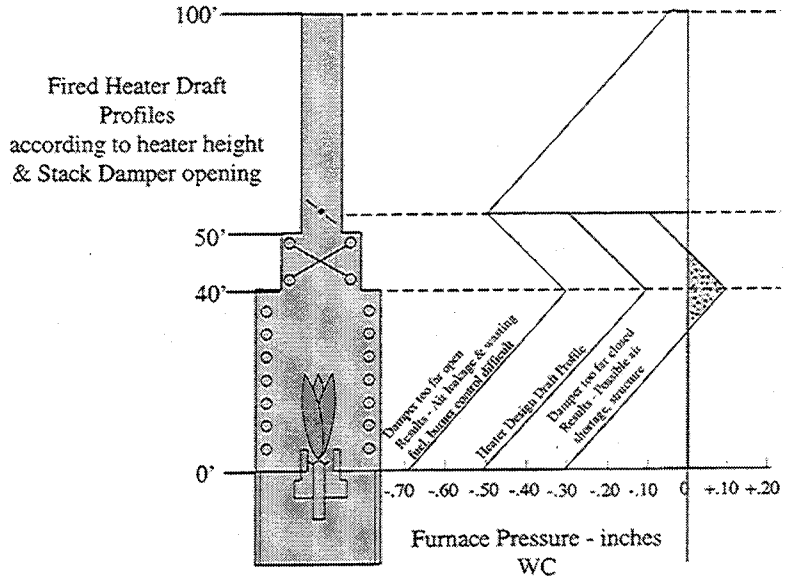
Flue Gas Pressure Drop

- **API Paper 1984**
- **Wide variation in published methods**
- **Most heater vendors use too low friction factor**



FH 2004/4-16

Draft Profile



FH 2004/4-17

Convection Heat Transfer

- **Governed by flue gas flow**
- **Process flow does not affect heat pickup**
- **This concept is missed by most process engineers**

$$U_o = \frac{hi \times ho}{hi + ho} = \frac{300 \times 6}{300 + 6} = 5.88 \ \frac{Btu}{hr - ft^2 - {}^\circ F}$$

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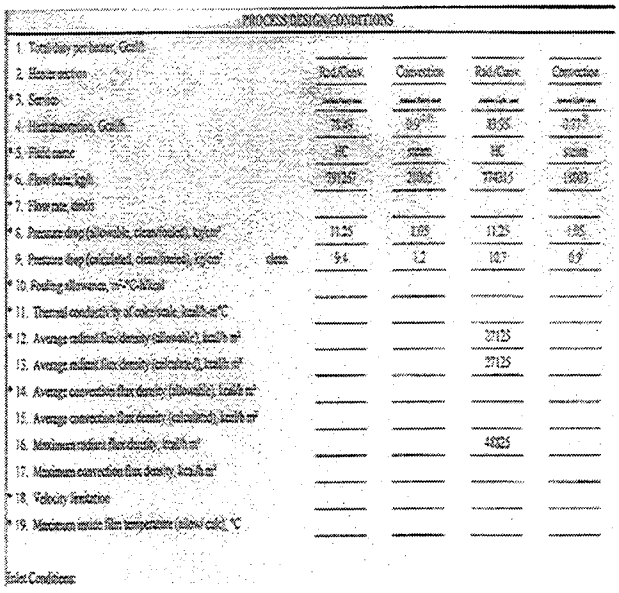
ESCOA Method

- **Engineering Manual and Calculation program available for free downloading from the Internet at:**

  http://www.fintubetech.com

- **Predicts heat transfer across entire bank of tubes**
- **Predicts accurate heat transfer coefficients**
- **Does not do row by row calculations**
- **Under predicts flue gas pressure drops**
- **Use UOP Charts , not the Escoa Method**

API Fired Heater Data Sheets



PROCESS DESIGN CONDITIONS

	Rad./Conv.	Convection	Rad./Conv.	Convection
1. Total duty per heater, Gcal/h				
2. Heater section				
*3. Service				
4. Heat absorption, Gcal/h	78.45	39.225	38.935	0.217
*5. Fluid name	HC	steam	HC	steam
*6. Flow Rate, kg/h	297267	20000	774815	10000
*7. Flow rate, kmol/h				
*8. Pressure drop (allowable, clean/fouled), kg/cm²	11.25	1.05	11.25	1.05
9. Pressure drop (calculated, clean/fouled), kg/cm² clean	9.4	1.2	10.7	0.9
*10. Fouling allowance, m²·°C/kcal				
*11. Thermal conductivity of rad. tube, kcal/h·m·°C				
*12. Average radiant flux density (allowable), kcal/h·m²			27125	
13. Average radiant flux density (calculated), kcal/h·m²			27125	
*14. Average convection flux density (allowable), kcal/h·m²				
15. Average convection flux density (calculated), kcal/h·m²				
16. Maximum radiant flux density, kcal/h·m²			40825	
17. Maximum convection flux density, kcal/h·m²				
*18. Velocity limitation				
*19. Maximum metal film temperature (skin calc), °C				

Inlet Conditions

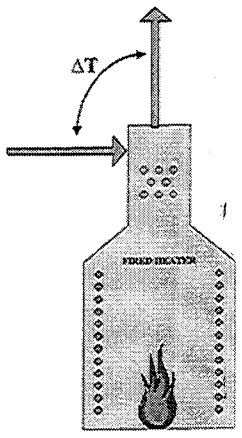
API Fired Heater Data Sheets

Coil Configuration:					
17. Tubes (vertical or horizontal)	Vertical	Horizontal ———			——→
18. No. of flow passes/tubes per row	2 x 5 \ -	10 \ 10		3 \ 5	
19. Effective tube length, m	18453	19236			
20. Bare tubes , number	2 x 110	20			
21. Bare tubes, total exposed surface, m²	2 x 1070	304.7			
22. Extended surface tubes, number			100	18	
23. Extended surface tubes, total exposed surface, m²			3867	581	
24. Tube spacing, center-to-center (staggered) (inline), mm	304.8	304.8 \ 355		381	
Tubes:					
* 25. Material (ASTM specification and grade)	←———	A335 Gr P9	———	→	A 106 B
26. Outside diameter, mm	168.3	168.3	168.3	141.3	
27. Wall thickness (minimum)(average), mm	7.11	7.11	7.11	6.55	
28. Overall tube length, m (straight)	17.952	19.710	19.710	19.710	
29. No. of intermediate welds	1	1	1	1	
Description of Extended Surface:					
30. Type (studs, segmented fins, or solid fins)			Studs	Studs	
31. Material			CS	CS	
32. Dimensions (height x diameter/thickness), mm			25 x 12.5	20 x 12.5	
33. Spacing Number of studs per plane			24	20	
34. Maximum tip temperature, °C			488	297	
35. Extension ratio			3.81	3.39	

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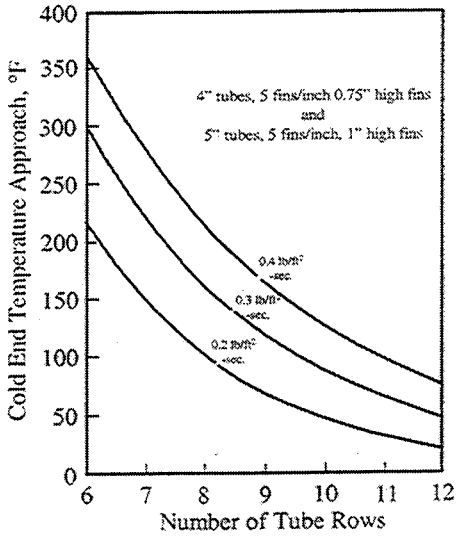
Cold End Approach

- **Cold End Approach is flue gas out- process inlet**
- **Bare tubes -250°F(139°C)**
- **Extended surface-75°F(42°C)**
- **Controls energy efficiency**
- **Plot cold end approach versus time to track fouling**



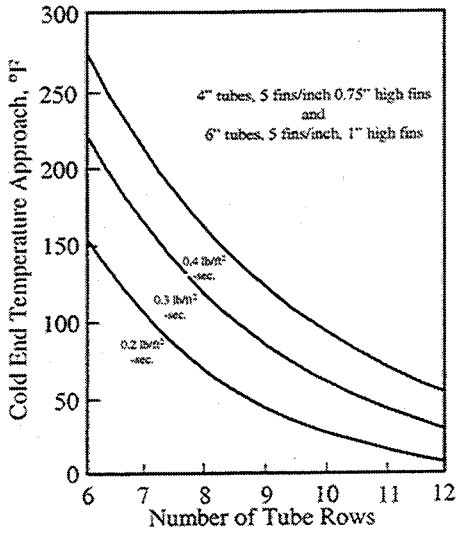
FH 2004/4-22

Cold End Approach
450 ° F Process Inlet - Fuel Gas



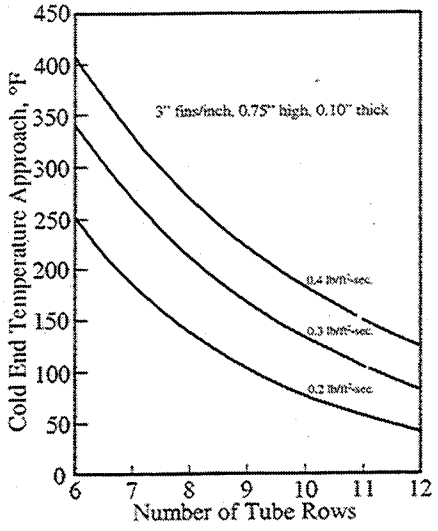
FH 2004/4-23

Cold End Approach
650 ° F Process Inlet - Fuel Gas



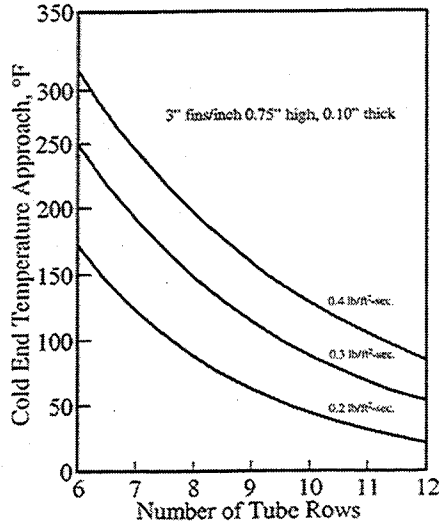
FH 2004/4-24

Cold End Approach
450 °F Process Inlet - Fuel Oil - NPS 6



FH 2004/4-25

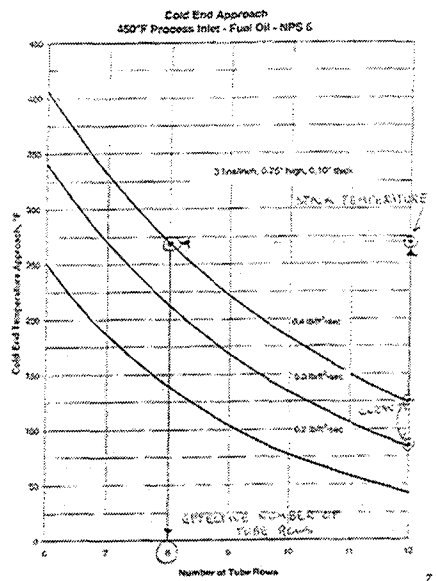
Cold End Approach
650 ° F Process Inlet - Fuel Oil - NPS 6



FH 2004/4-26

Convection Section Performance

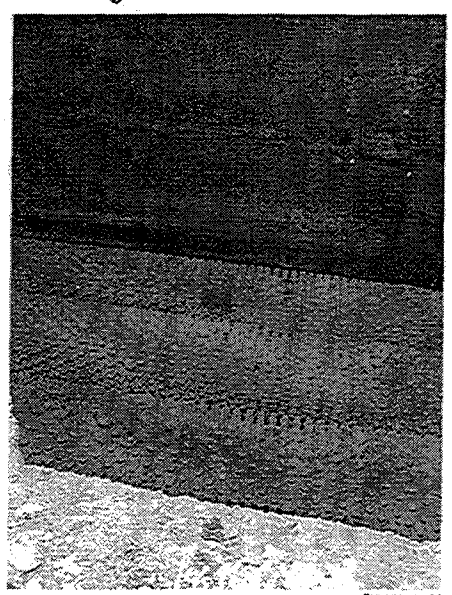
- Gas fired heaters can foul from ceramic fiber dust(most gas fired heaters have no fouling in the convection section
- Oil fired heater foul from soot and unburned fuel oil(30-50% surface
- Plot cold end approach versus time to track fouling



7

Convection Section Performance

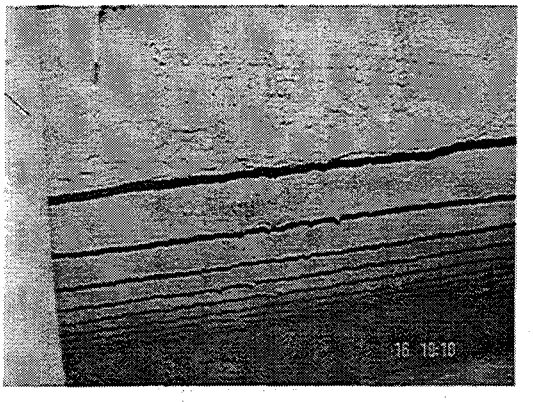
- Crude heater operating at 800°F
- Grey areas are clean tubes
- Oil fired heater, fouling from soot is brown colored and easily blown off tubes
- Soda Ash washing is recommended during turnarounds



FH 2004/4-28

Convection Section Performance

- Unburned fuel oil fouling
- CTP online cleaning with ammonium nitrate(Neutramex)
- CTP offline cleaning with phosphoric acid
- CTP(phbarre@ctp-environnement.com) 33-139-19-15-50



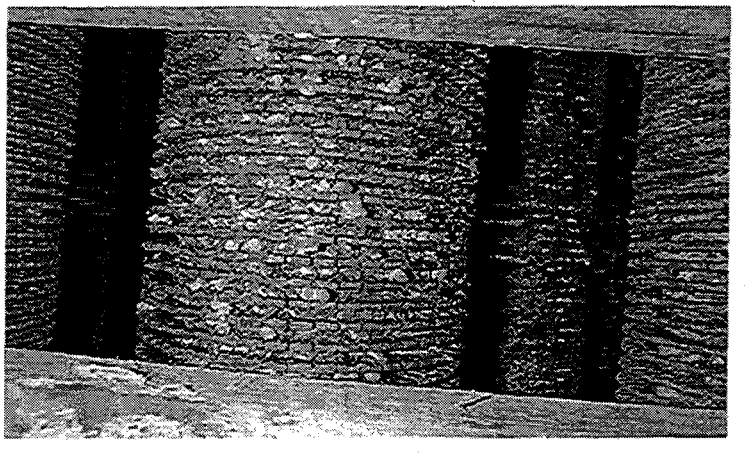
FH 2004/4-29

Fouled Convection Tubes-Gas Firing

Sticky sulfur deposits and refractory on tubes

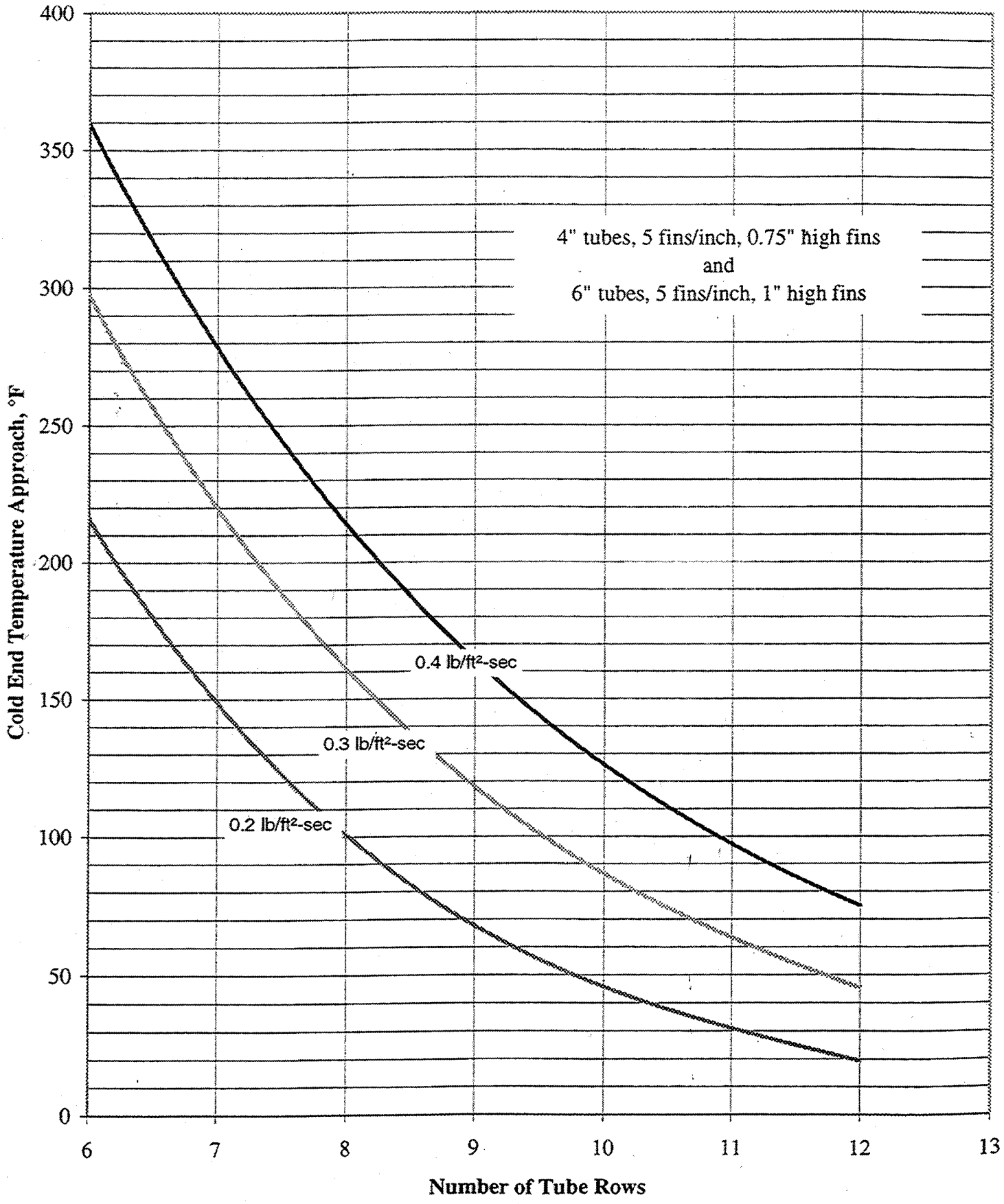
Convection bank need soda washing

Can refractory withstand soda wash?

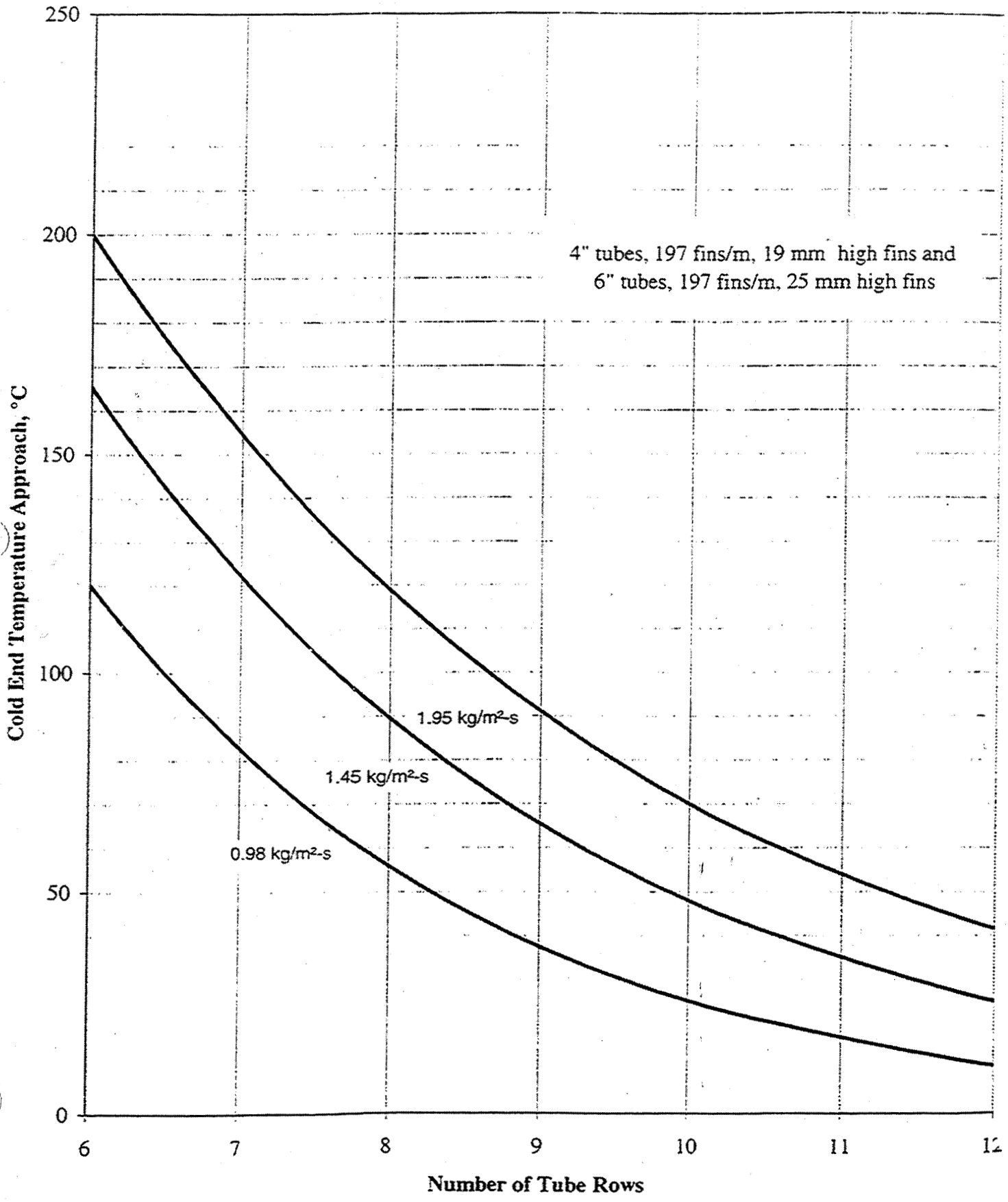


FH 2004/4-30

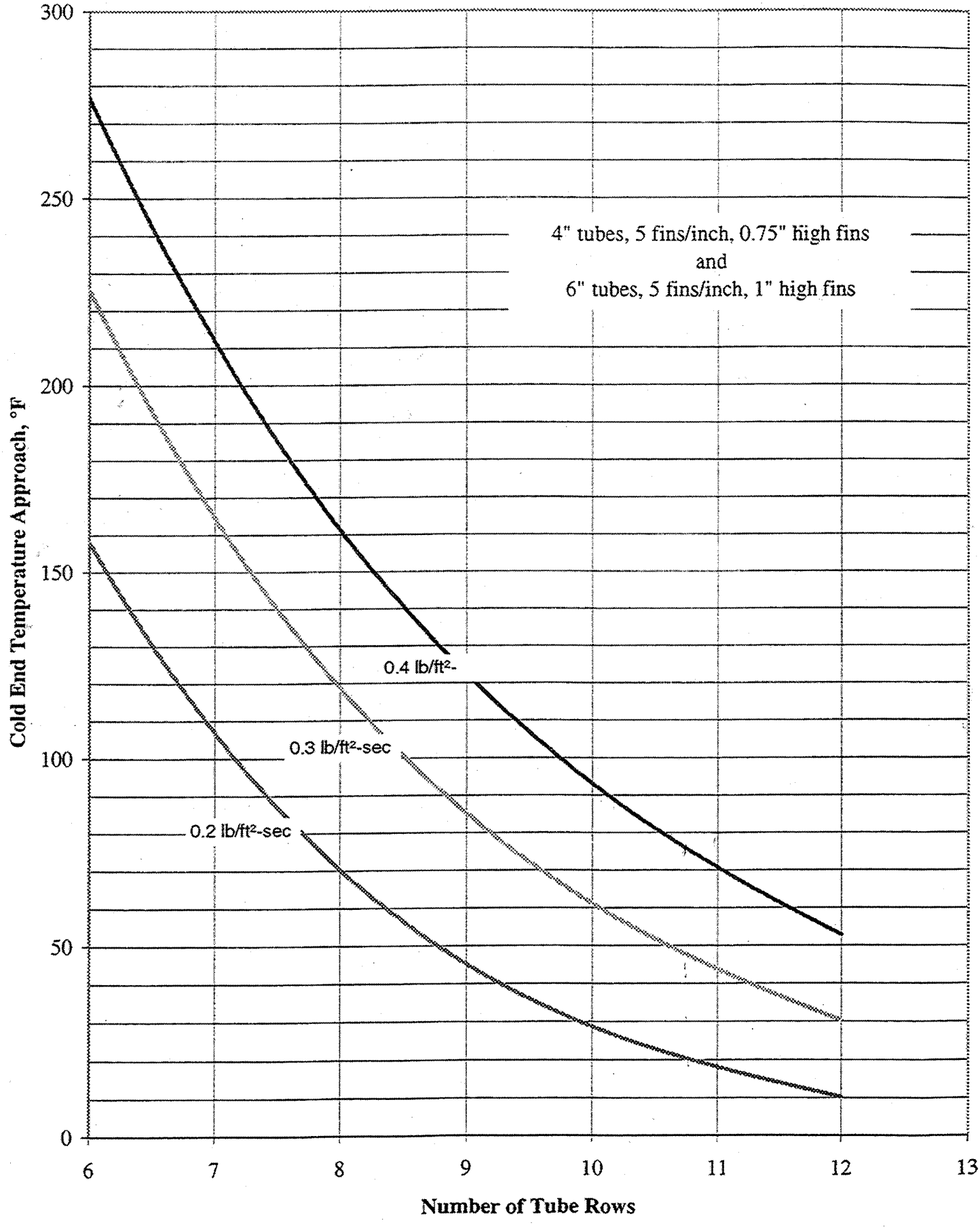
Effect of Cold End Temp. Approach On the Number of Tube Rows (Gas Fired, 450°F Process Inlet)



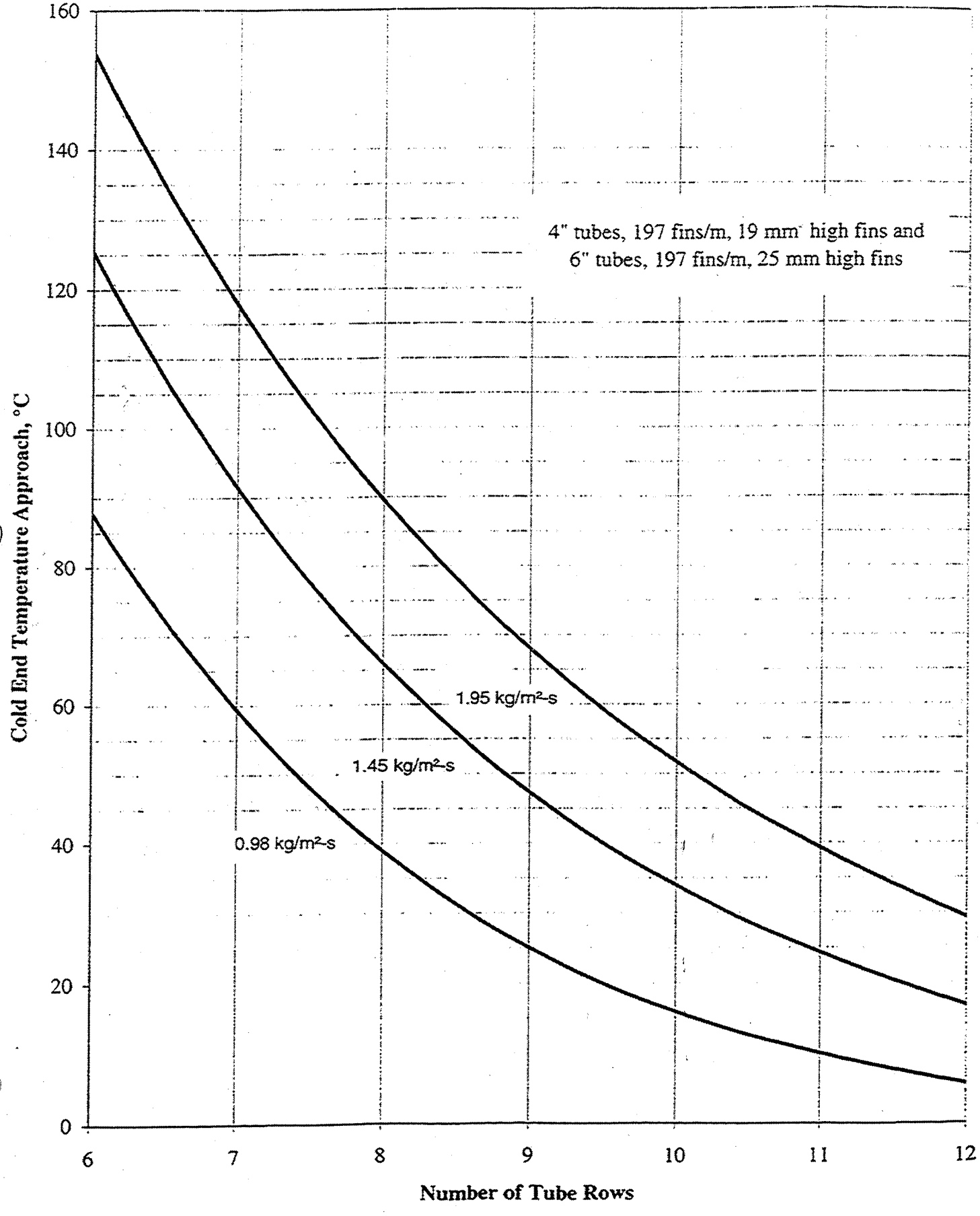
Effect of Cold End Temp. Approach On the Number of Tube Rows (Gas Fired, 232°C Process Inlet)



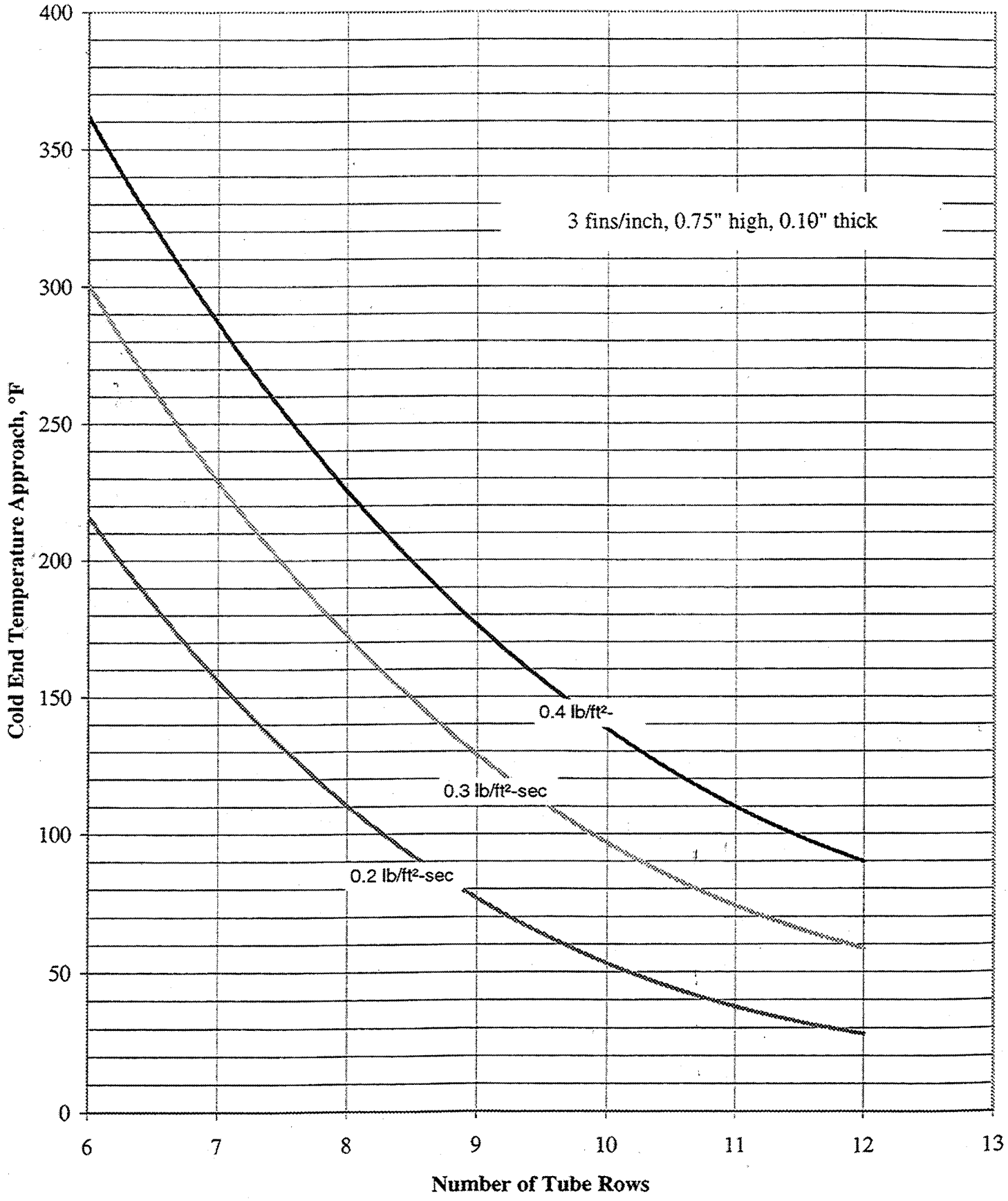
Effect of Cold End Temp. Approach On the Number of Tube Rows (Gas Fired, 650°F Process Inlet)



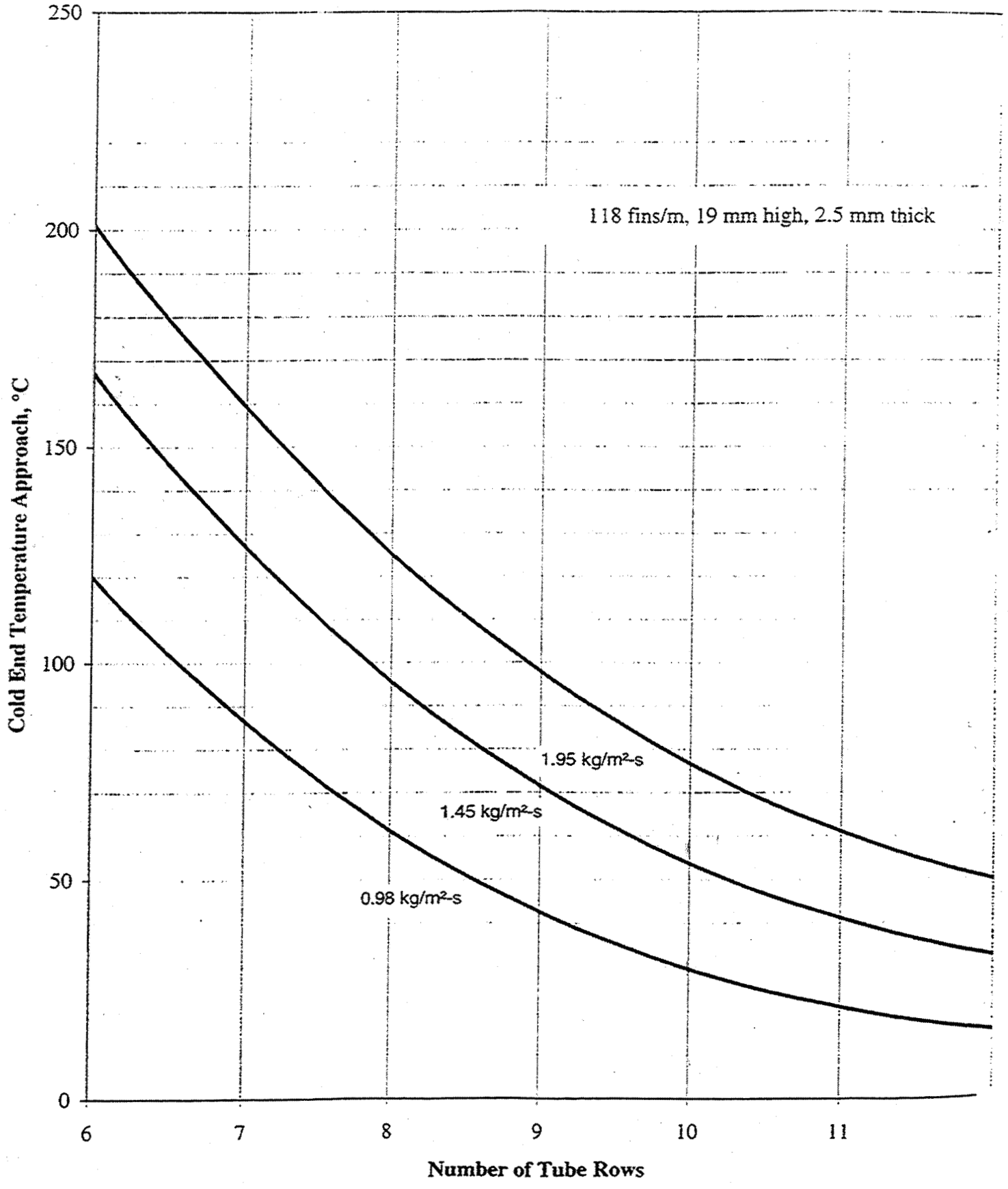
Effect of Cold End Temp. Approach On the Number of Tube Rows (Gas Fired, 343°C Process Inlet)



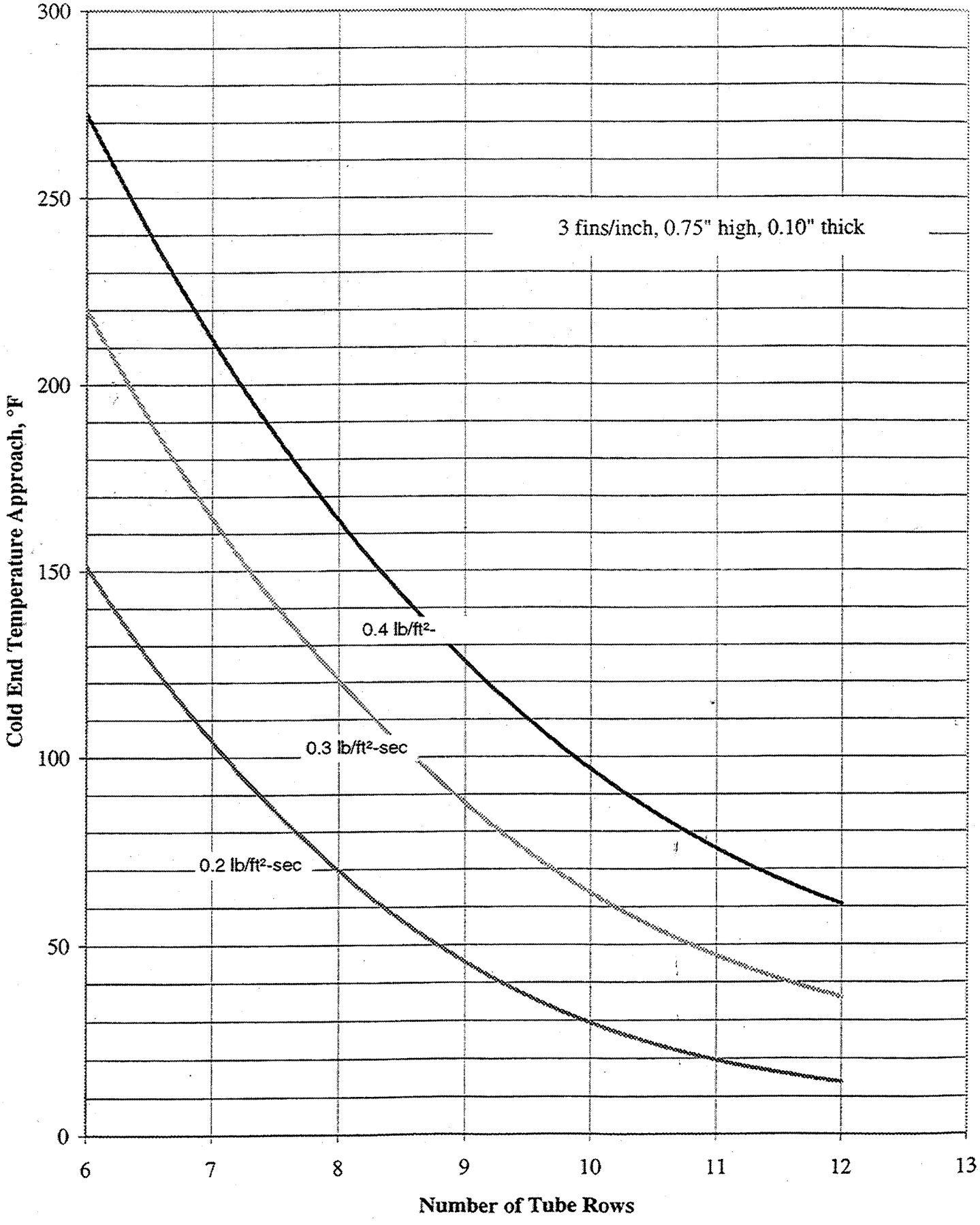
Effect of Cold End Temp. Approach On the Number of Tube Rows (Oil Fired, 450°F Process Inlet, 4" Tubes)



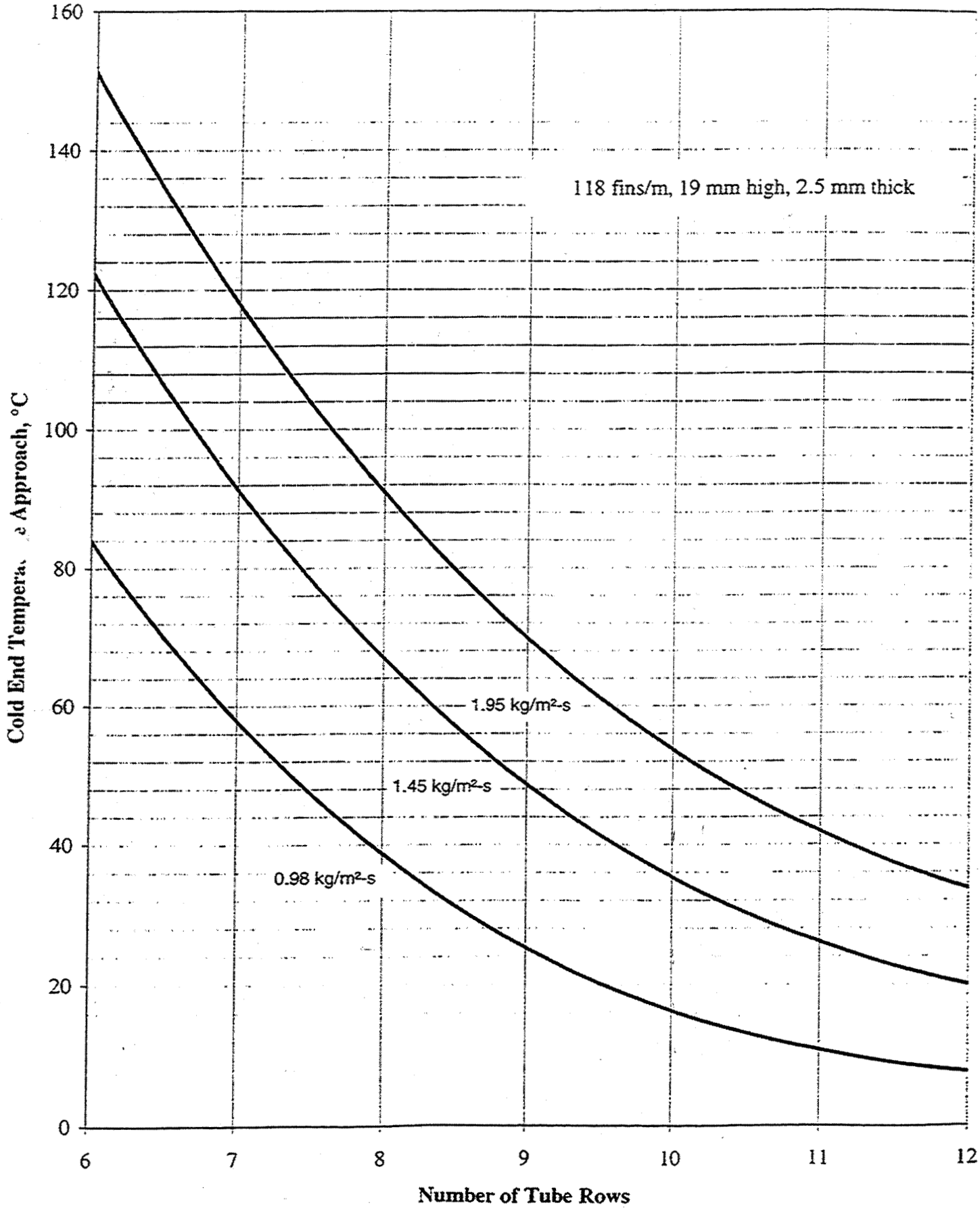
Effect of Cold End Temp. Approach On the Number of Tube Rows (Oil Fired, 232°C Process Inlet, 4" Tubes)



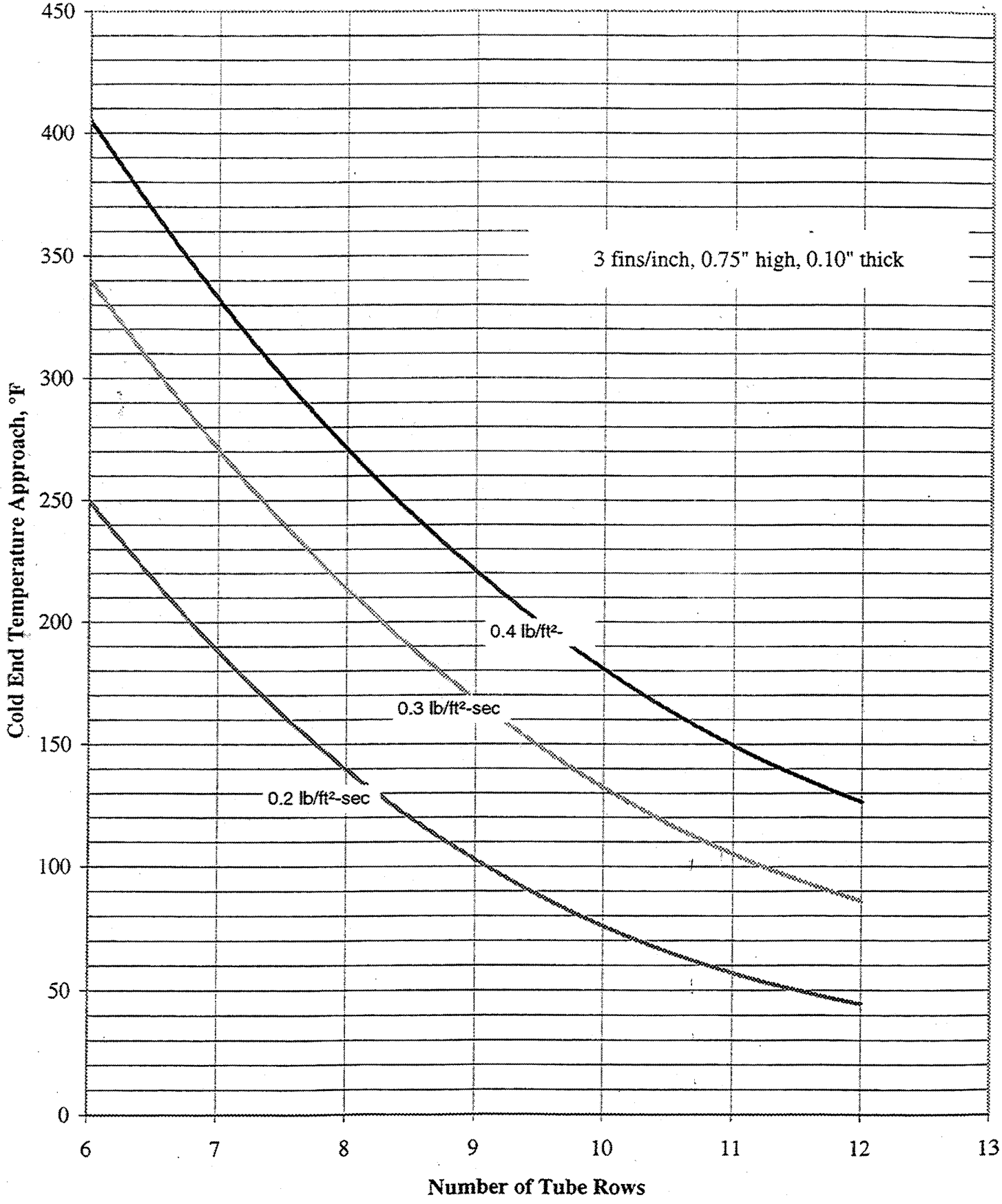
Effect of Cold End Temp. Approach On the Number of Tube Rows (Oil Fired, 650°F Process Inlet, 4" Tubes)



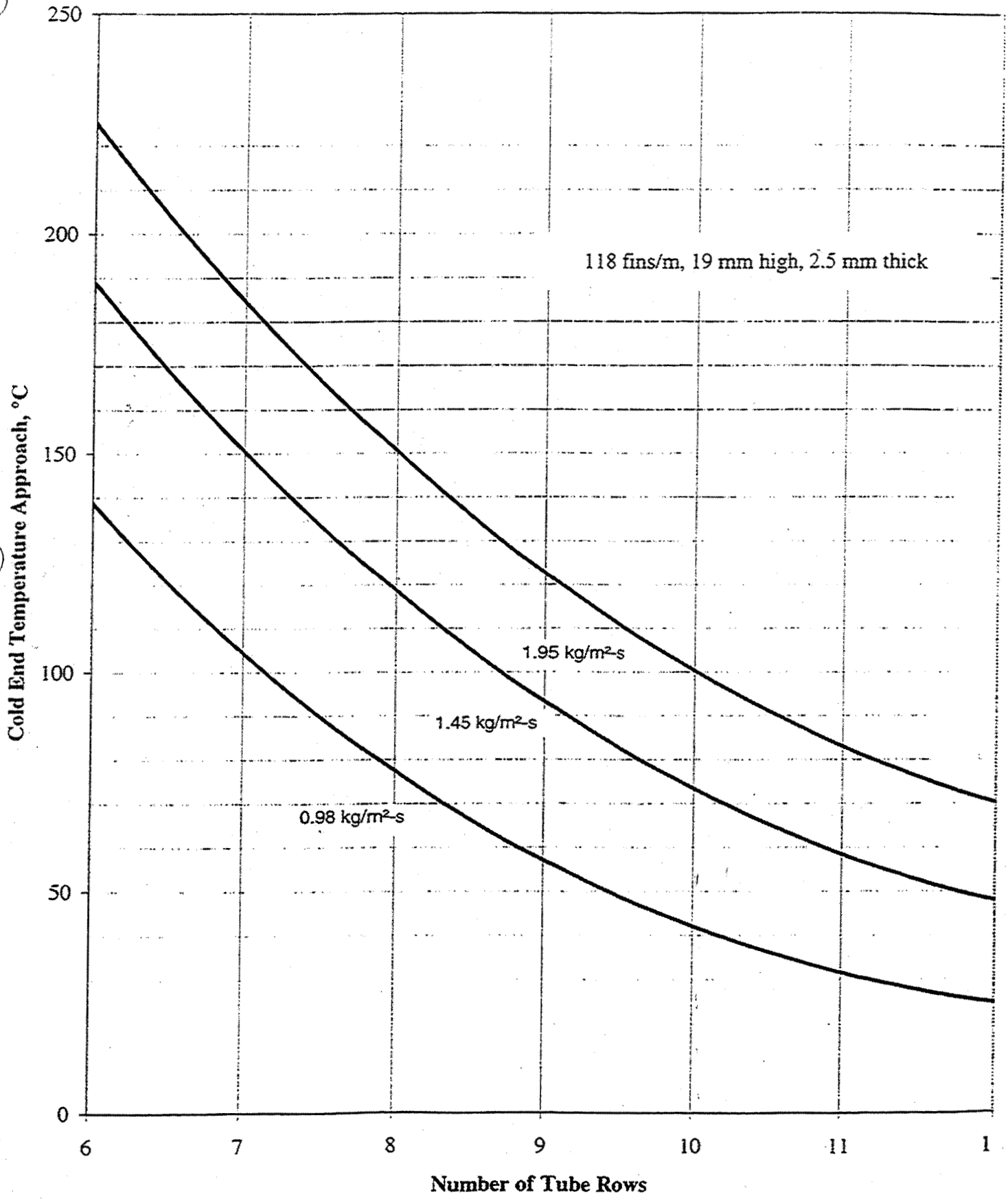
Effect of Cold End Temp. Approach On the Number of Tube Rows (Oil Fired, 343°C Process Inlet, 4" Tubes)



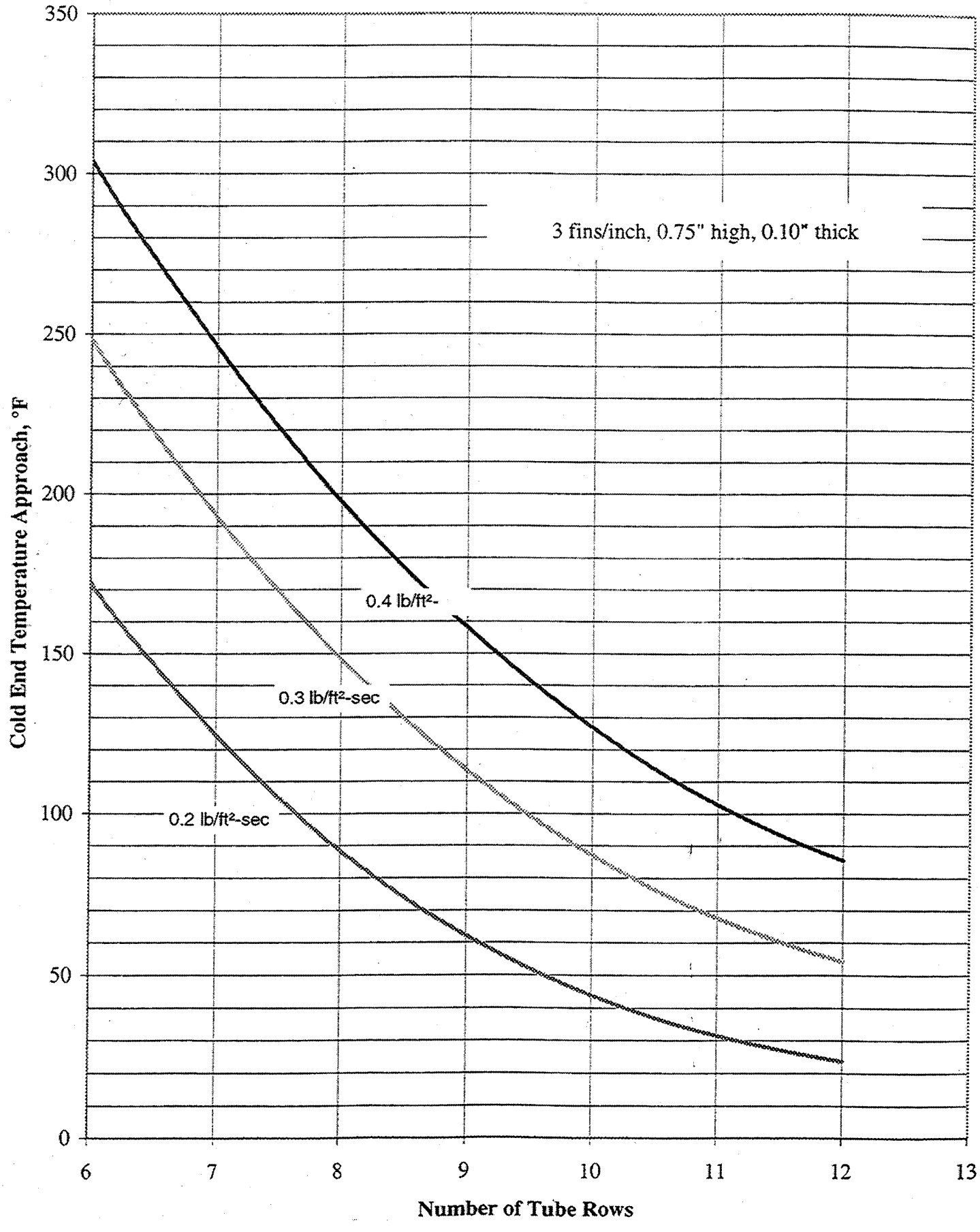
Effect of Cold End Temp. Approach On the Number of Tube Rows (Oil Fired, 450°F Process Inlet, 6" Tubes)



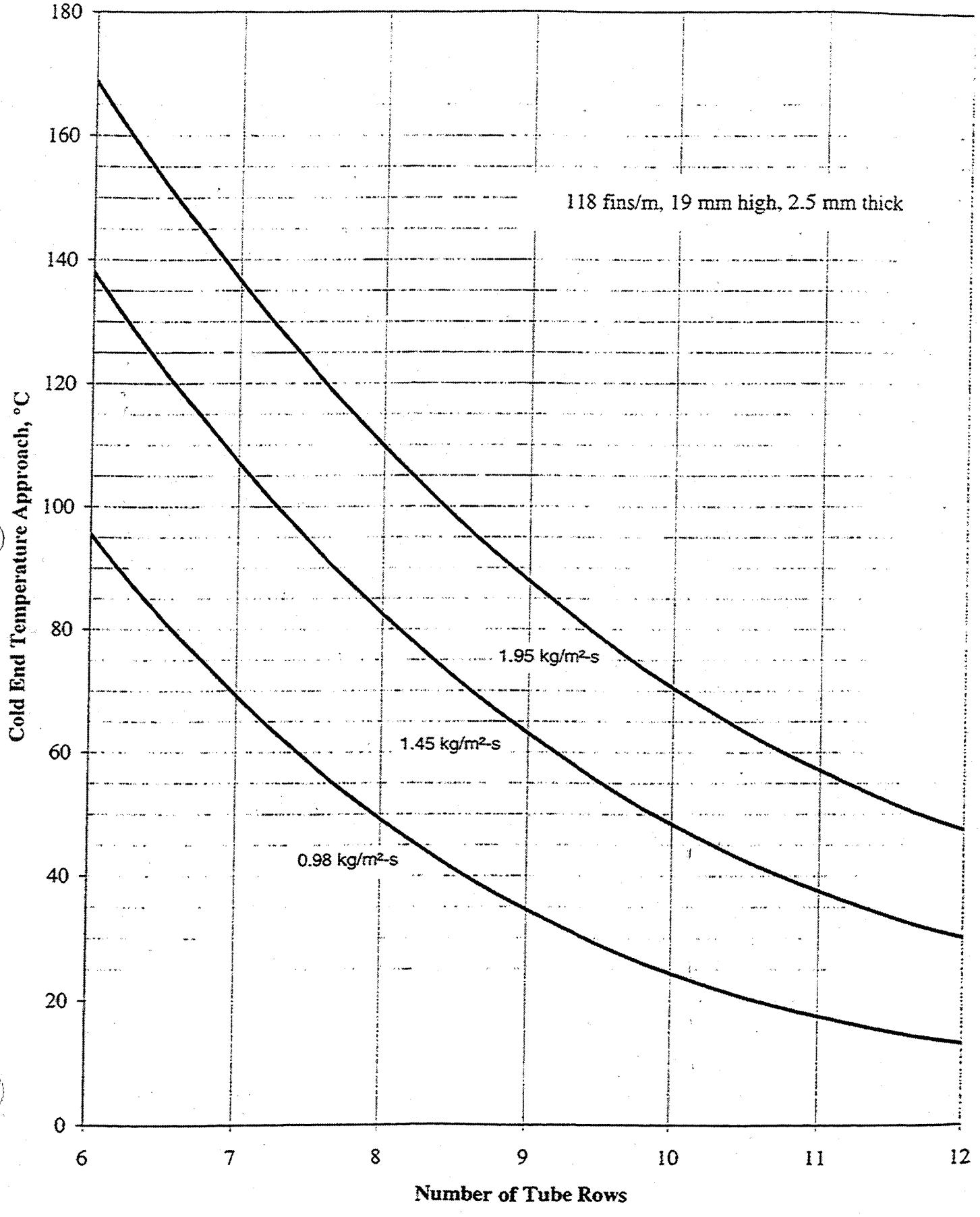
Effect of Cold End Temp. Approach On the Number of Tube Rows (Oil Fired, 232°C Process Inlet, 6" Tubes)



Effect of Cold End Temp. Approach On the Number of Tube Rows (Oil Fired, 650°F Process Inlet, 6" Tubes)



Effect of Cold End Temp. Approach On the Number of Tube Rows (Oil Fired, 343°C Process Inlet, 6" Tubes)



Convection Section Layout

Gas Fired, 450°F Process Inlet 5 fins/inch, 0.75"high(4"),1.0"high(6"),0.05" thick				
Number Tube Rows	**4" Tube Cold End Approach, °F**	**Flue Gas ΔP, Inch H_2O**	**6" Tube Cold End Approach, °F**	**Flue Gas ΔP, Inch H_2O**
		0.2 lb/sec/ft² Mass Velocity		
6	219	0.032	214	0.032
7	149	0.040	147	0.041
8	102	0.046	100	0.047
9	69	0.053	68	0.055
10	44	0.059	47	0.061
11	29	0.065	32	0.068
12	18	0.071	21	0.075
		0.3 lb/sec/ft² Mass Velocity		
6	298	0.072	298	0.073
7	218	0.089	220	0.092
8	160	0.104	163	0.108
9	117	0.119	121	0.123
10	83	0.132	89	0.139
11	60	0.147	66	0.154
12	43	0.159	48	0.167
		0.4 lb/sec/ft² Mass Velocity		
6	357	0.127	363	0.130
7	274	0.157	281	0.163
8	210	0.183	218	0.192
9	161	0.211	169	0.221
10	120	0.235	131	0.248
11	93	0.260	101	0.275
12	71	0.284	79	0.300

Convection Section Layout

Gas Fired, 650°F Process Inlet 5 fins/inch, 0.75"high(4"),1.0"high(6"),0.05" thick				
Number Tube Rows	4" Tube Cold End Approach, °F	Flue Gas ΔP, Inch H_2O	6" Tube Cold End Approach, °F	Flue Gas ΔP, Inch H_2O
		0.2 lb/sec/ft² Mass Velocity		
6	162	0.034	155	0.035
7	108	0.043	104	0.044
8	72	0.051	69	0.052
9	47	0.059	45	0.061
10	28	0.066	29	0.069
11	17	0.073	18	0.077
12	10	0.081	11	0.085
		0.3 lb/sec/ft² Mass Velocity		
6	227	0.076	225	0.078
7	164	0.095	164	0.098
8	118	0.113	119	0.117
9	86	0.131	87	0.135
10	58	0.147	63	0.154
11	42	0.163	45	0.171
12	29	0.179	32	0.189
		0.4 lb/sec/ft² Mass Velocity		
6	276	0.134	278	0.137
7	210	0.167	213	0.173
8	159	0.198	163	0.206
9	121	0.229	126	0.240
10	89	0.258	96	0.271
11	67	0.286	74	0.302
12	50	0.316	56	0.333

Convection Section Layout

Oil Fired, 450°F Process Inlet 3 fins/inch, 0.75"high,0.1" thick				
Number Tube Rows	**4" Tube Cold End Approach, °F**	**Flue Gas ΔP, Inch H_2O**	**6" Tube Cold End Approach, °F**	**Flue Gas ΔP, Inch H_2O**
		0.2 lb/sec/ft² Mass Velocity		
6	218	0.024	251	0.020
7	153	0.029	186	0.025
8	108	0.034	139	0.029
9	81	0.038	104	0.033
10	54	0.043	77	0.037
11	37	0.048	58	0.041
12	26	0.052	42	0.045
13	18	0.057	32	0.048
		0.3 lb/sec/ft² Mass Velocity		
6	302	0.053	341	0.046
7	227	0.066	270	0.056
8	171	0.077	213	0.066
9	130	0.087	168	0.075
10	98	0.098	134	0.085
11	75	0.107	106	0.093
12	56	0.118	84	0.102
13	42	0.127	67	0.110
		0.4 lb/sec/ft² Mass Velocity		
6	363	0.094	406	0.081
7	285	0.116	331	0.099
8	224	0.137	270	0.118
9	177	0.155	221	0.134
10	140	0.174	183	0.151
11	111	0.191	150	0.166
12	87	0.209	124	0.182
13	70	0.226	102	0.196

Convection Section Layout

Oil Fired, 650°F Process Inlet 3 fins/inch, 0.75"high,0.1" thick				
Number Tube Rows	**4" Tube Cold End Approach, °F**	**Flue Gas ΔP, Inch H_2O**	**6" Tube Cold End Approach, °F**	**Flue Gas ΔP, Inch H_2O**
0.2 lb/sec/ft² Mass Velocity				
6	152	0.025	173	0.021
7	103	0.032	123	0.027
8	69	0.037	88	0.031
9	46	0.042	63	0.036
10	31	0.048	45	0.041
11	20	0.054	32	0.045
12	12	0.058	22	0.050
13	7	0.065	14	0.055
0.3 lb/sec/ft² Mass Velocity				
6	221	0.056	249	0.048
7	163	0.070	192	0.059
8	120	0.083	148	0.071
9	88	0.095	114	0.081
10	65	0.107	88	0.092
11	48	0.119	68	0.102
12	34	0.131	53	0.112
13	25	0.142	40	0.122
0.4 lb/sec/ft² Mass Velocity				
6	273	0.099	305	0.084
7	211	0.123	244	0.105
8	163	0.145	197	0.124
9	126	0.167	159	0.144
10	98	0.189	128	0.162
11	76	0.209	104	0.180
12	59	0.230	83	0.198
13	45	0.250	68	0.216

Escoa Method for Convection Calculations

The Escoa method for calculating heat transfer across tube banks is used by many heater vendors. The engineering manual and calculation program can be downloaded off the Internet at

http://www.escoa.com

The website has very good pictures.

The Escoa method gives an accurate heat transfer coefficient. The Escoa program does not solve the heat transfer row by row as the UOP simulator does. It solves total surface.

The program is not easy to use. The program predicts low flue gas pressure drops.

It is recommended that The UOP charts be used.

Convection Section

1. General

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

The convective constituent is handled in a manner similar to that in a typical exchanger but with cross-flow considered. The radiant constituent considers the radiation emanating from the polar flue gas molecules as we considered in the radiant section. An additional radiative factor is considered. This is the radiation of heat from the convection section walls to the tubes.

Prior to the above evaluation, one must consider the leakage of radiant heat through the shield tubes into the radiant section.

2. Radiation Leakage from the Radiant Section

Radiant section tubes receive direct radiation from the combustion gases and rebound radiation off the refractory wall in a single-fired (burners located on only one side of the tubes) radiant section. There is no refractory wall behind the bottom shock (or shield) row. Some radiation passes past the tubes through the spaces between them. This radiant heat is absorbed primarily by the tubes about the bottom shock row. The remainder is absorbed by the bottom convection section walls.

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.

The flue gas temperature to the remainder of the convection section must be corrected for the heat lost due to this leakage of heat.

3. Convective Heat Transfer

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

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

where: h_{Cc}= Convective Coefficient of Heat Transfer in the Convection Section, $Btu/ft^2hr°F$

T_f = Outside Film Temperature, °R

G = Mass Velocity of the Flue Gas, lb/ft^2sec

D = Outside diameter of Tube, inches

We can simplify the calculation by estimating the average gas film temperature as the average intube temperature plus one-half the log mean temperature difference from flue gas to fluid. The flue gas rate is at the minimum cross-section in the convection section, the position in each row where a plane intersects all tube centers.

4. Radiation from Flue Gas

A method for calculating the heat transfer by radiation from flue gas to a bank of tubes has been presented by Hottel. His basic relation was solved using typical tube spacing of about two tube diameters, a tube surface absorptivity of 0.9, and a typical partial pressure of carbon dioxide plus water. The calculation is simplified by assuming the average gas temperature is equal to 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. The radiative coefficient of heat transfer is denoted by h_{Cr}. Its units are $Btu/ft^2hr°F$.

5. Radiation from Refractory Walls

Following Monrad's pattern, we can estimate the effective coefficient for wall-to-tube transfer as a function of tube wall temperature as follows:

$$h_{Cw} = 9.46\ (T_t/1000)^3$$

where: h_{Cw} = Wall Radiation Coefficient of Heat Transfer in the Convection Section, $Btu/ft^2hr°F$

T_t = Average Tube Wall Temperature, °F

This coefficient, together with the relative wall and tube areas in the convection section, determines a correction factor, f, as follows:

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

where: h_{Cw} = Wall Radiative Heat Transfer Coefficient in Convection Section, $Btu/ft^2hr\ °F$

h_{Cc} = Convective Heat Transfer Coefficient in Convection Section, $Btu/ft^2hr\ °F$

h_{Cr} = Radiative Heat Transfer Coefficient in Convection Section, $Btu/ft^2hr\ °F$

A_{cw} = Refractory Wall Area in Convection Section

A_{Ct} = Tube Area in Convection Section.

6. Overall Heat Transfer Coefficient

The flue gas side gas film coefficient, h_o, is determined from the individual coefficients as follows:

$$h_o = (1 + f)\,(h_{Cc} + h_{Cr})$$

The overall coefficient is derived from the following equation:

$$U_o = \frac{1}{\frac{D_o}{D_i h_{di}} + \frac{D_o}{D_i h_i} + \frac{t_w}{k_m}\frac{D_o}{D_L} + \frac{1}{h_o} + \frac{D_o}{h_{do}}}$$

where: U_o = Overall Outside Convection Coefficient, Btu/ft^{2}hr°F
D_o = Outside Tube Diameter, inches
D_i = Inside Tube Diameter, inches
D_L = Logarithmic Mean Tube Diameter, inches
h_i = Inside Film Coefficient, Btu/ft^{2}hr°F
h_o = Overall Flue Gas Side Film Coefficient, Btu/ft^{2}hr°F
h_{di} = Inside Fouling Factor, Btu/ft^{2}hr°F
H_{do} = Inside Fouling Factor, Btu/ft^{2}hr°F
t_w = Tube Wall Thickness, ft
k_m = Tube Wall Thermal Conductivity, Btu/ft-hr°F

We can simplify the previous equation by assuming the following:

a. The ratio of the inside diameter to the outside diameter approaches 1.0.

b. The resistance due to the tube wall is negligible in comparison with the other resistances. The center term in the denominator approaches zero.

c. Fouling is negligible.

The above equation becomes:

$$U_o = \frac{h_i h_o}{h_i + h_o}$$

6. Overall Heat Transfer Coefficient

The film coefficient for the finned tube is related to the coefficient for a plain tube in that the convective coefficient is adjusted for the added surface:

$$h_f = h_{Cc}\left(\frac{EA_f + A_t}{A_t}\right)$$

where: h_f = Convective Heat Transfer Coefficient in Convection Section (Extended Surface Tube Basis), Btu/ft^{2}hr°F

h_{Cc} = Convective Heat Transfer Coefficient in Convection Section (Bare Tube Basis), Btu/ft^{2}hr°F

A_f = Area of Fins or Studs Only , ft^2/ Row

A_t = Total Surface Area of Bare Tube, ft^2/Row

E = Fin or Stud Efficiency

Extended surface, studs or fins, is not as effective as the bare tube surface. The fin surface is multiplied by an efficiency factor (less than 1.0) to correct the surface. The fin or stud efficiency, E, depends on the dimensions of the extended surface, its thermal conductivity and the convective heat transfer coefficient (bare tube basis).

The remainder of the rating procedure is handled in the same manner as for a bare tube. The difference is that convective heat transfer coefficient (extended surface tube basis), h_f, is substituted for the convective heat transfer coefficient (bare tube basis), h_{Cc}.

6. Overall Heat Transfer Coefficient++

The total surface required in the convection section is:

$$A_t = \frac{Q_C}{\Delta T_{LM} U_o}$$

where: A_t = Bare Tube Surface Area Required, ft

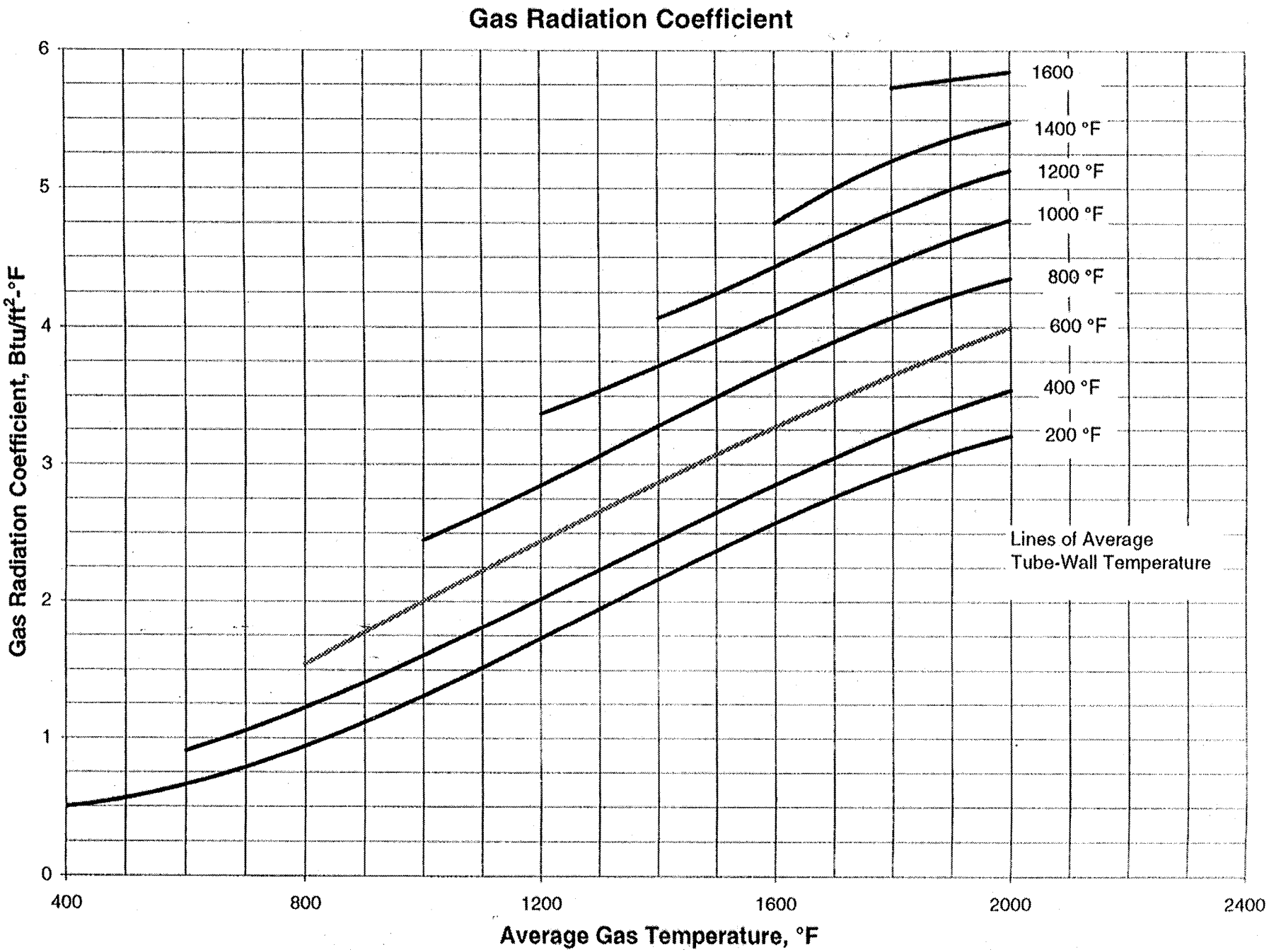
Q_C = Convection Section Duty, Btu/hr

ΔT_{LM} = Log Mean Temperature Difference, °F

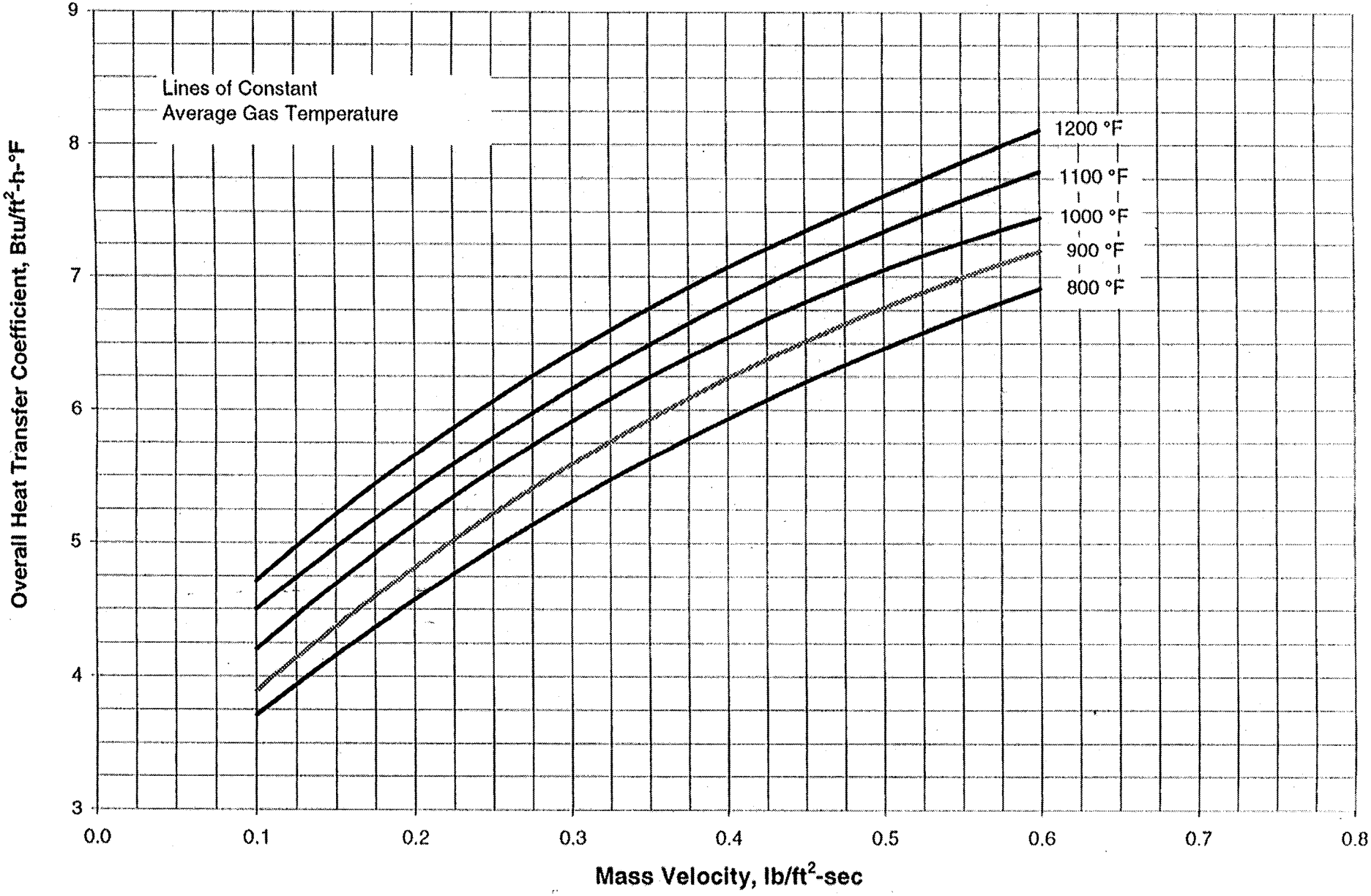
U_O = Overall Outside Convection Coefficient, Btu/ft^{2}hr°F

7. Extended Surface

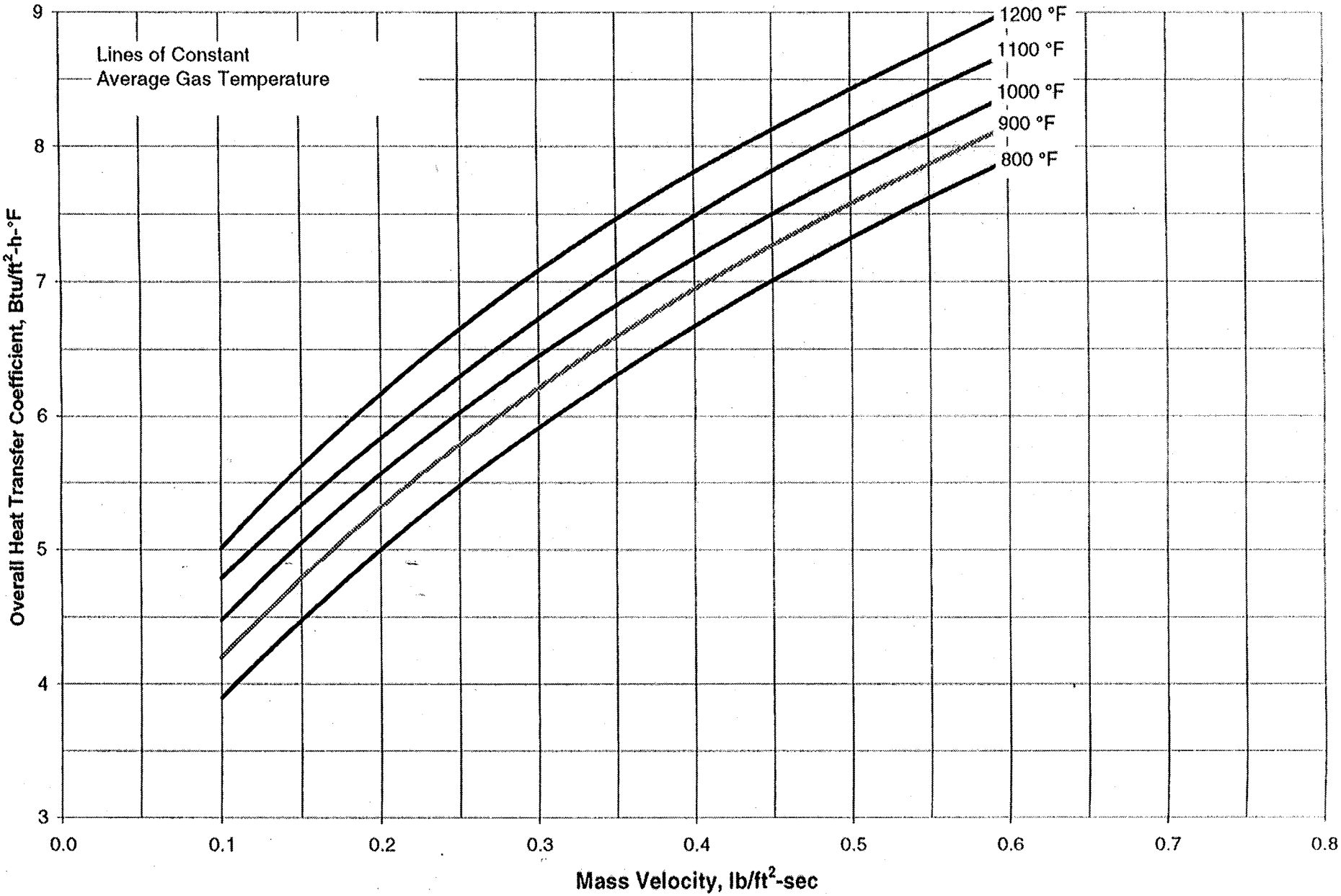
Extended surface (fins or studs) on convection section tubes is used to obtain additional heat transfer per length of tube. Convective heat transfer coefficients for such tubes depend upon the exact size and arrangement. They must be determined for the particular type being considered. The extended surface increases only the convective transfer. Radiant transfer must be computed on the basis of prime (bare) tube surface. Radiant transfer to convection section finned or studded tube is relatively small. It can be neglected in most cases.


Gas Radiation Coefficient
Gas Radiation Coefficient, Btu/ft²-°F
0
1
2
3
4
5
6
400
800
1200
1600
2000
2400
Average Gas Temperature, °F
1600
1400 °F
1200 °F
1000 °F
800 °F
600 °F
400 °F
200 °F
Lines of Average
Tube-Wall Temperature

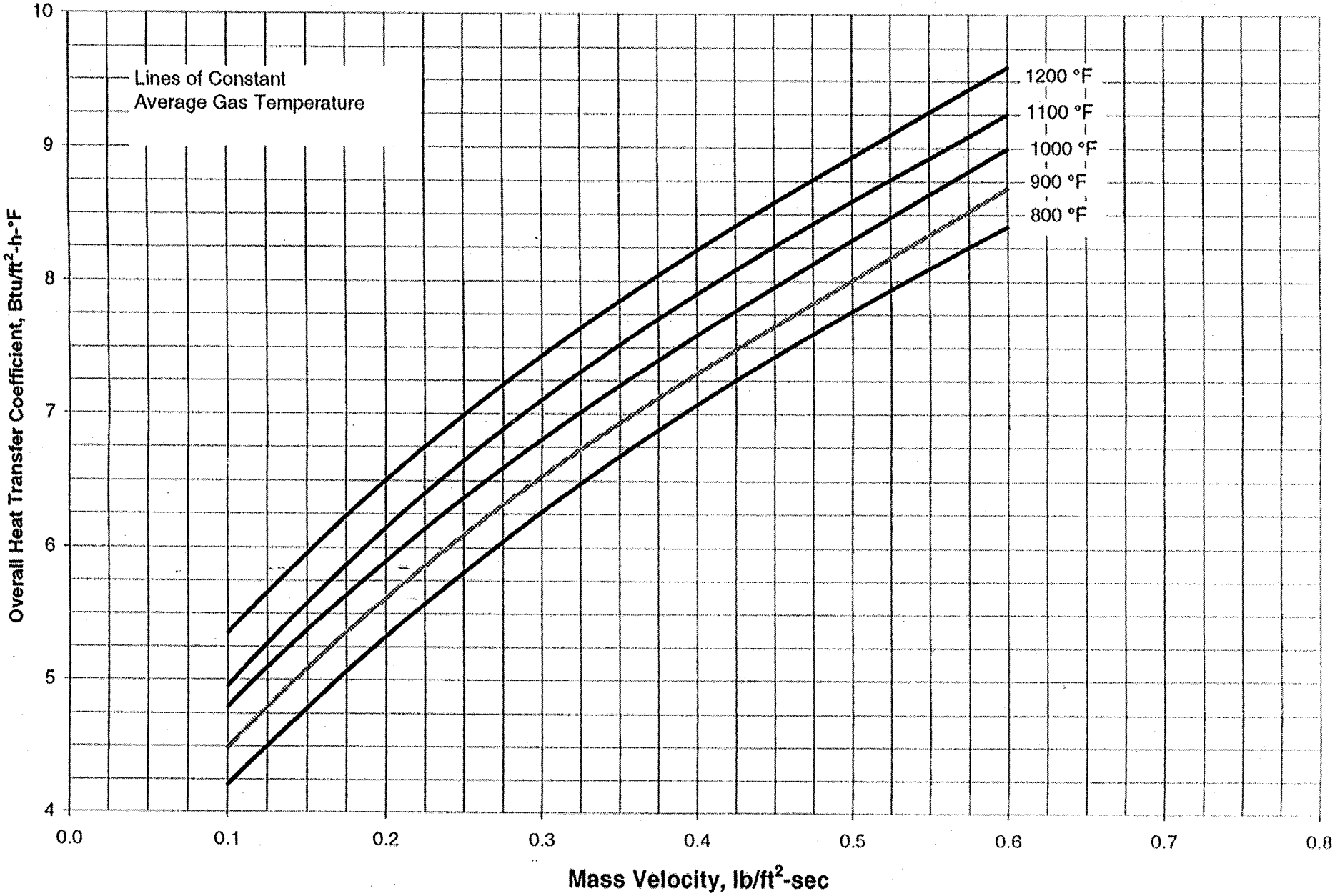
Overall Heat Transfer Coefficient
Convection Section - NPS 6



Overall Heat Transfer Coefficient
Convection Section - NPS 4



Overall Heat Transfer Coefficient
Convection Section - NPS 3

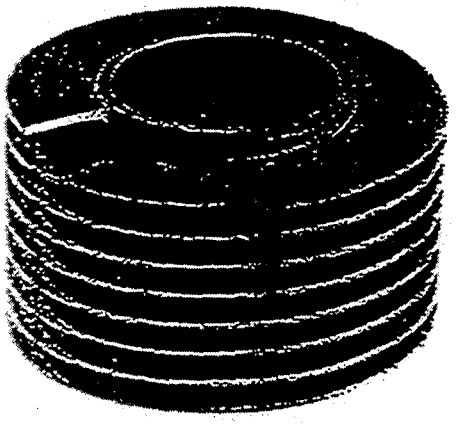


ESCOA

PRELIMINARY CONSIDERATIONS

WHY USE FINS?

The rate at which heat can be transferred from a hot fluid to a cold fluid through a tube wall depends on the heat transfer coefficient between the fluids and the tube wall along with the surface area to which each fluid is exposed. Using a bare tube, the fluid with the lowest heat transfer coefficient will control the overall heat transfer rate. In cases where the heat transfer coefficient of the fluid inside the tube is several times larger than that of the outside fluid, it is often advantageous to increase the outside surface area of the tube and thereby increase the overall heat transfer rate.



Escoa fins are resistance welded to tubes

TO INCREASE THE OUTSIDE SURFACE AREA OF THE TUBE

and

TO MAXIMIZE HEAT TRANSFER

in order

TO REDUCE THE TOTAL NUMBER OF TUBES REQUIRED

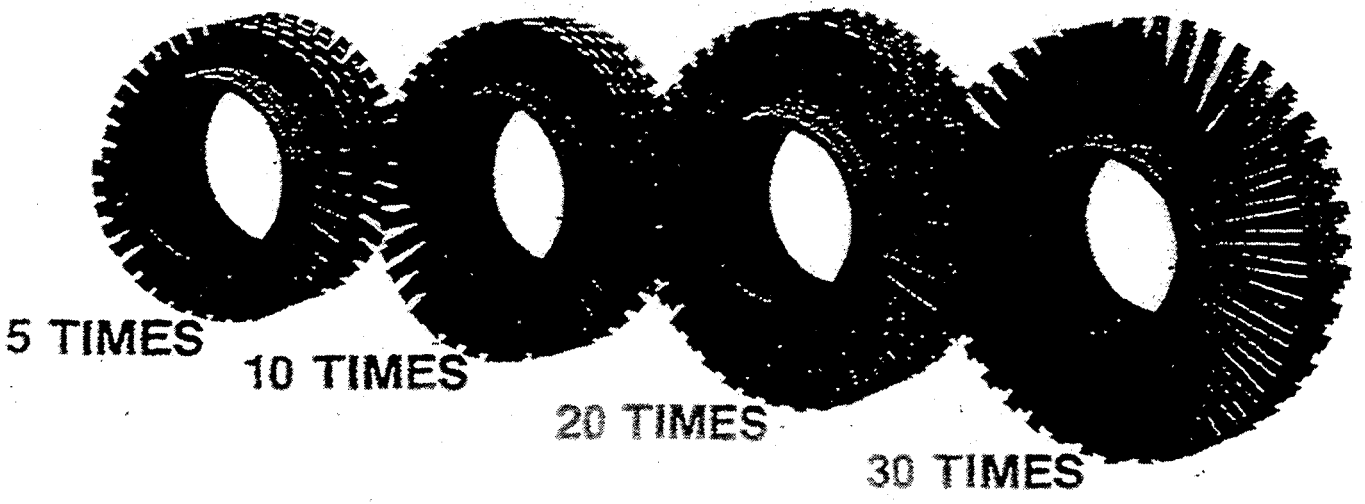
and

TO REDUCE OVERALL EQUIPMENT SIZE

in order

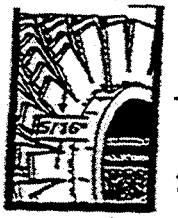
TO SAVE MONEY!

The outside surface area of a bare tube can be increased up to 30 times by finning, **giving the designer a wide range over which to optimize a design.**



In many common applications, one finned tube can replace 6 or more bare tubes at less than 1/3 the cost and 1/4 the volume.

ESCOA



TURB-X® HF

5/16" SEGMENT (8.0 mm)

$ft^2/ft \times 0.3048 = m^2/m$

TOTAL OUTSIDE SURFACE AREA, A_o, SQ. FT./FT. OF TUBE

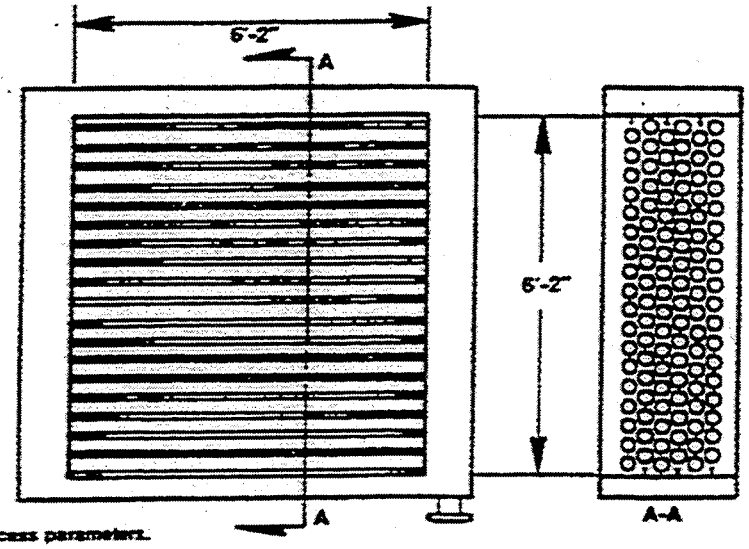
NOMINAL FIN THICKNESS INCHES (mm)	FINS INCH	FINS METRE	A_{po}	FIN HEIGHT, INCHES (mm) 0.375 (9.5)	0.500 (12.7)	0.625 (15.9)	0.750 (19.1)	0.875 (22.2)	1.000 (25.4)	1.250 (31.8)	1.500 (38.1)
0.036" (0.90)	1.0	39	1.14	2.15	2.50	2.84	3.18	3.52	3.87	4.55	5.24
	2.0	79	1.09	3.13	3.81	4.50	5.19	5.87	6.56	7.93	9.30
	2.5	98	1.07	3.61	4.47	5.33	6.19	7.04	7.90	9.62	11.33
	3.0	118	1.05	4.10	5.13	6.16	7.19	8.22	9.25	11.31	13.36
	3.5	138	1.03	4.59	5.79	6.99	8.19	9.39	10.59	12.99	15.39
	4.0	157	1.01	5.08	6.45	7.82	9.19	10.56	11.94	14.68	17.43
	4.5	177	0.99	5.56	7.11	8.65	10.19	11.74	13.28	16.37	19.46
	5.0	197	0.97	6.05	7.77	9.48	11.20	12.91	14.63	18.06	21.49
	5.5	217	0.95	6.54	8.42	10.31	12.20	14.08	15.97	19.74	23.52
	6.0	236	0.92	7.02	9.08	11.14	13.20	15.26	17.32	21.43	25.55
	7.0	276	0.88	8.00	10.40	12.80	15.20	17.60	20.01	24.81	29.61
0.049" (1.25)	1.0	39	1.12	2.17	2.53	2.88	3.24	3.59	3.85	4.66	5.37
	2.0	79	1.06	3.16	3.88	4.59	5.30	6.01	6.72	8.15	9.57
	2.5	98	1.03	3.66	4.55	5.44	6.33	7.22	8.11	9.89	11.67
	3.0	118	1.01	4.16	5.22	6.29	7.36	8.43	9.49	11.63	13.76
	3.5	138	0.98	4.65	5.90	7.14	8.39	9.64	10.88	13.37	15.86
	4.0	157	0.95	5.15	6.57	8.00	9.42	10.84	12.27	15.11	17.96
	4.5	177	0.92	5.65	7.25	8.85	10.45	12.05	13.65	16.86	20.06
	5.0	197	0.89	6.14	7.92	9.70	11.48	13.26	15.04	18.60	22.16
	5.5	217	0.86	6.64	8.60	10.55	12.51	14.47	16.43	20.34	24.25
	6.0	236	0.83	7.14	9.27	11.41	13.54	15.68	17.81	22.08	26.35
	7.0	276	0.77	8.13	10.62	13.11	15.60	18.09	20.58	25.57	30.55
0.059" (1.50)	1.0	39	1.11	2.19	2.55	2.92	3.28	3.65	4.01	4.75	5.48
	2.0	79	1.04	3.19	3.92	4.66	5.39	6.12	6.85	8.31	9.78
	2.5	98	1.00	3.70	4.61	5.52	6.44	7.35	8.27	10.10	11.92
	3.0	118	0.97	4.20	5.30	6.39	7.49	8.59	9.69	11.88	14.07
	3.5	138	0.94	4.70	5.98	7.26	8.54	9.82	11.10	13.66	16.22
	4.0	157	0.90	5.21	6.67	8.13	9.60	11.06	12.52	15.45	18.37
	4.5	177	0.87	5.71	7.36	9.00	10.65	12.29	13.94	17.23	20.52
	5.0	197	0.83	6.21	8.04	9.87	11.70	13.53	15.36	19.01	22.67
	5.5	217	0.80	6.72	8.73	10.74	12.75	14.76	16.77	20.80	24.82
	6.0	236	0.76	7.22	9.42	11.61	13.80	16.00	18.19	22.58	26.97
0.075" (1.90)	1.0	39	1.09		2.59	2.97	3.35	3.73	4.12	4.88	5.64
	2.0	79	1.00		4.00	4.76	5.53	6.29	7.05	8.58	10.10
	2.5	98	0.9		4.71	5.66	6.61	7.57	8.52	10.43	12.34
	3.0	118	0.91		5.41	6.56	7.70	8.85	9.99	12.28	14.57
	3.5	138	0.87		6.12	7.45	8.79	10.12	11.46	14.13	16.80
	4.0	157	0.83		6.82	8.35	9.88	11.40	12.93	15.98	19.03
	4.5	177	0.8		7.53	9.25	10.96	12.68	14.40	17.83	21.26
	5.0	197	0.7		8.24	10.14	12.05	13.96	15.86	19.68	23.49
0.102" (2.60)	1.0	39	1.06				3.47	3.88	4.29	5.10	5.92
	2.0	79	0.94				5.76	6.58	7.40	9.03	10.66
	2.5	98	0.88				6.91	7.83	8.85	10.99	13.03
	3.0	118	0.82				8.06	9.28	10.50	12.85	15.40
	3.5	138	0.76				9.20	10.63	12.06	14.92	17.77
	4.0	157	0.70				10.35	11.98	13.61	16.88	20.14
0.134" (3.40)	1.0	39	1.02				*3.61*	*4.05*	*4.49*	*5.37*	*6.25*
	2.0	79	0.86				*6.04*	*6.92*	*7.80*	*9.56*	*11.32*
	2.5	98	0.78				*7.26*	*8.36*	*9.46*	*11.66*	*13.85*
	3.0	118	0.71				*8.48*	*9.80*	*11.11*	*13.75*	*16.39*
0.165" (4.20)	1.0	39	0.96				3.75	4.22	4.69	5.63	6.57
	2.0	79	0.79				6.32	7.26	8.20	10.08	11.96
	2.5	98	0.69				7.60	8.78	9.95	12.30	14.65
	3.0	118	0.60				8.89	10.30	11.71	14.53	17.35

RED ITALIC INDICATES GEOMETRIES GENERALLY AVAILABLE IN

ESCOA®

SOLIDFIN® HF
SAMPLE CALCULATION

Problem: Determine the number of tube rows required in a gas turbine heat recovery unit to heat 65000 lb/hr of Dowtherm G from 290 F to 510 F by cooling 132000 lb/hr of gas turbine exhaust from 800 F to 600 F. The bundle is enclosed in a square duct and the entire tube length inside the duct is finned. Every sixth tube in the first row is welded into a common header so the heat transfer fluid makes six passes per row. The tubes are finned with SOLIDFIN HF extended surface and are arranged in a staggered equilateral layout. Baffles are welded to the duct wall at alternating ends of each tube row to minimize bypassing. Assume inside and outside fouling resistances of 0.001 and 0.002 respectively. Assume heat transferred by radiation is negligible.



1. **Basic geometry and process parameters.**

d_o	=2.375 in	L_t	=6.17 ft	W	=132000 lb/hr	c_p	=0.254 Btu/lb F
t_w	=0.154 in	N_t	=18/row	R_s	=0.001 hr ft² F/Btu	μ_b	=0.0764 lb/hr ft
P_t	=4.00 in	A_d	=38.03 ft²	R_{fo}	=0.002 hr ft² F/Btu	ρ_b	=0.0342 lb/ft³
P_l	=3.46 in	X_c	=0.93 ft²	T_o	=700 F	ρ_t	=0.0314 lb/ft³
i_f	=0.500 in	T_1	=800 F	T_i	=410 F	ρ_2	=0.0374 lb/ft³
t_f	=0.059 in	T_2	=600 F	T_b	=497 F*	k_b	=0.0284 Btu/hr ft F
n_f	=5.0/in	t_1	=300 F	T_w	=439 F*	k_f	=27.0 Btu/hr ft F
		t_2	=520 F	*Approximations		k_w	=27.5 Btu/hr ft F

2. **Additional geometry and process parameter calculations.**
   a. $d_i = 2.375 - 2 \times 0.154 = 2.067$ in
   b. $A_i = \pi 2.067/12 = 0.54$ ft²/ft
   c. $A_w = \pi(2.375 - 0.154)/12 = 0.58$ ft²/ft
   d. $d_f = 2.375 + 2 \times 0.500 = 3.375$ in
   e. $s_f = (1/5.0) - 0.059 = 0.141$ in
   f. $A_{po} = 0.44$ ft²/ft from page D-25
      $A_o = 4.46$ ft²/ft from page D-25
   g. $A_{fo} = 4.46 - 0.44 = 4.02$ ft²/ft
   h. $A_c = (2.375 + 2 \times 0.500 \times 0.059 \times 5.0)/12 = 0.22$ ft²/ft
   i. $A_n = 38.03 - 0.22 \times 6.17 \times 18 - 0.93 = 12.67$ ft²
   j. $G_n = 132000/12.67 = 10418$ lb/hr ft²
   k. $Re = 10418 \times 2.375/12 \times 0.0764 = 26988$

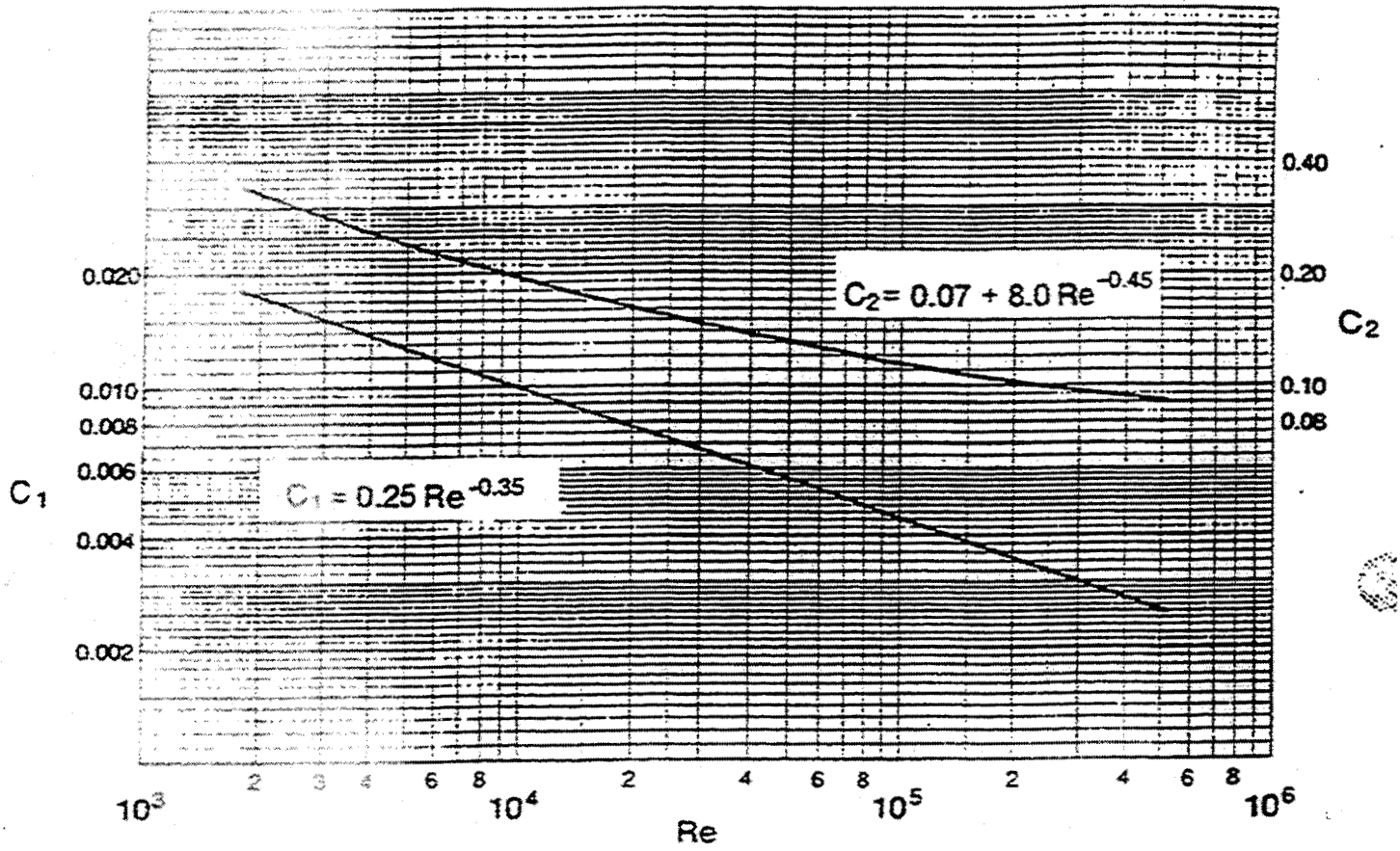
3. **Heat transfer calculations.**
   a. $C_1 = 0.0070$ for $Re = 26988$ (page D-10, upper graph)
   b. $C_3 = 0.618$ for $i_f/s_f = 3.55$ (page D-10, lower graph)
   c. $C_5 = 1.000$ for $P_t/P_l = 0.866$ and $N_r > 4$ (page D-11, upper graph)
   d. $j = 0.0070 \times 0.618 \times 1.000\ (3.375/2.375)^{0.5}\ [(700 + 460)/(497 + 460)]^{0.25} = 0.0054$
   e. $h_c = 0.0054 \times 10418 \times 0.245\ (0.0284/0.254 \times 0.0764)^{0.67} = 17.79$ Btu/hr ft² F
   f. $h_r = 0$
   g. $h_o = 1/[(1/17.79) + 0.002] = 17.18$ Btu/hr ft² F
   h. $E = 0.80$ (page D-14)
   i. $h_e = 17.18\ (0.80 \times 4.02 \div 0.44)\ /\ 4.46 = 14.06$ Btu/hr ft² F
   j. $R_o = 1/14.06 = 0.0710$ hr ft² F/Btu
   k. $R_{wo} = (0.154/12 \times 27.5)\ (4.46/0.58) = 0.0034$ hr ft² F/Btu
   l. $h_i = 240$ Btu/hr ft² F (page 6, Dow Chemical Bulletin 176-1213-72)
   m. $R_{io} = [(1/240) \div 0.001]\ (4.46/0.54) = 0.0427$ hr ft² F/Btu
   n. $R_{to} = 0.0710 + 0.0034 + 0.0427 = 0.1171$ hr ft² F/Btu
   o. $U_o = 1/0.1171 = 8.54$ Btu/hr ft² F
   p. $\Delta T = [(800-520)-(600-300)]/ln\ [(800-520)/(600-300)] = 290$ F
   q. $Q = 132000 \times 0.254\ (800 - 600) = 6.71 \times 10^6$ Btu/hr
   r. $A_t = 6.71 \times 10^6/290 \times 8.54 = 2709$ ft²
   s. $N_r = 2709/(4.46 \times 6.17 \times 18) = 5.47$ use $N_r = 6$

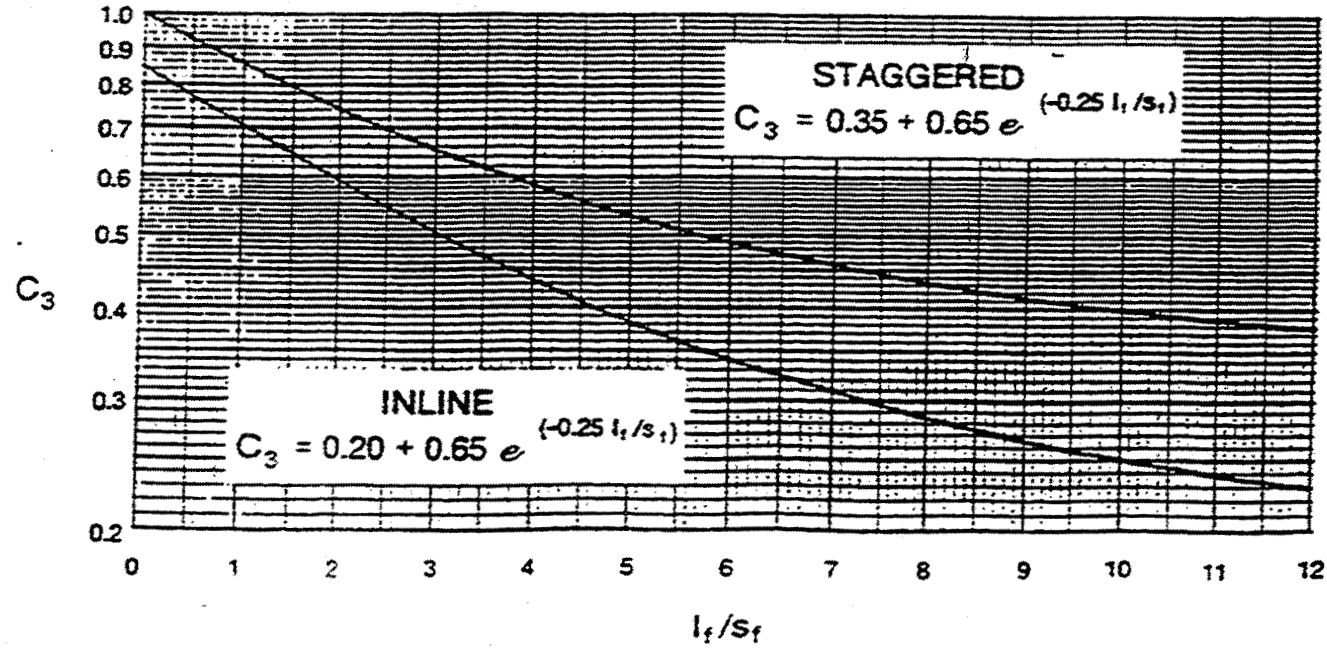
4. **Pressure drop calculations.**
   a. $C_2 = 0.151$ for $Re = 26988$ (page D-10, upper graph)
   b. $C_4 = 1.027$ for $P_l/d_o = 1.68$ and $i_f/s_f = 3.55$ (page D-13, upper graph)
   c. $C_6 = 1.000$ for $P_t/P_l = 0.866$ and $N_r > 4$ (page D-13, upper graph)
   d. $f = 0.151 \times 1.027 \times 1.000\ (3.375/2.375)^{0.5} = 0.185$
   e. $\beta^2 = (12.67/38.03)^2 = 0.111$
   f. $a = [(1 + 0.111)/(4 \times 6)]\ 0.0342\ [(1/0.0374) - (1/0.0314)] = -0.008$
   g. $\Delta P = (0.185 - 0.008)(10418)^2\ 6/(0.0342 \times 1.083 \times 10^9) = 3.11$ in of water.

SOLIDFIN® HF

C_1, C_2 AND C_3

ESCOA®





NATURAL CONVECTION

ESCOA

Natural Convection Coefficient
Horizontal Tube in Air

$$h_c = h_n \gamma_n$$

A_b = bare tube outside surface area, ft²/ft

A_o = total outside surface area, ft²/ft

h_c = average outside convection heat transfer coefficient, Btu/hr ft² F

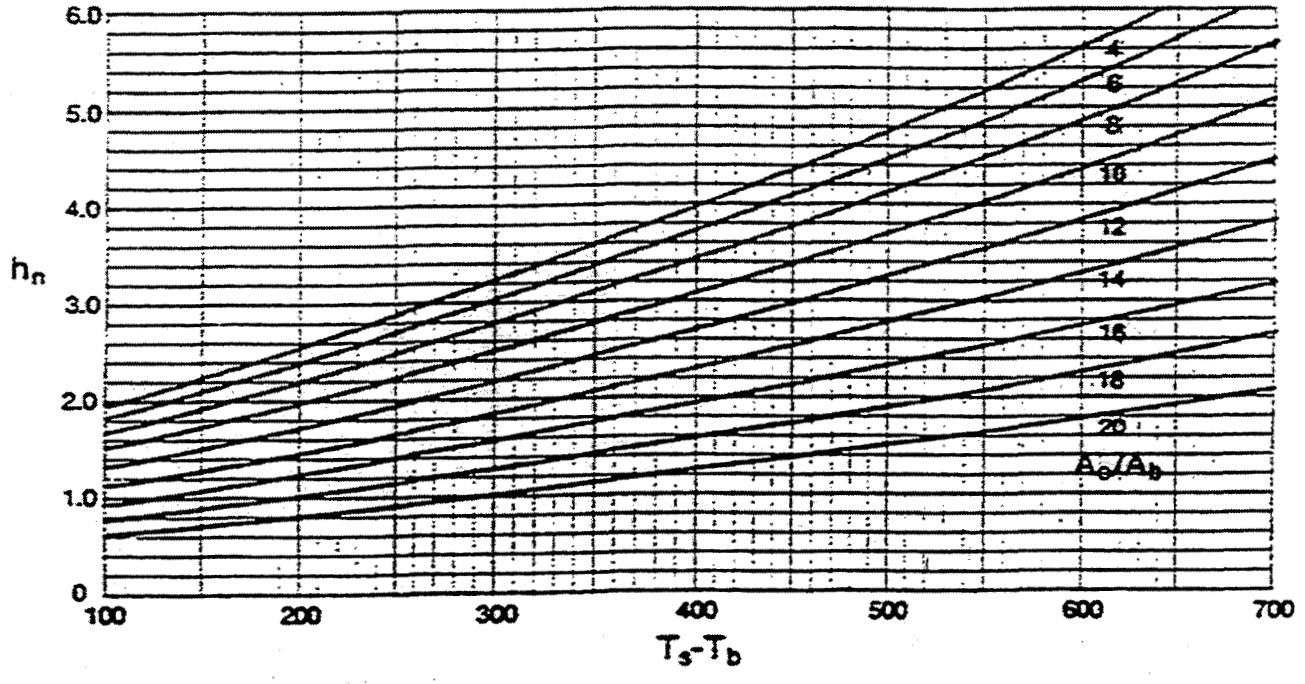
h_n = natural convection heat transfer coefficient in 90 F air, Btu/hr ft² F

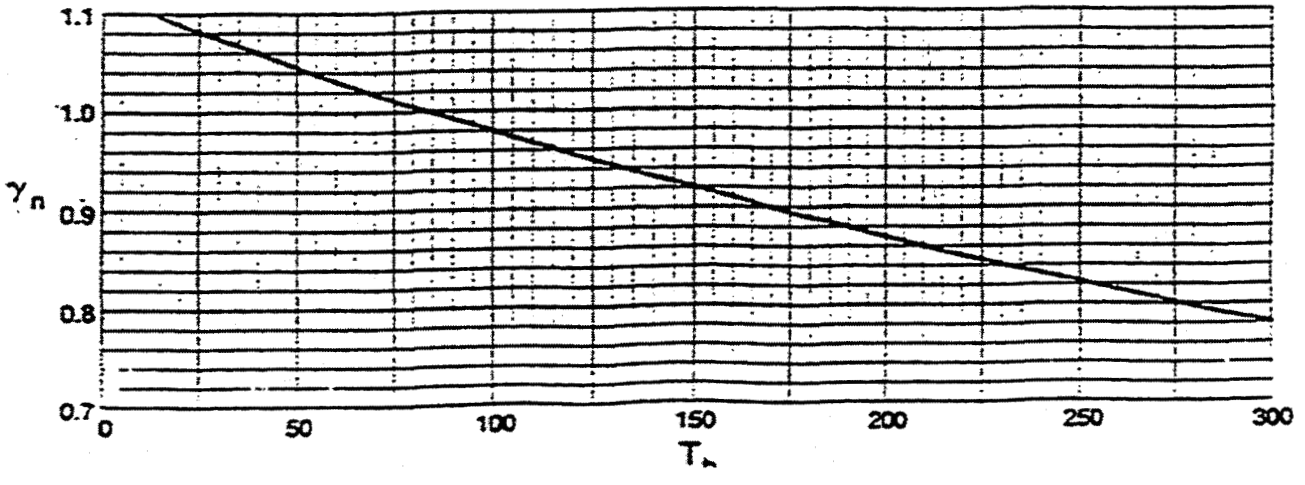
T_b = ambient air temperature, F

T_i = average inside fluid temperature, F

T_s = average fin surface temperature, F
if unknown, assume $T_s = T_i - 0.1\ (T_i - T_b)$

γ_n = temperature correction factor





ESCOA®

FIN TIP TEMPERATURE

Tube Wall Temperature*

$$T_w = T_i + R_{io} (T_o - T_i) / (R_{io} + R_o)$$

$$R_{io} = (A_o / A_i) (R_{fi} + 1/h_i)$$

$$R_o = 1/h_e$$

Fin Tip Temperature*

$$T_f = T_w + \theta (T_o - T_w)$$

Fin Surface Temperature

$$T_s = (T_w - T_f) / 2$$

* These equations give the average value for the tube wall and fin tip temperatures. If the maximum tube wall and fin tip temperatures are to be determined, h_o should be replaced by h_m when calculating E and h_e.

A_o = total outside surface area, ft²/ft

A_i = total inside surface area, ft²/ft

E = fin efficiency

h_i = average inside heat transfer coefficient, Btu/hr ft² F

h_m = maximum or minimum outside heat transfer coefficient, Btu/hr ft² F
$h_m = 1.5\ h_o$ for $T_o > T_i$
$h_m = 0.5\ h_o$ for $T_o < T_i$

h_e = effective outside heat transfer coefficient, Btu/hr ft² F
$h_e = (h_o/A_o)(E A_{to} + A_{po})$

h_o = average outside heat transfer coefficient, Btu/hr ft² F

R_{fi} = inside fouling resistance, hr ft² F/Btu

R_{io} = total inside thermal resistance, hr ft² F/Btu

R_o = total outside thermal resistance, hr ft² F/Btu

T_f = fin tip temperature, F

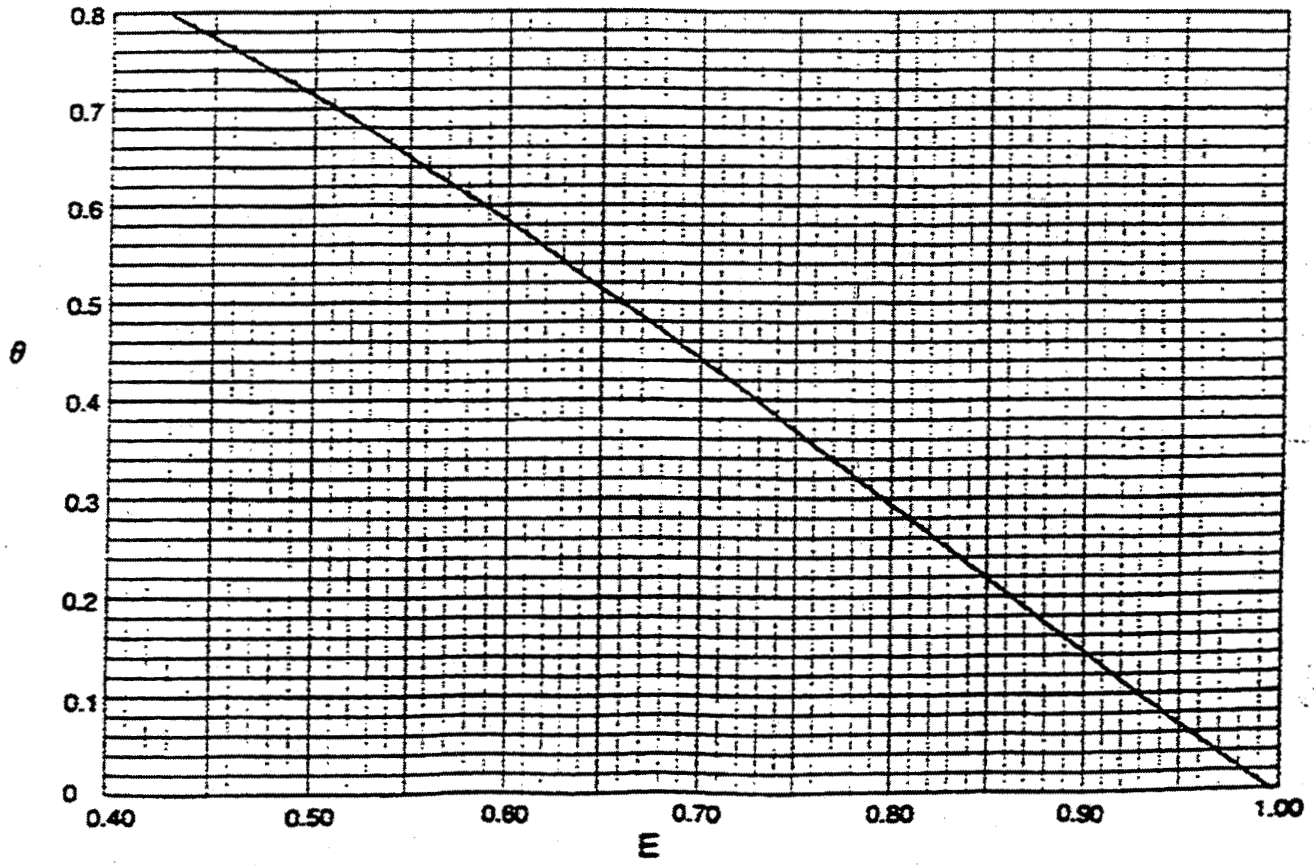
T_i = fluid temperature inside tube, F

T_o = fluid temperature outside tube, F

T_s = average fin surface temperature, F

T_w = average tube wall temperature, F

θ = temperature differential factor



TUBE AND PIPE WEIGHTS

ESCOA

TABLE A5 CARBON STEEL TUBING WEIGHT*, LB./FT.

TUBE O.D. INCHES	TUBE WALL THICKNESS, INCHES										
	0.049	0.060	0.065	0.075	0.083	0.095	0.105	0.110	0.120	0.134	0.150
0.500	.2360	.2820	.3020	.3404	.3696	.4109	.4430	.4582	.4870	.5238	.5607
	.2655	.3176	.3367	.3803	.4109	.4552	.4870	.5033	.5311	.5670	.6008
0.625	.3014	.3621	.3888	.4406	.4805	.5377	.5831	.6050	.6472	.7027	.7610
	.3403	.4097	.4355	.4951	.5377	.6007	.6472	.6715	.7140	.7713	.8291
0.750		.4422	.4755	.5407	.5913	.6646	.7233	.7519	.8074	.8816	.9612
	.4151	.5018	.5343	.6099	.6646	.7462	.8074	.8397	.8969	.9755	1.057
0.875		.5223	.5623	.6408	.7021	.7914	.8635	.8987	.9676	1.060	1.161
		.5940	.6330	.7247	.7914	.8917	.9676	1.008	1.080	1.180	1.286
1.000			.6491	.7409	.8129	.9182	1.004	1.046	1.128	1.239	1.362
		.6861	.7318	.8395	.9182	1.037	1.128	1.176	1.263	1.384	1.514
1.250				.9412	1.034	1.172	1.284	1.339	1.448	1.597	1.762
			.9294	1.069	1.172	1.328	1.448	1.513	1.628	1.793	1.971
1.500					1.256	1.426	1.564	1.633	1.769	1.955	2.163
				1.299	1.426	1.619	1.769	1.849	1.994	2.201	2.427
1.750						1.679	1.845	1.927	2.089	2.313	2.563
					1.679	1.910	2.089	2.185	2.360	2.610	2.884

TUBE O.D. INCHES	TUBE WALL THICKNESS, INCHES										
	0.165	0.110	0.120	0.134	0.150	0.165	0.188	0.203	0.238	0.250	0.300
2.000	2.125	2.220	2.409	2.670	2.964	3.234	3.638	3.896	4.479	4.673	5.447
	2.409	2.522	2.726	3.018	3.340	3.656	4.099	4.381	5.020	5.235	6.070
2.250	2.405	2.514	2.730	3.028	3.364	3.674	4.140	4.438	5.114	5.340	6.248
	2.730	2.858	3.092	3.427	3.797	4.160	4.673	5.000	5.745	5.999	6.986
2.500		2.808	3.050	3.386	3.785	4.115	4.642	4.980	5.750	6.008	7.049
	3.050	3.195	3.457	3.835	4.253	4.665	5.247	5.620	6.472	6.763	7.902
3.000			3.691	4.102	4.566	4.996	5.646	6.084	7.021	7.343	8.651
		3.867	4.189	4.652	5.167	5.674	6.395	6.858	7.925	8.290	9.733
5.000				6.964	7.770	8.520	9.662	10.40	12.10	12.68	15.06
			7.115	7.920	8.819	9.711	10.99	11.81	13.73	14.40	17.06
6.000					9.372	10.28	11.67	12.57	14.65	15.35	18.26
				9.554	10.65	11.73	13.28	14.29	16.64	17.45	20.72
10.000										26.03	31.08
									28.26	29.67	35.38
12.000										31.37	37.49
									34.07	35.78	42.70

* WEIGHTS SHOWN IN BLACK ARE FOR AVERAGE WALL. WEIGHTS SHOWN IN RED ARE FOR MINIMUM WALL.

TABLE A6 CARBON STEEL PIPE WEIGHT, LB./FT.

NOMINAL PIPE SIZE	ACTUAL PIPE O.D. INCHES	SCHEDULE 10*		SCHEDULE 40		SCHEDULE 80		SCHEDULE 160	
		WALL INCHES	WEIGHT LB./FT.	WALL INCHES	WEIGHT LB./FT.	WALL INCHES	WEIGHT LB./FT.	WALL INCHES	WEIGHT LB./FT
¾	1.050	.083	.8572	.113	1.131	.154	1.474	.218	1.937
1	1.315	.109	1.404	.133	1.679	.179	2.172	.250	2.844
1¼	1.660	.109	1.806	.140	2.273	.191	2.997	.250	3.765
1½	1.900	.109	2.085	.145	2.718	.200	3.631	.281	4.859
2	2.375	.109	2.638	.154	3.653	.218	5.022	.343	7.444
2½	2.875	.120	3.531	.203	5.793	.276	7.661	.375	10.01
3	3.500	.120	4.332	.216	7.576	.300	10.25	.438	14.32
3½	4.000	.120	4.973	.226	9.109	.318	12.50		
4	4.500	.120	5.613	.237	10.79	.337	14.98	.531	22.51
5	5.563	.134	7.770	.258	14.62	.375	20.78	.625	32.96
6	6.625	.134	9.289	.280	18.97	.432	28.57	.718	45.30
8	8.625	.148	13.40	.322	28.55	.500	43.39	.906	74.69
10	10.750	.165	18.70	.365	40.48	.593	64.33	1.125	115.7
12	12.750	.180	24.20	.406	53.53	.687	88.51	1.312	160.3

* GENERALLY AVAILABLE IN STAINLESS STEEL GRADES ONLY.

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PHYSICAL PROPERTIES

TABLE A1 PROPERTIES OF DRY AIR AT 1 ATMOSPHERE

TEMP. F.	ρ_b lb/ft³	c_p Btu/lb F	μ_b lb/hr ft	k_b Btu/hr ft F	$c_p\left[\frac{k_b}{c_p \mu_b}\right]^{.667}$
70	0.0749	0.2401	0.0440	0.0149	0.302
100	0.0709	0.2404	0.0459	0.0157	0.304
200	0.0601	0.2413	0.0520	0.0180	0.307
300	0.0521	0.2429	0.0575	0.0203	0.312
400	0.0462	0.2451	0.0626	0.0225	0.317
500	0.0413	0.2476	0.0675	0.0245	0.321
600	0.0374	0.2505	0.0722	0.0266	0.324
700	0.0342	0.2536	0.0764	0.0284	0.328
800	0.0314	0.2568	0.0806	0.0303	0.332
900	0.0292	0.2600	0.0846	0.0321	0.335
1000	0.0272	0.2630	0.0884	0.0337	0.337
1100	0.0254	0.2663	0.0922	0.0353	0.340
1200	0.0239	0.2690	0.0957	0.0369	0.342
1300	0.0225	0.2718	0.0992	0.0385	0.345
1400	0.0213	0.2745	0.1027	0.0400	0.347
1500	0.0202	0.2770	0.1063	0.0416	0.349
1600	0.0191	0.2795	0.1100	0.0431	0.350
1700	0.0182	0.2817	0.1136	0.0447	0.352
1800	0.0174	0.2839	0.1172	0.0461	0.353

TABLE A2 PROPERTIES OF FLUE GAS 76% N_2, 12% CO_2, 12% H_2O (VOLUME)

TEMP. F.	ρ_g lb/ft³	c_g Btu/lb F	μ_g lb/hr ft	k_g Btu/hr ft F	$c_g\left[\frac{k_g}{c_g \mu_g}\right]^{.667}$
200	0.0602	0.258	0.0469	0.0168	0.321
300	0.0522	0.261	0.0521	0.0191	0.328
400	0.0460	0.265	0.0571	0.0213	0.333
500	0.0411	0.268	0.0619	0.0235	0.338
600	0.0373	0.272	0.0665	0.0255	0.342
700	0.0341	0.277	0.0709	0.0276	0.348
800	0.0313	0.281	0.0751	0.0296	0.353
900	0.0290	0.285	0.0791	0.0314	0.356
1000	0.0270	0.289	0.0829	0.0332	0.360
1100	0.0253	0.293	0.0865	0.0349	0.363
1200	0.0238	0.296	0.0900	0.0367	0.367
1300	0.0224	0.300	0.0933	0.0383	0.370
1400	0.0212	0.304	0.0965	0.0399	0.374
1500	0.0201	0.307	0.0996	0.0414	0.376
1600	0.0191	0.310	0.1026	0.0429	0.379
1700	0.0181	0.312	0.1055	0.0443	0.381
1800	0.0172	0.314	0.1083	0.0457	0.383
1900	0.0163	0.317	0.1111	0.0470	0.385
2000	0.0155	0.319	0.1139	0.0483	0.386

TABLE A3 THERMAL CONDUCTIVITY OF FIN MATERIALS, k_t, BTU/HR FT F

10 TO 0.067HK A-36 > 0.067HK A-596

ASTM MATERIAL SPECIFICATION

TEMP. F.	CARBON STEEL	A167 TP 304	A167 TP 310	A167 TP 316	A167 TP 321	A176 TP 409	A176 TP 410	A176 TP 430	A176 XM-33	B168 INCONEL	B409 INCOLOY	B162 NICKEL
200	29.4	9.2	8.0	9.2	9.2	14.3	14.7	13.7	10.4	8.6	7.3	36.4
300	28.6	9.5	8.6	9.6	9.6	14.6	15.0	13.9	10.8	9.1	7.9	35.1
400	27.8	10.1	9.1	10.1	10.1	14.8	15.2	14.1	11.1	9.7	8.4	33.8
500	27.0	10.6	9.6	10.6	10.6	15.1	15.5	14.3	11.4	10.3	9.0	32.5
600	26.2	11.1	10.1	11.1	11.1	15.3	15.7	14.4	11.7	10.9	9.5	31.2
700	25.4	11.6	10.6	11.6	11.6	15.6	16.0	14.6	12.1	11.5	10.1	29.9
800	24.6	12.0	11.2	12.0	12.0	15.8	16.2	14.8	12.4	12.0	10.6	30.7
900	23.8	12.5	11.7	12.5	12.5	16.1	16.5	14.9	12.7	12.6	11.2	31.5
1000	23.0	13.0	12.2	13.0	13.0	16.3	16.7	15.1	13.0	13.2	11.7	32.4
1100	22.2	13.5	12.7	13.5	13.5	16.6	17.0	15.3	13.4	13.8	12.3	33.2
1200	21.4	14.0	13.2	14.0	14.0	16.8	17.2	15.4	13.7	14.4	12.8	34.0
1300	20.8	14.4	13.8	14.4	14.4	17.1	17.5	15.6	14.0	14.9	13.4	34.9
1400	20.4	14.9	14.3	14.9	14.9	17.3	17.7	15.8	14.3	15.5	13.9	35.7
1500	20.0	15.4	14.8	15.4	15.4	17.6	18.0	16.0	14.6	16.1	14.5	36.5
1600	19.7	15.9	15.3	15.9	15.9	17.8	18.2	16.1	15.0	16.7	15.0	37.4
1700	19.5	16.4	15.8	16.4	16.4	18.1	18.5	16.3	15.3	17.3	15.6	38.2
1800	—	16.8	16.4	16.8	16.8	18.3	18.7	16.5	15.6	17.8	16.1	39.0
1900	—	17.3	16.9	17.3	17.3	18.6	19.0	16.6	15.9	18.4	16.7	39.8
2000	—	17.8	17.4	17.8	17.8	18.8	19.2	16.8	16.3	19.0	17.2	40.7
MAX. TEMP *	950	1650	2000	1650	1650	1250	1250	1250	2000	2000	2000	2000

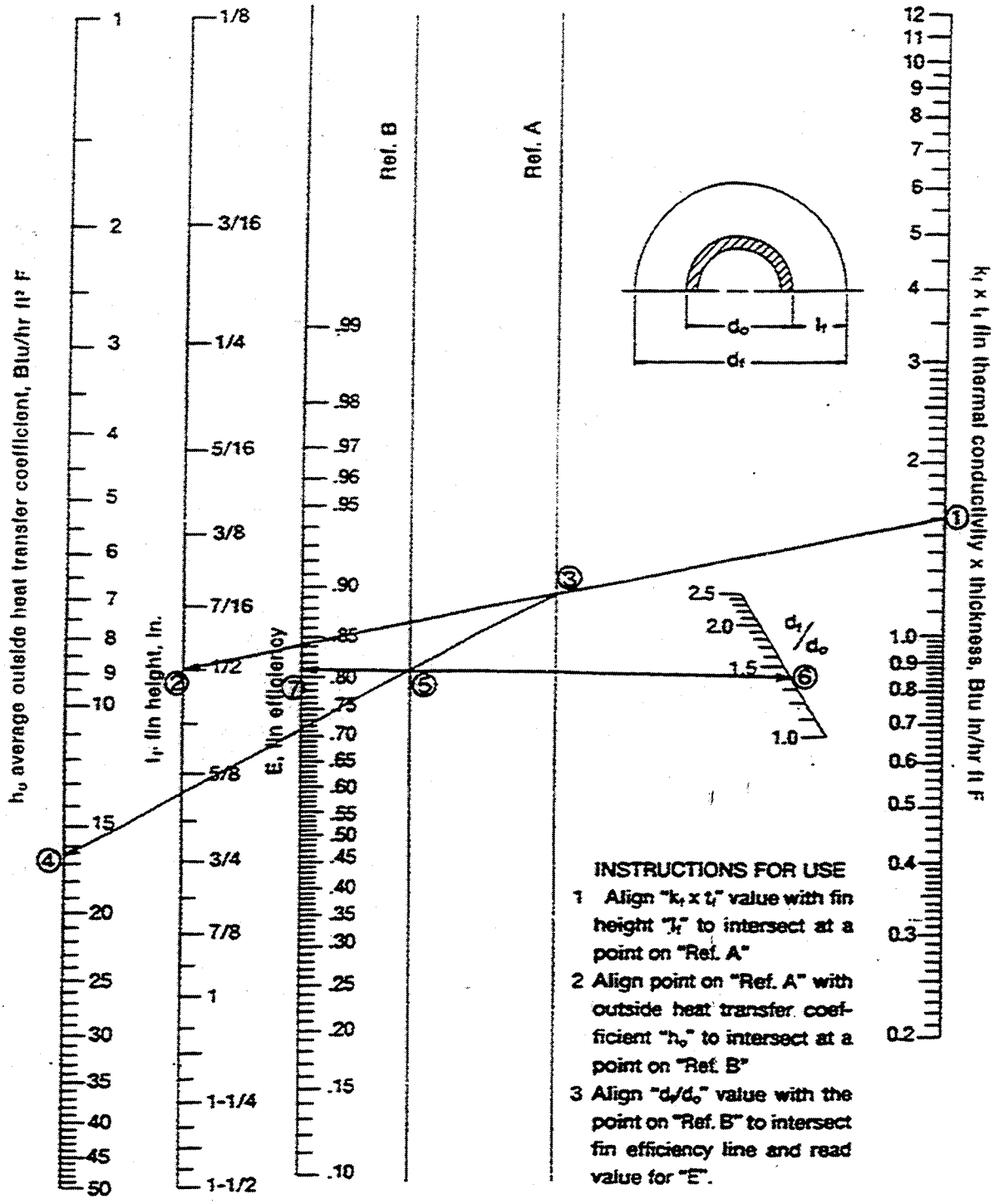
* MATERIAL MANUFACTURERS RECOMMENDED MAXIMUM METAL TEMPERATURE TO AVOID EXCESSIVE SCALING DUE TO OXIDATION IN ORDINARY FLUE GASES. WHERE SERVICE IN MORE CORROSIVE ATMOSPHERES IS INTENDED, THE MAXIMUM METAL TEMPERATURE SHOULD BE ADJUSTED ACCORDINGLY.

TABLE A4 OUTSIDE FOULING FACTORS, R_{fo}, FOR FLUE GASES. HR FT² F/BTU AND RECOMMENDED MINIMUM FIN SPACINGS, s_t, INCHES

FUEL	DRY AIR	NATURAL GAS	PROPANE	BUTANE	#2 FUEL OIL	#6 FUEL OIL	CRUDE OIL	RESIDUAL OIL	COAL	WOOD WASTES
R_{fo}	.0005	.001-.003	.001-.003	.001-.003	.002-.004	.003-.007	.008-.015	.01-.02	.01-.05	.01-.05
s_t	0.05	0.07	0.07	0.07	0.12	0.18	0.20	0.20	0.34	0.34

SOLIDFIN® HF
FIN EFFICIENCY

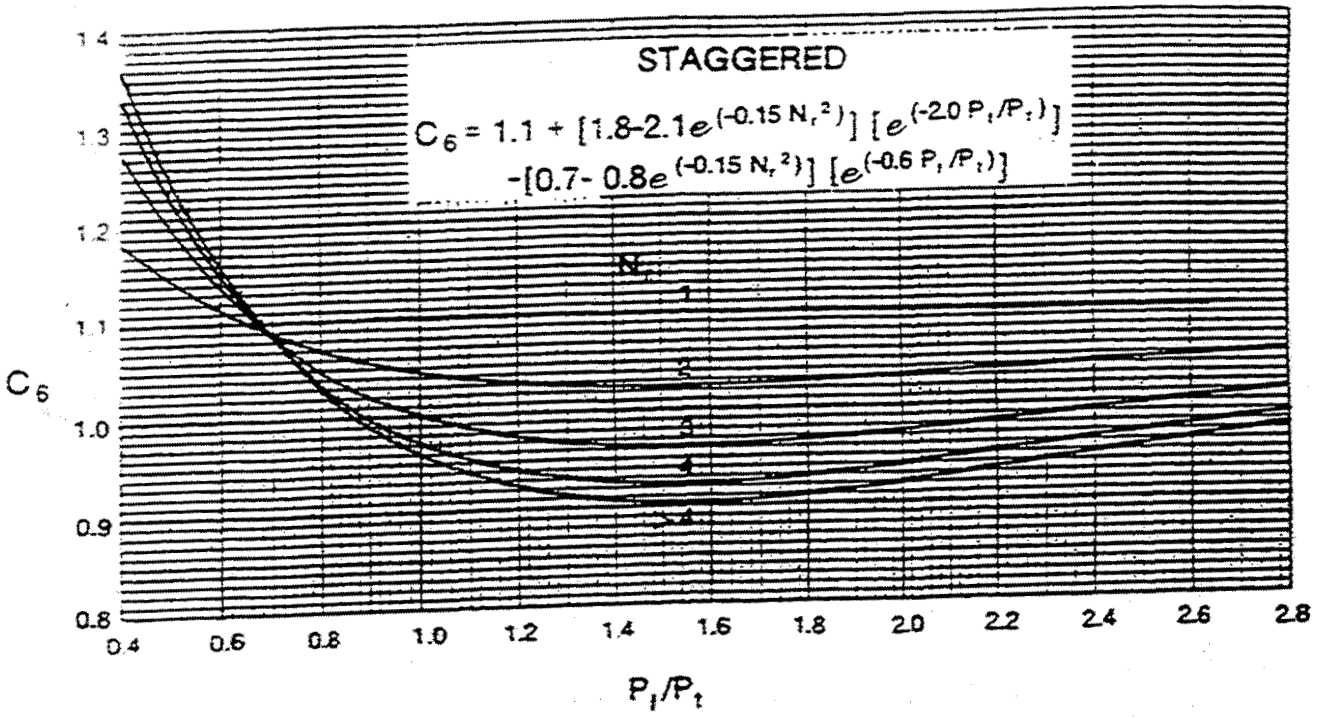
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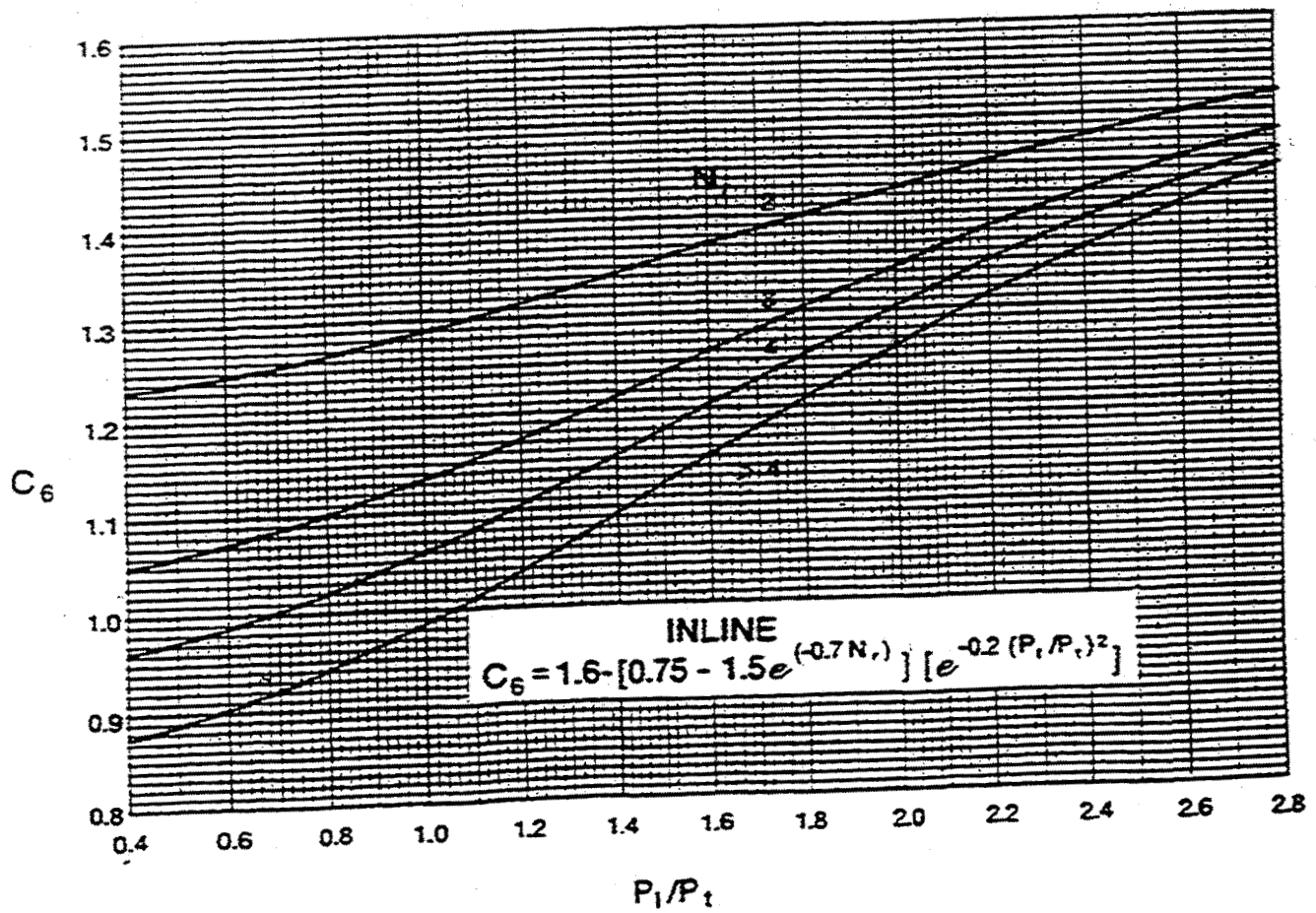


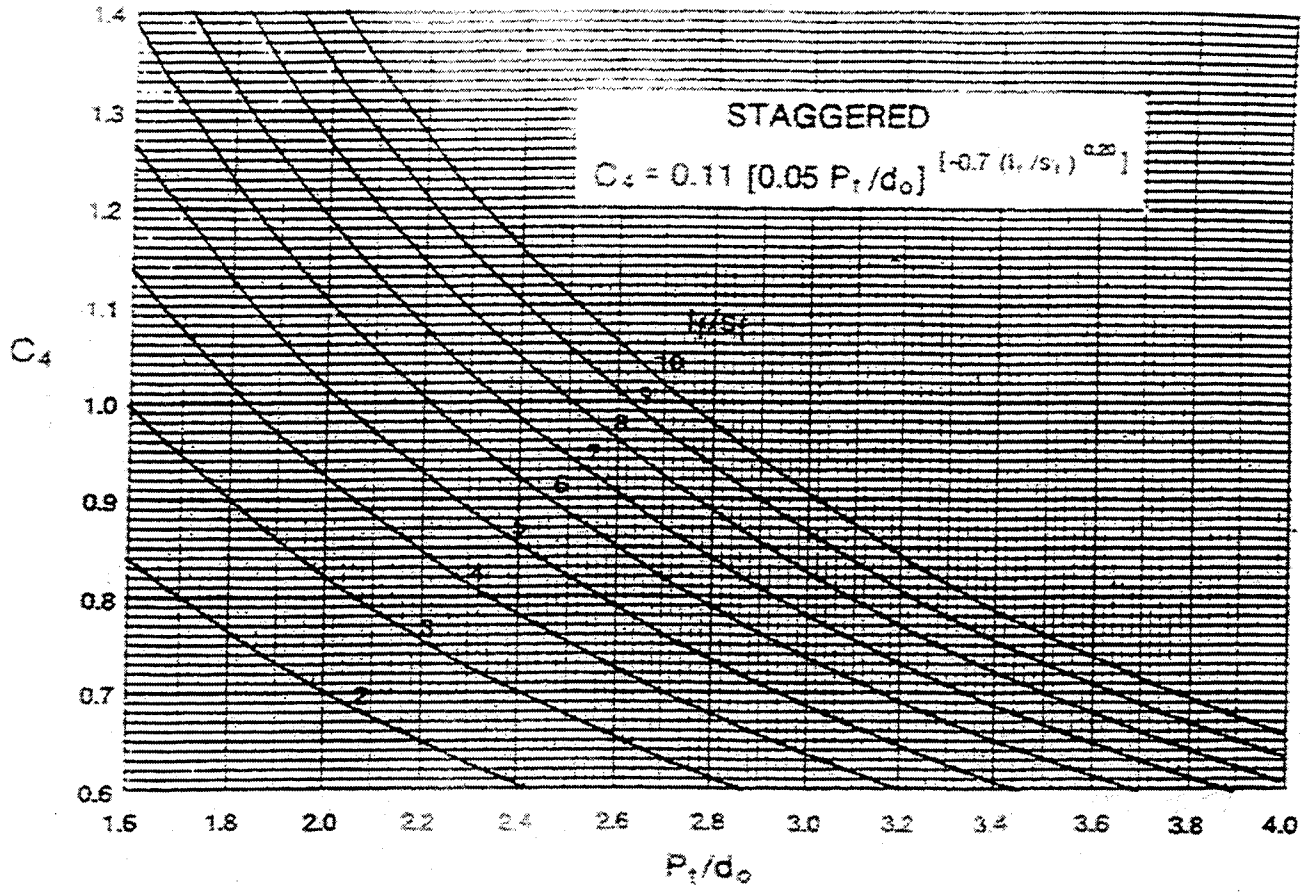
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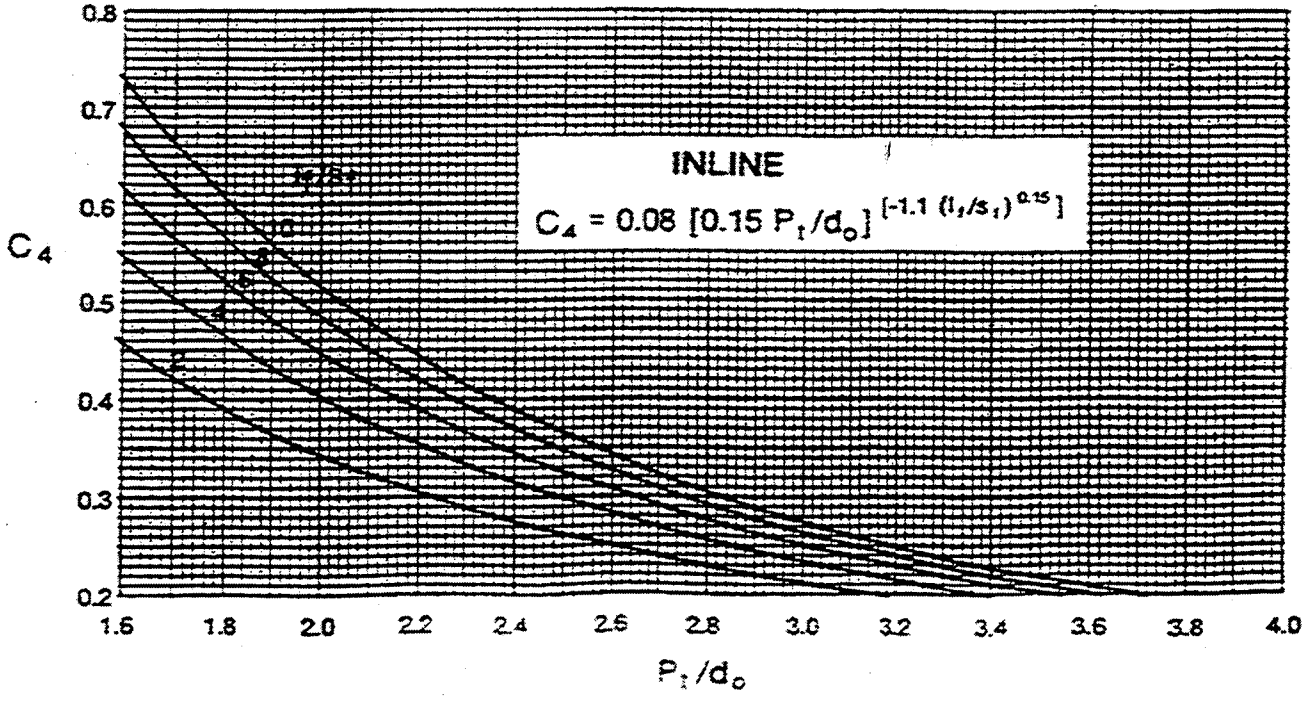
SOLIDFIN® HF

C_6





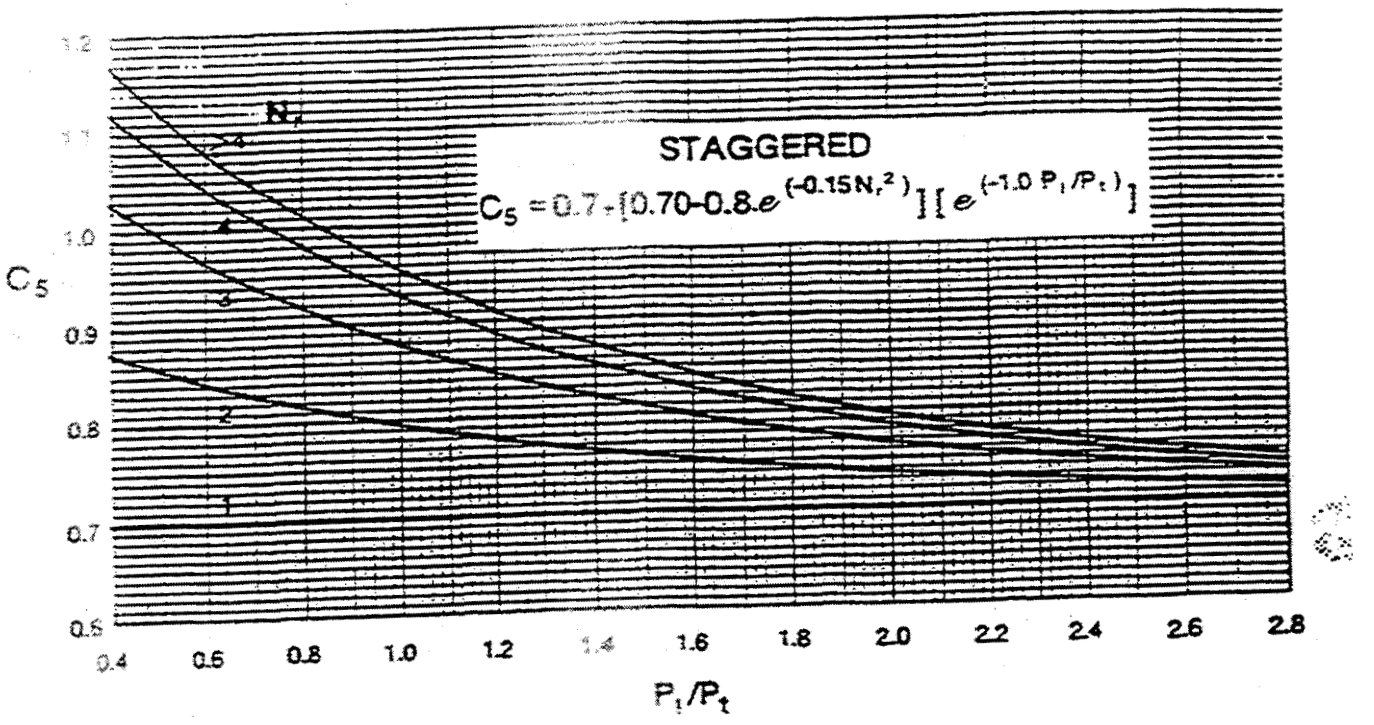

STAGGERED
$C_4 = 0.11\ [0.05\ P_t/d_o]^{[-0.7\ (l_r/s_t)^{0.20}]}$
l_r/s_t
10
8
6
4
3
2
C_4
1.4
1.3
1.2
1.1
1.0
0.9
0.8
0.7
0.6
1.6
1.8
2.0
2.2
2.4
2.6
2.8
3.0
3.2
3.4
3.6
3.8
4.0
P_t/d_o

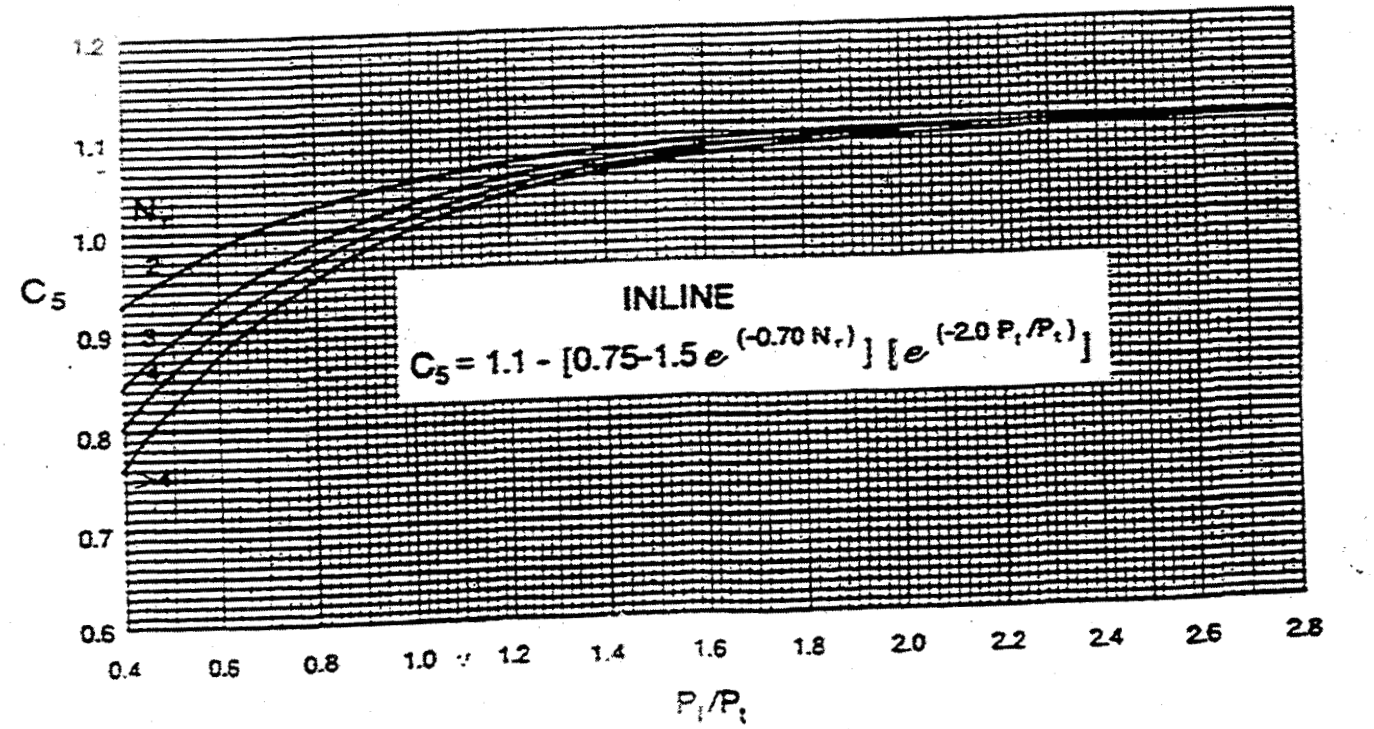

INLINE
$C_4 = 0.08\ [0.15\ P_t/d_o]^{[-1.1\ (l_r/s_t)^{0.15}]}$
l_r/s_t
10
8
6
4
2
C_4
0.8
0.7
0.6
0.5
0.4
0.3
0.2
1.6
1.8
2.0
2.2
2.4
2.6
2.8
3.0
3.2
3.4
3.6
3.8
4.0
P_t/d_o

ESCOA

SOLIDFIN® HF

C_5





ESCOA®

FLUE GAS RADIATION

A_b = bare tube outside surface area, ft²/ft

A_o = total outside surface area, ft²/ft

d_o = outside diameter of tube, in

h_r = average outside radiation heat transfer coefficient, Btu/hr ft² F

L = mean radiating length, ft
$L = (L/d_o)\, d_o$

P_l = longitudinal tube pitch, in

P_t = transverse tube pitch, in

p = total partial pressure of CO_2 and H_2O, atm. if unknown, assume $p = 0.22$

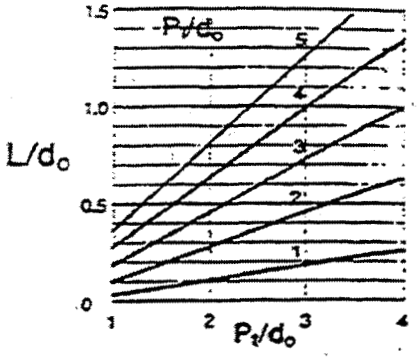
T_b = average flue gas temperature, F

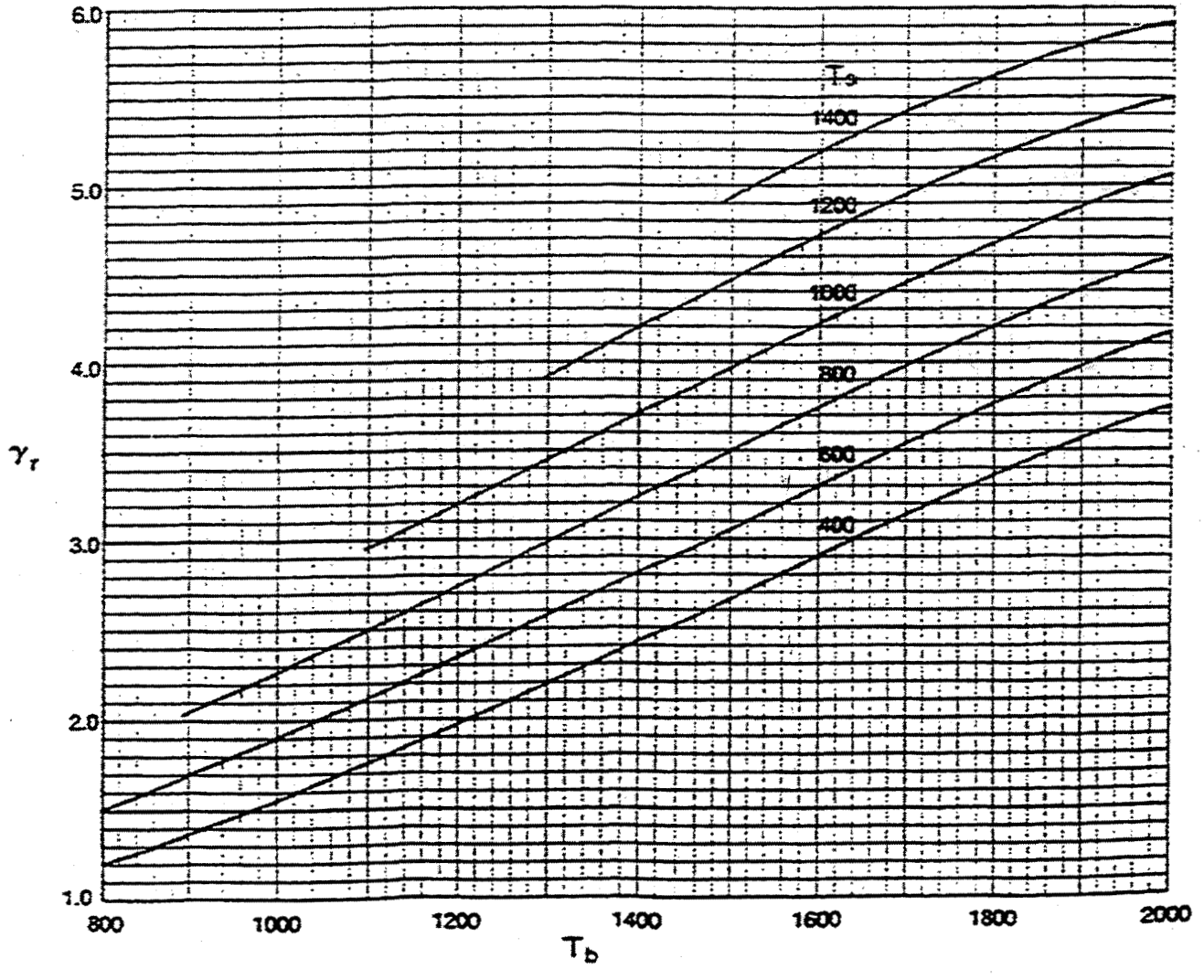
T_i = average inside fluid temperature, F

T_s = average fin surface temperature, F if unknown, assume $T_s = T_i + 0.3\,(T_b - T_i)$

γ_r = outside radiation factor, Btu/hr ft² F

Outside Radiation Coefficient
Staggered and Inline Layouts
$h_r = 2.2\ \gamma_r\ (pL)^{0.50}\ (A_s/A_o)^{0.75}$





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Training Services

Flame Temperature

Section Outline

- Combustion Basics
- Combustion Chemistry
- Flame Speed
- Ignition Temperature
- Excess Air Effects
- Air Preheat Effects
- Fuel Effects
- Flue Gas Oxygen vs. Excess Air

Combustion Basics

Three mandatory elements for combustion

- Fuel
- Air
- Ignition Source

FH 2004/5-3

Combustion Chemistry

- Carbon Combustion

  $C + O_2 + 3.75\ N_2 \rightarrow CO_2 + 3.75\ N_2$ 14,100 Btu/lb

- Hydrogen Combustion

  $2\ H_2 + O_2 + 3.75\ N_2 \rightarrow 2H_2O + 3.75\ N_2$ 51,600 Btu/lb

- Sulfur Combustion

  $S + O_2 + 3.75\ N_2 \rightarrow SO_2 + 3.75\ N2$ 4,000 Btu/lb

FH 2004/5-4

Air Requirements
Example Calculation for 1 x 10^6 Btu/h

- **Hydrogen**

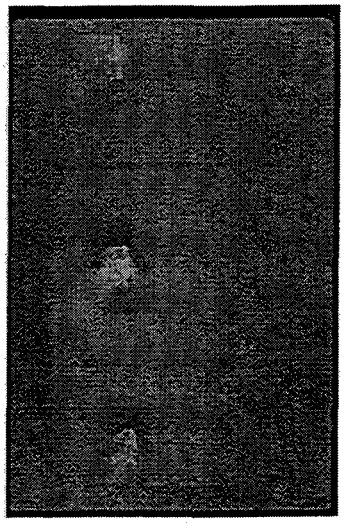
$$\text{Fuel Required} = \frac{1{,}000{,}000}{275} = 3636 \text{ sft}^3/\text{h}$$

$$3636\ H_2 + 1818\ O_2 + 6836\ N_2 \rightarrow 3636\ H_2O + 6836\ N_2$$

Air Required = 1818 + 6836 = 8,654 sft^3/h

Hydrogen Flame

- **Blue Hydrogen Flame**
- **Burns Quickly**
- **Low Visible Light**



Air Requirements
Example Calculation for 1 x 10^6 Btu/h

- Methane (CH_4)

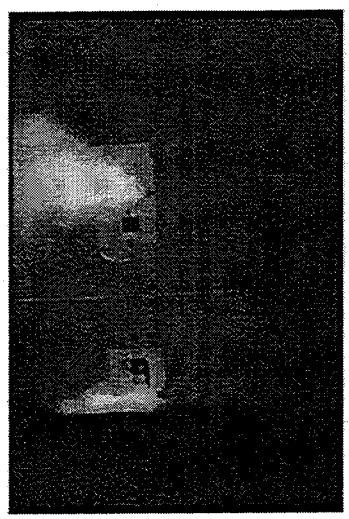
$$\text{Fuel Required} = \frac{1{,}000{,}000}{911} = 1098 \ \text{sft}^3/\text{h}$$

$$1098\ CH_4 + 2196\ O_2 + 8257\ N_2 \rightarrow 1098\ CO_2 + 2196\ H_2O + 8257\ N_2$$

Air Required = 2196 + 8257 = 10,453 sft^3/h

Refinery Fuel

- Fuel Composition Change
- Haze-Unburned Fuel
- Burners Need Air
- Yellow Flame is carbon burning



Flame Speed

Methane	1.5 ft/sec	(0.46 m/sec)
Ethane	2.3	(0.70)
Propane	2.8	(0.85)
Butane	2.9	(0.88)
Hydrogen	9.3	(2.83)
Carbon Monoxide	1.7	(0.52)

Ignition Temperature

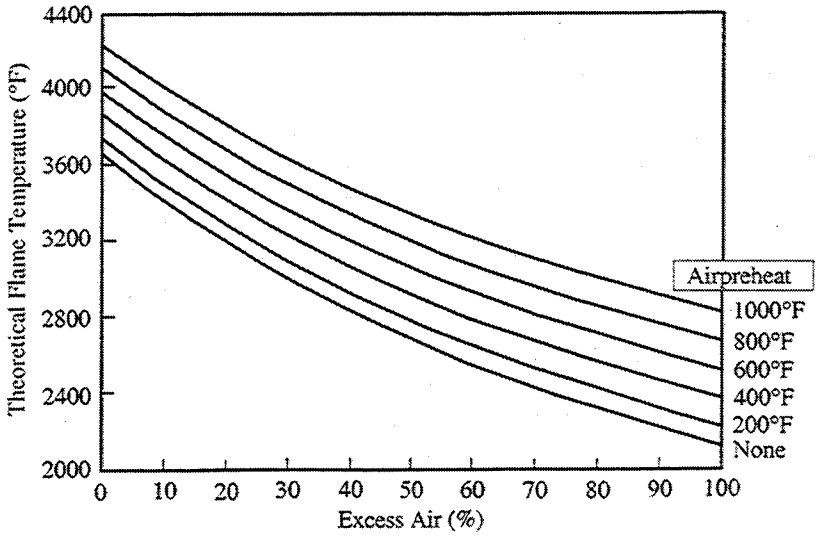
Methane	1170°F	632°C
Ethane	882°F	472°C
Propane	871°F	466°C
Butane	896°F	480°C
Hydrogen	1062°F	572°C
Carbon Monoxide	1128°F	609°C

Flame Temperature

- Excess Air
- Air Preheat

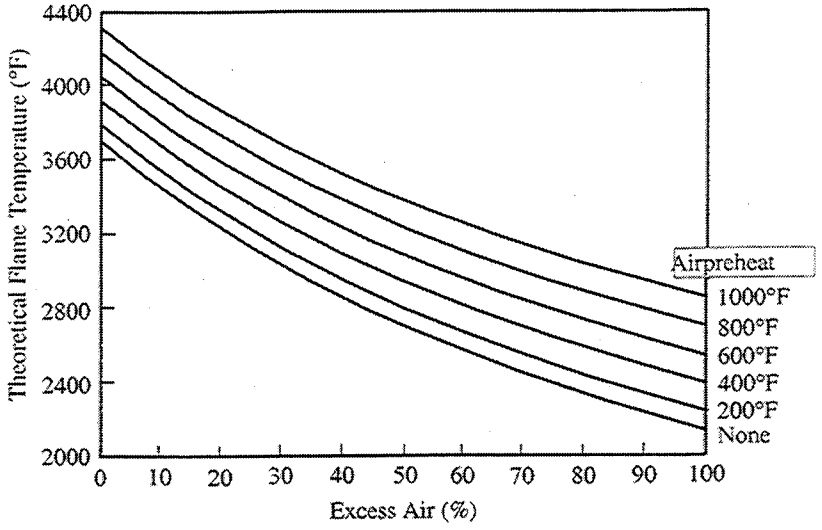
FH 2004/5-11

Flame Temperature - Natural Gas



FH 2004/5-12

Flame Temperature - Fuel Oil



FH 2004/S-13

Excess Air Effects

- **Increasing excess air will DECREASE the flame temperature**

FH 2004/S-14

Air Preheat Effects

- Increasing air preheat temperature will INCREASE the flame temperature

Fuel Effects

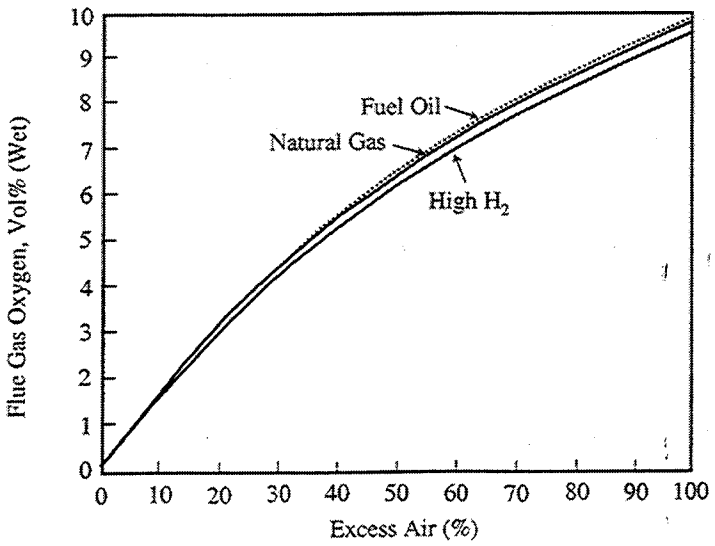
- Hydrogen Content
  - Increasing hydrogen content INCREASES the flame temperature
- Heating Value Change
  - Decreasing the fuel gas heating value slightly INCREASES the flame temperature

Flame Effects

Fuel	High H2 @ 15% E.A.	Natural Gas @ 15%E.A	Fuel Oil @ 25%E.A
Flue gas rate difference	base	121%	133%
Flame temp, °F	3542	3303	3181
Heat Release	base	105.2%	108.4%

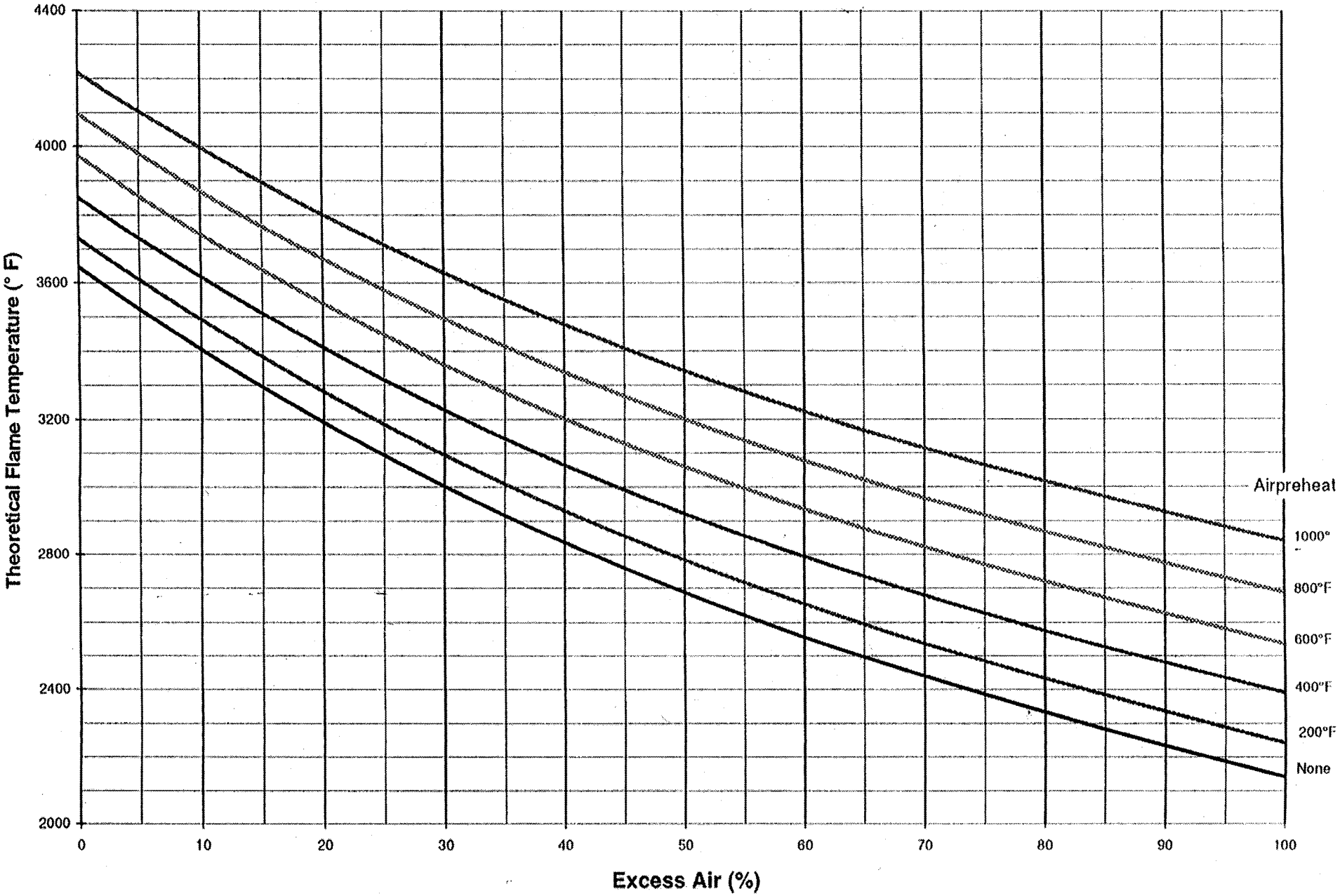
FH 2004/5-17

Flue Gas Oxygen vs. Excess Air

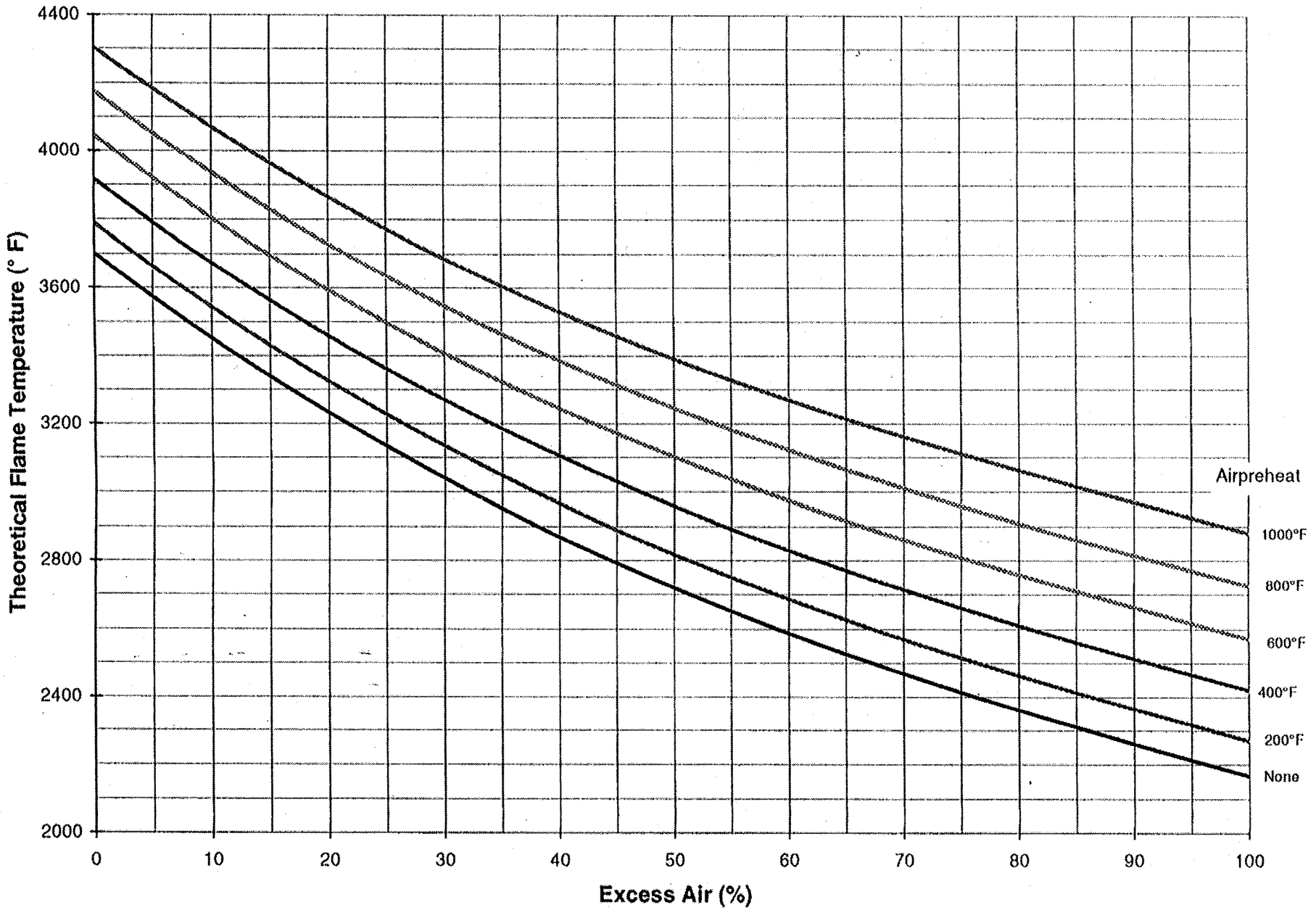


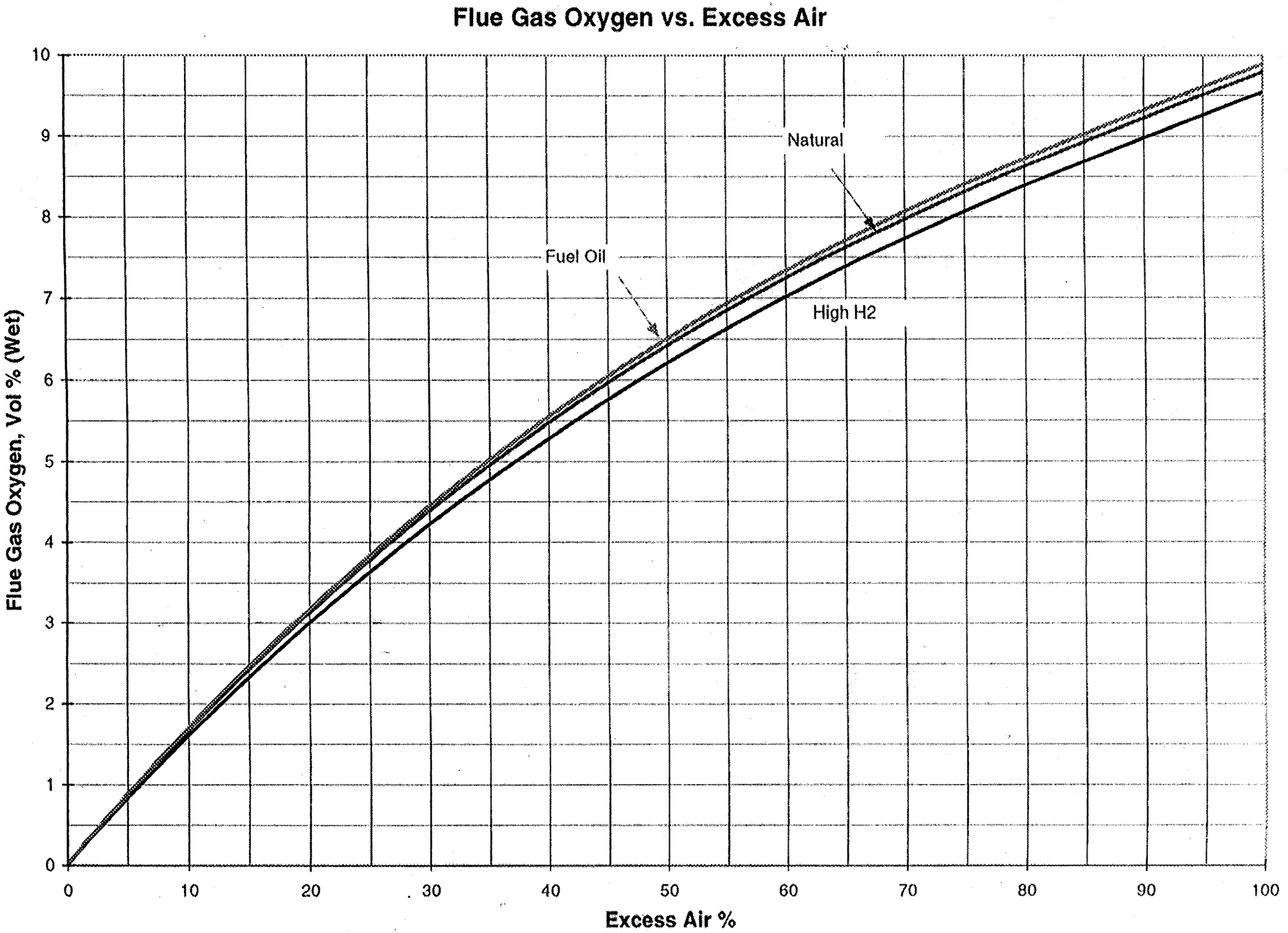
FH 2004/5-18

Flame Temperature - Natural Gas



Flame Temperature - Fuel Oil




Flue Gas Oxygen vs. Excess Air
Flue Gas Oxygen, Vol % (Wet)
Excess Air %
Natural
Fuel Oil
High H2
0
1
2
3
4
5
6
7
8
9
10
0
10
20
30
40
50
60
70
80
90
100

Combustion Theory

What is combustion? There are three(3) elements of combustion - fuel, air and a source of ignition. Burning or combustion is a rapid chemical reaction that releases heat and light. The reactants in the reaction are a combustible material and oxygen. The oxygen normally comes from air which is about 21% oxygen and 78% nitrogen by volume.

Combustible materials contained in most fuels are carbon, hydrogen and sometimes sulphur. Therefore, the combustion of a mixture of air and fuel combines oxygen (O_2) with carbon (C), hydrogen (H_2) or sulphur (S). Examples of the chemical reaction of the burning of carbon, hydrogen or sulphur in air are as follows:

$$C + O_2 + 3.75N_2 \rightarrow CO_2 + 3.75N_2$$

$$2\,H_2 + O_2 + 3.75N_2 \rightarrow 2\,H_2O + 3.75N_2$$

$$S + O_2 + 3.75N_2 \rightarrow SO_2 + 3.75N_2$$

The burning of the air and fuel produces carbon dioxide (CO_2), water (H_2O), sulphur dioxide (SO_2) and heat. The three products of combustion listed above am called chemical compounds. Chemical compounds are made up of molecules in which elements are combined in certain fixed proportions. To form carbon dioxide, each atom of carbon reacts with two(2) atoms of oxygen from the air. To form water vapor, two(2) atoms of hydrogen react with one(1) atom of oxygen from the air.

Most fuels are not composed of simple atoms of carbon, hydrogen, and sulphur. Instead the fuels contain various molecules or compounds that are combinations of these elements. For example, natural gas is approximately 95% methane (CH_4). Methane is a compound composed of one(1) atom of carbon, and four(4) atoms of hydrogen. An equation that describes the combustion of methane in air is shown below.

$$CH_4 + 2\,O_2 + 7.5\,N_2 \rightarrow CO_2 + 2\,H_2O + 7.5\,N_2$$

Combustion Theory

Note that in this case, one(1) atom of carbon combines with two(2) atoms of oxygen to form carbon dioxide, two(2) atoms of hydrogen combine with one(1) atom of oxygen to form one(1)

molecule of water but since four(4) atoms of hydrogen are present in the fuel, two(2) molecules of water are formed.

The heat that is given off in the combustion process is excess energy which new molecules are forced to liberate because of their internal makeup. The excess energy that is given off during the combustion reaction is normally seen as a temperature difference between the reactants in the reaction.

Fuel 1 - 100% CH

$$\text{Fuel Required} = \frac{1{,}000{,}000}{911} = 1098 scfh$$

$$1098\ CH_4 + 2196\ O_2 + 8257\ N_2 \rightarrow 1098\ CO_2 + 2196\ H_2O + 8257\ N_2$$

$$\text{Air} = 2196 + 8257 = 10{,}453 \text{scfh}$$

Fuel 2- 100% H_2

$$\text{Fuel Required} = \frac{1{,}000{,}000}{275} = 3636 scfh$$

$$3636\ H_2 + 1818\ O_2 + 6836\ N_2 \rightarrow 3636\ H_2O + 6836\ N_2$$

$$\text{Air} = 1818 + 6836 = 8654 \text{scfh}$$

Fuel Effects

Many operators report that the heater problems began when the heater fuel gas source was switched from high hydrogen fuel to refinery fuel. This is a very good observation overlooked by many people.

Hydrogen fuel requires less oxygen to burn. It burns with a hotter flame temperature and a faster flame speed. Hydrogen has a flame speed of approximately 7 fps versus 1 fps for most hydrocarbons.

The following table is based on an all radiant, single fired, xylene isomerization heater at 10,000 Btu/h-ft2 flux and a 920°F TWT.

Fuel	High H2 @15% E.A.	Natural Gas @ 15% E.A.	Fuel Oil @ 25% E.A.
Flue gas rate difference	**base**	**121%**	**133%**
Flame temp, °F	**3542**	**3303**	**3181**
Heat Release	**base**	**105.2%**	**108.4%**

Radiant heat transfer is T^4flame-T^4tube. Hotter flame temperatures improve radiant efficiency and reduce the fuel firing rate. Hydrogen takes 18-23% less air to burn than hydrocarbon fuels for an equivalent heat release. The higher flame temperature and less air requirements for hydrogen fuel reduce the flue gas rate dramatically.

The pressure drop across the convection section ratios directly with the square of the flue gas rate. This means the pressure drop is 46% greater for natural fuel than the high hydrogen fuel. It would be 77% higher for oil fuel. This is a major problem when the heater has a stack limitation.

The high flame speed of hydrogen results in rapid combustion. This eliminates the haze problems and afterburning problems.

Fuel Effects

The high hydrogen fuel is Platformer off gas with 92.3 % H2 with a MW of 4.2 and a heating value of 379 Btu/scf. The natural gas fuel has a MW of 19.0 and a heating value of 942 Btu/scf. The fuel oil is a 20 API gravity oil with a heating value of 17935 Btu/lb.

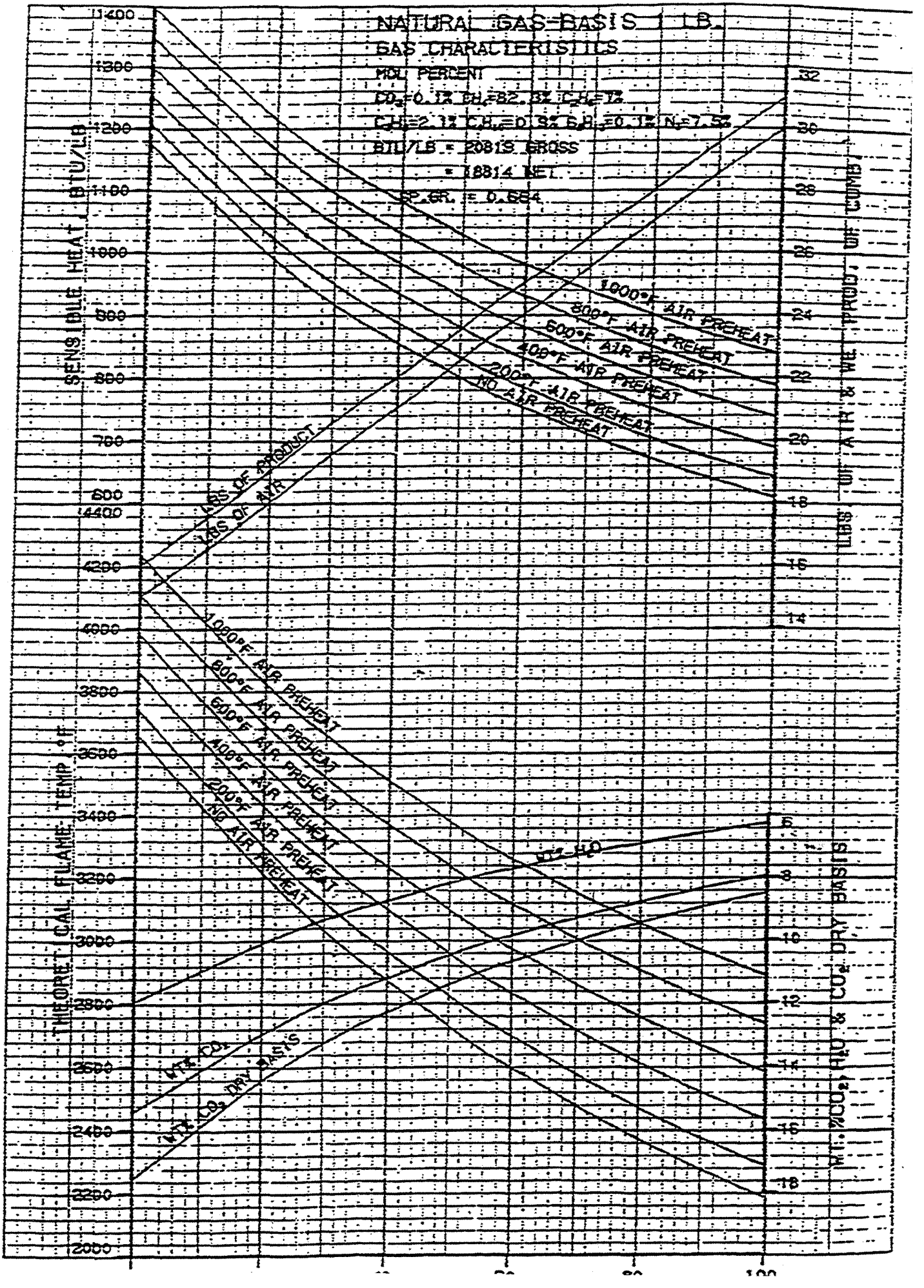
It is interesting to note that natural gas has a net heating value of 18814 Btu/lb while the fuel oil has 17935 Btu/lb. The heating values are similar because the carbon/hydrogen ratios are similar. The high hydrogen fuel has a heating value of 34159 Btu/lb. Hydrogen has a higher Btu/lb heat release than carbon.

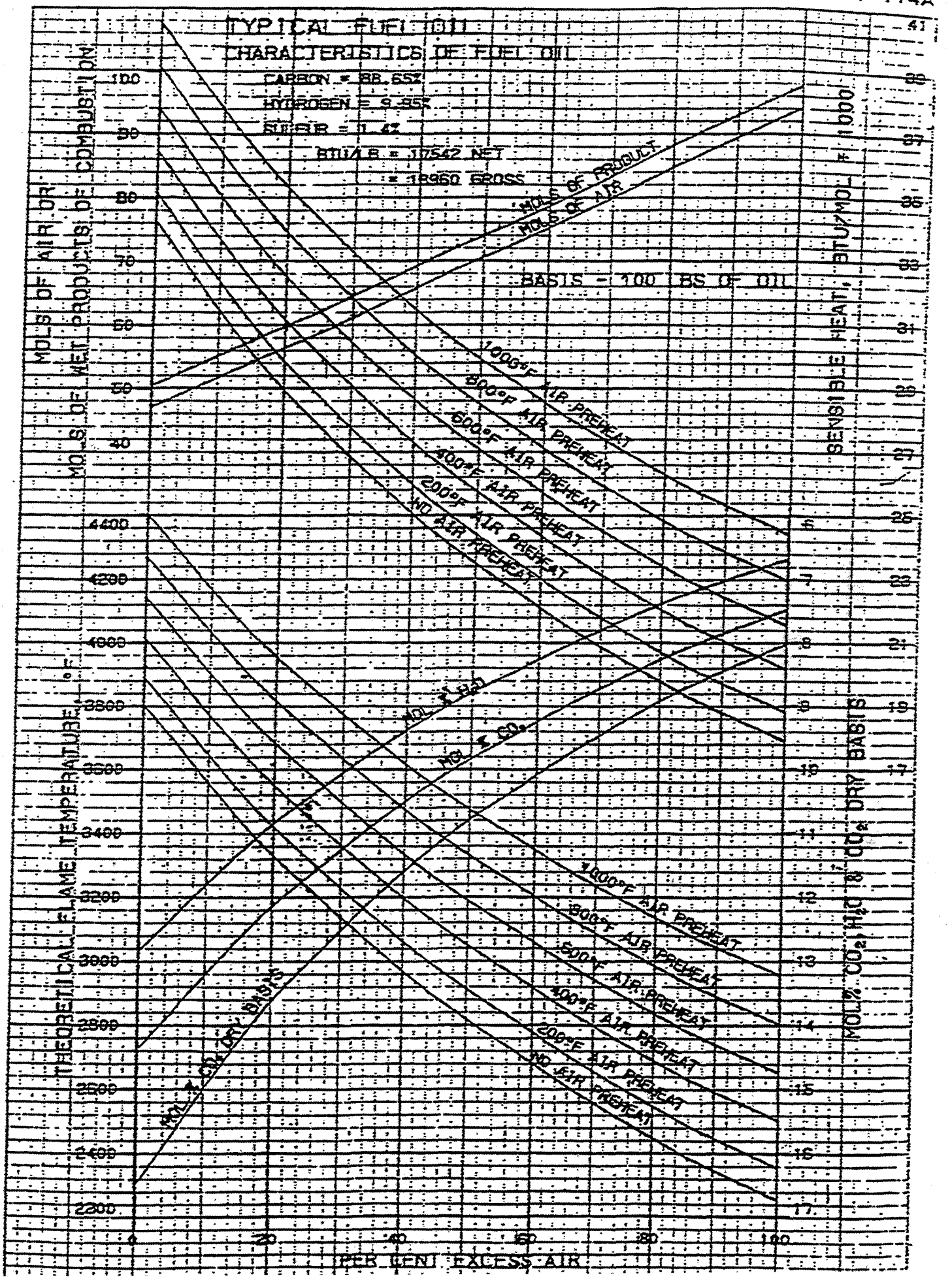
Air Preheat Effects

Increasing the combustion air temperature from 60°F to 400°F for an all radiant heater has the following effects:

1. The flame temperature increases by 200°F
2. The BWT barely changes by a few degrees
3. The fuel rate decreases by 10-12%

On a radiant-convection heater, the duty shift from the convection section to the radiant section. The radiant flux rate increases by 7-10%. The hotter flame temperature increases the radiant efficiency. Less fuel firing decreases the flue gas rate and the heat pickup in the convection section.


NATURAL GAS-BASIS 1 LB.
GAS CHARACTERISTICS
MOL PERCENT
CO_2=0.1% CH_4=82.3% C_2H_6=1%
C_3H_8=2.1% C_4H_{10}=0.9% C_5H_{12}=0.1% N_2=7.5%
BTU/LB = 20813 GROSS
= 18814 NET
SP. GR. = 0.664
1000°F AIR PREHEAT
800°F AIR PREHEAT
600°F AIR PREHEAT
400°F AIR PREHEAT
200°F AIR PREHEAT
NO AIR PREHEAT
LBS OF PRODUCT
LBS OF AIR
SENSIBLE HEAT BTU/LB
THEORETICAL FLAME TEMP °F
LBS AIR & WT. PROD. OF COMB.
WT. % CO_2, H_2O & CO_2 DRY BASIS
H_2O
CO_2
WT. % CO_2 DRY BASIS


TYPICAL FUEL OIL
CHARACTERISTICS OF FUEL OIL
CARBON = 88.65%
HYDROGEN = 9.85%
SULFUR = 1.4%
BTU/LB = 17542 NET
= 18960 GROSS
BASIS - 100 LBS OF OIL
MOLS OF PRODUCT
MOLS OF AIR
1000°F AIR PREHEAT
800°F AIR PREHEAT
600°F AIR PREHEAT
400°F AIR PREHEAT
200°F AIR PREHEAT
NO AIR PREHEAT
MOL % H2O
MOL % CO2
MOL % CO2 DRY BASIS
MOLS OF AIR OR MOLS OF WET PRODUCTS OF COMBUSTION
THEORETICAL FLAME TEMPERATURE °F
SENSIBLE HEAT, BTU/MOL °F - 1000
MOL% CO2, H2O & O2 DRY BASIS
PER CENT EXCESS AIR

Training Services

Efficiency

FH 2004/6-1

Section Outline

- Efficiency
- Fuels Program
- Sample Problems

FH 2004/6-2

Efficiency

Fuel Fired = Process Absorption + Loss to Stack + Radiation Losses

$$\text{Fuel Efficiency} = \frac{\text{Process Absorption}}{\text{Fuel Fired}}$$

Fuels Program

- Microsoft EXCEL Spreadsheet software
- Fuel Gas and Fuel Oil versions
- Calculates
  - Theoretical flame temperature
  - Flue gas enthalpy vs. temperature
  - Flue gas dewpoint temperature
  - Radiant efficiency
  - Heater efficiency
  - Air requirements for combustion
  - Corrects for air preheat, humidity in air

Sample Problem (1)

The stack O_2 is 3.2 mol-% (wet), the fuel is natural gas, and the stack temperature is 400°F. What is the fuel efficiency?

Fuel composition is:	**Nitrogen**	**7.5 mol %**
	Carbon dioxide	**0.1**
	Methane	**82.3**
	Ethane	**7.0**
	Propane	**2.1**
	n-Butane	**0.9**
	n-Pentane	**0.1**

Sample Problem (1)
(continued)

The stack O_2 is 3.2%, the fuel is natural gas, and the stack temperature is 400°F. What is the fuel efficiency?

- **3.2 O_2 is 20% excess air**
- **400°F stack is 9.3% heat loss**
- **Assume 2% radiation loss**
- **Fuel efficiency: 100 - 2.0 - 9.3 = 88.7%**

Sample Problem (2)

The stack O_2 is 5.0 mol-% (wet), the fuel is fuel oil, and the stack temperature is 500°F. What is the fuel efficiency?

Fuel oil composition is:

Carbon	88.72 wt %
Hydrogen	11.08
Sulfur	0.20
Nitrogen	0
Ash	0
Atomizing Steam Rate	0.3 lb steam/lb oil
Steam Pressure	139 psig

Sample Problem (2)
(continued)

The stack O_2 is 5.0%, the fuel is fuel oil, and the stack temperature is 500°F. What is the fuel efficiency?

- **5.0% O_2 is 34.8% excess air**
- **500°F stack is 13.2% heat loss**
- **Assume 2% radiation loss**
- **Fuel efficiency: 100 - 2.0 - 13.2 = 84.8%**

Sample Problem (3)

The stack O_2 is 2.4 vol-% (wet), the fuel is high hydrogen fuel gas, and the stack temperature is 300°F. What is the fuel efficiency?

Fuel composition is:	**Hydrogen**	**92.3 mol %**
	Methane	**2.9**
	Ethane	**2.7**
	Propane	**1.4**
	n-Butane	**0.4**
	n-Pentane	**0.3**

Sample Problem (3)
(continued)

The stack O_2 is 2.4%, the fuel is high hydrogen fuel gas, and the stack temperature is 300°F. What is the fuel efficiency?

- **2.4% O_2 is 15% excess air**
- **300°F stack is 5.7% heat loss**
- **Assume 2% radiation loss**
- **Fuel efficiency: 100 - 2.0 - 5.7 = 92.3%**

Fuel Efficiency Problems

Fuel Fired = Process Absorption + Heat to Stack + Radiation Losses (2%)

Fuel Efficiency = Process Absorption / Fuel Fired

Problem 1

The Stack 0_2 is 3.2%, the fuel is natural gas, and the stack temperature is 400°F. What is the fuel efficiency?

1. 3.2% 0_2 is 20 % among air.
2. 400°F stack is 9.2 % heat loss
3. Assume 2 % radiation loss
4. Fuel Efficiency = 100 - 9.2 - 2.0 = 88.8

Problem 2

The stack 0_2 is 5.0%, the fuel is fuel oil, and the stack is 500°F. What is the fuel efficiency?

1. 5.0% 0_2 is 35 % excess air
2. 500°F Stack is 13.0 % heat loss
3. Assume 2 % radiation loss
4. Fuel Efficiency = 100 - 13.0 - 2.0 = 85.0

Problem 3

The stack 0_2 is 2.4%, the fuel is high hydrogen gas, and the stack temperature is 300°F. What is the fuel efficiency?

1. 2.4% 0_2 is 15 % excess air
2. 300°F stack is 5.7 % heat loss
3. Assume 2 % radiation loss
4. Fuel Efficiency = 100 - 5.7 - 2.0 = 92.3

NATURAL GAS

FUEL GAS

ITEM	MOLE PERCENT
N2	7.50
C1	82.30
C2	7.00
CO2	0.10
C3	2.10
NC4	0.90
NC5	0.10
TOTAL	100.00

MOLECULAR WEIGHT OF FUEL GAS IS 18.97

INLET CONDITIONS

AIR HUMIDITY = 0.0110 LB/LB DRY AIR
AIR TEMP. = 60.0 F
FUEL TEMP. = 100.0 F
MAX. EXCESS AIR = 15.00%
EXCESS AIR INCREMENT = 15.00%
AIR HUMIDITY = 100.%
AMBIENT AIR TEMP = 60.0 F

	BTU/LB	BTU/SCF	BTU/MOLE
LHV	18814.8	941.8	356982.8
HHV	20819.3	1042.1	394959.6

THEORETICAL LBS OF DRY AIR REQUIRED PER LB OF FUEL = 14.97
THEORETICAL LBS OF HUMID AIR REQUIRED PER LB OF FUEL = 15.14

UOP PROCESS DIVISION - UOP INC
FUEL GAS EVALUATION P714.1A

NATURAL GAS

15.0 PER CENT EXCESS AIR

	HEAT CONTENT BTU/LB	HEAT CONTENT BTU/MOLE	PERCENT EXTRACTED
.0	0.000	0.000	100.000
.0	10.533	292.059	98.970
.0	37.061	1027.624	96.377
.0	63.901	1771.843	93.754
.0	91.086	2525.626	91.097
.0	118.642	3289.697	88.403
.0	146.589	4064.588	85.672
.0	174.936	4850.641	82.901
.0	203.697	5648.074	80.090
.0	232.868	6456.922	77.239
.0	262.440	7277.129	74.347
.0	292.431	8108.484	71.417
.0	322.806	8950.719	68.448
.0	353.560	9803.457	65.442
.0	384.676	10666.242	62.400
.0	416.137	11538.582	59.325
.0	447.922	12419.910	56.218
.0	480.011	13309.672	53.082
.0	512.382	14207.230	49.918
.0	545.015	15112.086	46.728
.0	577.888	16023.570	43.515
.0	610.982	16941.211	40.280
.0	644.280	17864.480	37.026
.0	677.764	18792.930	33.753
.0	711.422	19726.195	30.463
.0	745.245	20664.027	27.157
.0	779.225	21606.215	23.836
.0	813.360	22552.719	20.499
.0	847.653	23503.574	17.147
.0	882.112	24459.043	13.779
.0	916.749	25419.461	10.394
.0	951.565	26385.398	6.989
.0	986.647	27357.578	3.562
.0	1021.969	28336.969	0.109
.1	1023.085	28367.926	0.000

High Hydrogen Fuel

FUEL GAS PROGRAM OUTPUT

FUEL GAS

ITEM	MOLE PERCENT
H2	92.30
C1	2.90
C2	2.70
C3	1.40
NC4	0.40
NC5	0.30
TOTAL	100.00

MOLECULAR WEIGHT OF FUEL GAS IS 4.20

INLET CONDITIONS

AIR HUMIDITY = 0.0110 LB/LB DRY AIR
AIR TEMP. = 60.0 F
FUEL TEMP. = 100.0 F
MAX. EXCESS AIR = 15.00%
EXCESS AIR INCREMENT = 15.00%
AIR HUMIDITY = 100.%
AMBIENT AIR TEMP = 60.0 F

	BTU/LB	BTU/SCF	BTU/MOLE
LHV	34159.2	376.6	143557.0
HHV	39412.8	457.0	165635.0

THEORETICAL LBS OF DRY AIR REQUIRED PER LB OF FUEL = 24.05
THEORETICAL LBS OF HUMID AIR REQUIRED PER LB OF FUEL = 24.32

UOP PROCESS DIVISION - UOP INC
FUEL GAS EVALUATION P714.1A

High Hydrogen Fuel

FUEL GAS PROGRAM OUTPUT

15.0 PER CENT EXCESS AIR

TEMP	HEAT CONTENT BTU/LB	BTU/MOLE	PERCENT EXTRACTED
60.0	0.000	0.000	100.000
100.0	11.216	291.821	99.051
200.0	39.407	1025.290	96.665
300.0	67.861	1765.617	94.257
400.0	96.627	2514.076	91.823
500.0	125.747	3271.703	89.359
600.0	155.249	4039.310	86.862
700.0	185.157	4817.461	84.331
800.0	215.486	5606.562	81.765
900.0	246.243	6406.805	79.162
1000.0	277.430	7218.246	76.523
1100.0	309.044	8040.777	73.847
1200.0	341.077	8874.223	71.137
1300.0	373.517	9718.262	68.391
1400.0	406.349	10572.488	65.613
1500.0	439.555	11436.453	62.803
1600.0	473.116	12309.645	59.965
1700.0	507.011	13191.543	57.094
1800.0	541.219	14081.574	54.200
1900.0	575.722	14979.289	51.280
2000.0	610.501	15884.160	48.337
2100.0	645.537	16795.725	45.372
2200.0	680.817	17713.660	42.386
2300.0	716.330	18637.648	39.381
2400.0	752.069	19567.512	36.357
2500.0	788.034	20503.254	33.313
2600.0	824.227	21444.926	30.250
2700.0	860.658	22392.820	27.167
2800.0	897.345	23347.344	24.063
2900.0	934.314	24309.199	20.934
3000.0	971.597	25279.242	17.779
3100.0	1009.237	26258.570	14.594
3200.0	1047.287	27248.566	11.374
3300.0	1085.812	28250.922	8.114
3400.0	1124.884	29267.504	4.807
3500.0	1164.594	30300.687	1.447
3542.5	1181.693	30745.570	0.000

UOP PROCESS DIVISION - UOP INC
FUEL GAS EVALUATION P714.1A

High Hydrogen Fuel

% STACK LOSS VS FLUE GAS TEMPERATURE
AT 60. F AIR TEMPERATURE

(THIS TABLE DOES NOT ALLOW FOR RADIATION LOSSES)

FUEL GAS PROGRAM OUTPUT

EXC. AIR	FLUE GAS TEMPERATURE, F																				
	60.	100.	200.	300.	400.	500.	600.	700.	800.	900.	1000.	1100.	1200.	1300.	1400.	1500.	1600.	1700.	1800.	1900.	2000.
0	0.0	0.8	3.0	5.1	7.3	9.5	11.7	14.0	16.3	18.6	20.9	23.3	25.7	28.2	30.7	33.2	35.7	38.3	40.9	43.5	46.1
5	0.0	0.9	3.1	5.3	7.6	9.9	12.2	14.5	16.9	19.3	21.8	24.3	26.8	29.3	31.9	34.5	37.2	39.8	42.5	45.2	48.0
10	0.0	0.9	3.2	5.5	7.9	10.3	12.7	15.1	17.6	20.1	22.6	25.2	27.8	30.5	33.2	35.9	38.6	41.4	44.2	47.0	49.8
15	0.0	0.9	3.3	5.7	8.2	10.6	13.1	15.7	18.2	20.8	23.5	26.2	28.9	31.6	34.4	37.2	40.0	42.9	45.8	48.7	51.7
20	0.0	1.0	3.5	6.0	8.5	11.0	13.6	16.2	18.9	21.6	24.3	27.1	29.9	32.7	35.6	38.5	41.5	44.4	47.4	50.5	53.5
25	0.0	1.0	3.6	6.2	8.8	11.4	14.1	16.8	19.6	22.3	25.2	28.0	30.9	33.9	36.9	39.9	42.9	46.0	49.1	52.2	55.4
30	0.0	1.1	3.7	6.4	9.1	11.8	14.6	17.4	20.2	23.1	26.0	29.0	32.0	35.0	38.1	41.2	44.3	47.5	50.7	53.9	57.2
35	0.0	1.1	3.8	6.6	9.4	12.2	15.0	17.9	20.9	23.9	26.9	29.9	33.0	36.2	39.3	42.5	45.8	49.1	52.4	55.7	59.0
40	0.0	1.1	3.9	6.8	9.7	12.6	15.5	18.5	21.5	24.6	27.7	30.9	34.1	37.3	40.6	43.9	47.2	50.6	54.0	57.4	60.9
45	0.0	1.2	4.1	7.0	10.0	13.0	16.0	19.1	22.2	25.4	28.6	31.8	35.1	38.4	41.8	45.2	48.7	52.1	55.6	59.2	62.7
50	0.0	1.2	4.2	7.2	10.3	13.3	16.5	19.6	22.9	26.1	29.4	32.8	36.1	39.6	43.0	46.5	50.1	53.7	57.3	60.9	64.6
55	0.0	1.2	4.3	7.4	10.6	13.7	16.9	20.2	23.5	26.9	30.3	33.7	37.2	40.7	44.3	47.9	51.5	55.2	58.9	62.7	66.4
60	0.0	1.3	4.4	7.6	10.8	14.1	17.4	20.8	24.2	27.6	31.1	34.6	38.2	41.8	45.5	49.2	53.0	56.7	60.6	64.4	68.3
65	0.0	1.3	4.5	7.8	11.1	14.5	17.9	21.3	24.8	28.4	32.0	35.6	39.3	43.0	46.7	50.6	54.4	58.3	62.2	66.1	70.1
70	0.0	1.3	4.7	8.0	11.4	14.9	18.4	21.9	25.5	29.1	32.8	36.5	40.3	44.1	48.0	51.9	55.8	59.8	63.8	67.9	72.0
75	0.0	1.4	4.8	8.2	11.7	15.3	18.9	22.5	26.1	29.9	33.6	37.5	41.3	45.3	49.2	53.2	57.3	61.4	65.5	69.6	73.8
80	0.0	1.4	4.9	8.5	12.0	15.7	19.3	23.0	26.8	30.6	34.5	38.4	42.4	46.4	50.5	54.6	58.7	62.9	67.1	71.4	75.7
85	0.0	1.4	5.0	8.7	12.3	16.0	19.8	23.6	27.5	31.4	35.5	39.4	43.4	47.5	51.7	55.9	60.1	64.4	68.8	73.1	77.5
90	0.0	1.5	5.2	8.9	12.6	16.4	20.3	24.2	28.1	32.1	36.2	40.3	44.5	48.7	52.9	57.2	61.6	66.0	70.4	74.9	79.3
95	0.0	1.5	5.3	9.1	12.9	16.8	20.8	24.7	28.8	32.9	37.0	41.2	45.5	49.8	54.2	58.6	63.0	67.5	72.0	76.6	81.2
100	0.0	1.5	5.4	9.3	13.2	17.2	21.2	25.3	29.4	33.6	37.9	42.2	46.5	50.9	55.4	59.9	64.5	69.0	73.7	78.5	83.0

UOP PROCESS DIVISION - UOP INC
FUEL GAS EVALUATION P714.1A

High Hydrogen Fuel

% STACK LOSS VS FLUE GAS TEMPERATURE
AT 400. F AIR TEMPERATURE

(THIS TABLE DOES NOT ALLOW FOR RADIATION LOSSES)

FUEL GAS PROGRAM OUTPUT

XC; R	FLUE GAS TEMPERATURE, F																				
	60.	100.	200.	300.	400.	500.	600.	700.	800.	900.	1000.	1100.	1200.	1300.	1400.	1500.	1600.	1700.	1800.	1900.	2000.
0	0.0	0.8	2.8	4.8	6.9	9.0	11.1	13.2	15.3	17.5	19.8	22.0	24.3	26.6	29.0	31.3	33.7	36.1	38.6	41.1	43.5
5	0.0	0.8	2.9	5.0	7.1	9.3	11.5	13.7	15.9	18.2	20.5	22.8	25.2	27.6	30.0	32.5	35.0	37.5	40.0	42.6	45.2
0	0.0	0.9	3.0	5.2	7.4	9.6	11.9	14.2	16.5	18.9	21.2	23.7	26.1	28.6	31.1	33.7	36.2	38.8	41.5	44.1	46.8
5	0.0	0.9	3.1	5.4	7.7	10.0	12.3	14.7	17.1	19.5	22.0	24.5	27.0	29.6	32.2	34.8	37.5	40.2	42.9	45.6	48.4
0	0.0	0.9	3.2	5.6	7.9	10.3	12.7	15.2	17.6	20.2	22.7	25.3	27.9	30.6	33.3	36.0	38.7	41.5	44.3	47.1	50.0
5	0.0	0.9	3.3	5.7	8.2	10.6	13.1	15.6	18.2	20.8	23.4	26.1	28.8	31.5	34.3	37.1	39.9	42.8	45.7	48.6	51.5
0	0.0	1.0	3.4	5.9	8.4	11.0	13.5	16.1	18.8	21.4	24.2	26.9	29.7	32.5	35.4	38.3	41.2	44.1	47.1	50.1	53.1
5	0.0	1.0	3.5	6.1	8.7	11.3	13.9	16.6	19.3	22.1	24.9	27.7	30.6	33.5	36.4	39.4	42.4	45.4	48.5	51.6	54.7
0	0.0	1.0	3.6	6.3	8.9	11.6	14.3	17.1	19.9	22.7	25.6	28.5	31.4	34.4	37.5	40.5	43.6	46.7	49.9	53.0	56.2
5	0.0	1.1	3.7	6.4	9.2	11.9	14.7	17.6	20.4	23.3	26.3	29.3	32.3	35.4	38.5	41.6	44.8	48.0	51.2	54.5	57.8
0	0.0	1.1	3.8	6.6	9.4	12.3	15.1	18.0	21.0	24.0	27.0	30.1	33.2	36.3	39.5	42.7	46.0	49.3	52.6	55.9	59.3
5	0.0	1.1	3.9	6.8	9.7	12.6	15.5	18.5	21.5	24.6	27.7	30.9	34.0	37.3	40.5	43.8	47.2	50.6	54.0	57.4	60.8
0	0.0	1.2	4.0	7.0	9.9	12.9	15.9	19.0	22.1	25.2	28.4	31.6	34.9	38.2	41.6	44.9	48.4	51.8	55.3	58.8	62.3
5	0.0	1.2	4.3	7.3	10.4	13.6	16.7	20.0	23.2	26.5	29.9	33.3	36.7	40.2	43.7	47.3	50.9	54.5	58.2	61.9	65.6
0	0.0	1.2	4.2	7.3	10.4	13.5	16.7	19.9	23.2	26.4	29.8	33.2	36.6	40.1	43.6	47.1	50.7	54.3	58.0	61.7	65.4
5	0.0	1.2	4.3	7.5	10.6	13.8	17.1	20.4	23.7	27.1	30.5	33.9	37.4	41.0	44.6	48.2	51.9	55.6	59.3	63.1	66.9
0	0.0	1.3	4.4	7.6	10.9	14.1	17.5	20.8	24.2	27.7	31.2	34.7	38.3	41.9	45.6	49.3	53.0	56.8	60.6	64.5	68.3
5	0.0	1.3	4.5	7.8	11.1	14.5	17.8	21.3	24.7	28.3	31.8	35.5	39.1	42.8	46.6	50.4	54.2	58.1	61.9	65.9	69.8
0	0.0	1.3	4.6	8.0	11.4	14.8	18.2	21.7	25.3	28.9	32.5	36.2	40.0	43.7	47.6	51.4	55.3	59.3	63.3	67.3	71.3
5	0.0	1.3	4.7	8.1	11.6	15.1	18.6	22.2	25.8	29.5	33.2	37.0	40.8	44.6	48.5	52.5	56.5	60.5	64.6	68.6	72.6
0	0.0	1.4	4.8	8.3	11.8	15.4	19.0	22.6	26.3	30.1	33.9	37.7	41.6	45.5	49.5	53.5	57.6	61.7	65.8	70.0	74.2

UOP PROCESS DIVISION - UOP INC
FUEL OIL EVALUATION P714.1D

FUEL OIL PROGRAM OUTPUT REPORT

INLET CONDITIONS

AIR HUMIDITY = 0.0110 LB/LB DRY AIR
AIR TEMP. = 60.0 F
FUEL TEMP. = 100.0 F
MAX. EXCESS AIR = 25.00%
EXCESS AIR INCREMENT = 25.00%
AIR HUMIDITY = 100.%
AMBIENT AIR TEMP. = 60.0 F

F U E L O I L D A T A

NET HEATING VALUE = 17934.6 BTU/LB OIL
GROSS HEATING VALUE = 19022.5 BTU/LB OIL

FUEL OIL ANALYSIS

ITEM	WEIGHT PERCENT
CARBON	88.72
HYDROGEN	11.08
OXYGEN	0.00
NITROGEN	0.00
SULFUR	0.20

A T O M I Z I N G M E D I A D A T A

MEDIA IS STEAM
RATIO IS 0.30 LB/LB OIL
TEMP. IS 360.0 F

UOP PROCESS DIVISION - UOP INC
FUEL OIL EVALUATION P714.1D

FUEL OIL PROGRAM OUTPUT REPORT

25.0 PER CENT EXCESS AIR

TEMP	HEAT CONTENT BTU/LB	HEAT CONTENT BTU/MOLE	PERCENT EXTRACTED
60.0	0.000	0.000	100.000
100.0	10.173	292.107	98.929
200.0	35.824	1028.642	96.227
300.0	61.613	1774.865	93.490
400.0	88.163	2531.495	90.715
500.0	114.896	3299.087	87.899
600.0	142.023	4078.016	85.042
700.0	169.553	4868.496	82.145
800.0	197.409	5670.621	79.201
900.0	225.828	6484.340	76.216
1000.0	254.565	7309.508	73.190
1100.0	283.691	8145.828	70.122
1200.0	313.195	8992.992	67.015
1300.0	343.062	9850.566	63.869
1400.0	373.273	10718.055	60.680
1500.0	403.812	11594.934	57.471
1600.0	434.657	12480.609	54.223
1700.0	465.789	13374.520	50.944
1800.0	497.185	14276.004	47.638
1900.0	528.026	15164.543	44.305
2000.0	560.690	16099.480	40.949
2100.0	592.759	17020.293	37.572
2200.0	625.015	17946.477	34.175
2300.0	657.441	18877.570	30.760
2400.0	690.025	19813.176	27.328
2500.0	722.757	20753.020	23.881
2600.0	755.628	21696.863	20.419
2700.0	788.635	22644.609	16.943
2800.0	821.778	23596.281	13.452
2900.0	855.064	24552.043	9.946
3000.0	888.503	25512.191	6.425
3100.0	922.111	26477.219	2.885
3181.1	949.506	27263.824	0.000

UOP PROCESS DIVISION - UOP INC
FUEL OIL EVALUATION P714.1B

% STACK LOSS VS FLUE GAS TEMPERATURE
AT 400. F AIR TEMPERATURE

(THIS TABLE DOES NOT ALLOW FOR RADIATION LOSSES)

FUEL OIL PROGRAM OUTPUT REPORT

% EXC. AIR	FLUE GAS TEMPERATURE, F																			
	60.	100.	200.	300.	400.	500.	600.	700.	800.	900.	1000.	1100.	1200.	1300.	1400.	1500.	1600.	1700.	1800.	1900.	2000.
0	0.0	0.8	2.9	5.0	7.2	9.3	11.6	13.8	16.1	18.4	20.8	23.1	25.5	28.0	30.5	33.0	35.5	38.0	40.6	43.2	45.8
5	0.0	0.9	3.0	5.2	7.5	9.7	12.0	14.4	16.7	19.1	21.6	24.1	26.6	29.1	31.7	34.3	36.9	39.5	42.2	44.9	47.6
10	0.0	0.9	3.1	5.4	7.7	10.1	12.5	14.9	17.4	19.9	22.4	25.0	27.6	30.2	32.8	35.5	38.3	41.0	43.8	46.6	49.4
15	0.0	0.9	3.3	5.6	8.0	10.5	12.9	15.4	18.0	20.6	23.2	25.8	28.5	31.3	34.0	36.8	39.6	42.5	45.3	48.2	51.1
20	0.0	1.0	3.4	5.8	8.3	10.8	13.4	16.0	18.6	21.3	24.0	26.7	29.5	32.3	35.2	38.1	41.0	43.9	46.9	49.8	52.8
25	0.0	1.0	3.5	6.0	8.6	11.2	13.8	16.5	19.2	22.0	24.8	27.6	30.5	33.4	36.4	39.3	42.3	45.4	48.4	51.5	54.6
30	0.0	1.0	3.6	6.2	8.9	11.6	14.3	17.0	19.8	22.7	25.6	28.5	31.4	34.4	37.5	40.5	43.6	46.8	49.9	53.1	56.3
35	0.0	1.1	3.7	6.4	9.1	11.9	14.7	17.6	20.4	23.4	26.3	29.4	32.4	35.5	38.6	41.8	45.0	48.2	51.4	54.7	58.0
40	0.0	1.1	3.8	6.6	9.4	12.3	15.1	18.1	21.0	24.1	27.1	30.2	33.4	36.6	39.7	43.0	46.3	49.6	52.9	56.3	59.7
45	0.0	1.1	3.9	6.8	9.7	12.6	15.6	18.6	21.6	24.7	27.9	31.1	34.3	37.6	40.9	44.2	47.6	51.0	54.4	57.9	61.4
50	0.0	1.1	4.0	7.0	9.9	13.0	16.0	19.1	22.2	25.4	28.7	31.9	35.2	38.6	42.0	45.4	48.9	52.4	55.9	59.4	63.0
55	0.0	1.2	4.2	7.2	10.2	13.3	16.4	19.6	22.8	26.1	29.4	32.8	36.2	39.6	43.1	46.6	50.2	53.8	57.4	61.0	64.7
60	0.0	1.2	4.3	7.4	10.5	13.6	16.9	20.1	23.4	26.8	30.2	33.6	37.1	40.6	44.2	47.8	51.5	55.1	58.8	62.6	66.3
65	0.0	1.2	4.4	7.5	10.7	14.0	17.3	20.6	24.0	27.5	30.9	34.5	38.0	41.7	45.3	49.0	52.7	56.5	60.3	64.1	68.0
70	0.0	1.3	4.5	7.7	11.0	14.3	17.7	21.1	24.6	28.1	31.7	35.3	39.0	42.7	46.4	50.2	54.0	57.9	61.7	65.7	69.6
75	0.0	1.3	4.6	7.9	11.3	14.7	18.1	21.6	25.2	28.8	32.4	36.1	39.9	43.7	47.5	51.4	55.3	59.2	63.2	67.2	71.2
80	0.0	1.3	4.7	8.1	11.5	15.0	18.5	22.1	25.8	29.4	33.2	37.0	40.8	44.7	48.6	52.5	56.5	60.6	64.6	68.7	72.8
85	0.0	1.4	4.8	8.3	11.8	15.3	19.0	22.6	26.3	30.1	33.9	37.8	41.7	45.6	49.6	53.7	57.8	61.9	66.0	70.2	74.4
90	0.0	1.4	4.9	8.5	12.0	15.7	19.4	23.1	26.9	30.7	34.6	38.6	42.6	46.6	50.7	54.8	59.0	63.2	67.5	71.7	76.0
95	0.0	1.4	5.0	8.6	12.3	16.0	19.8	23.6	27.5	31.4	35.4	39.4	43.5	47.6	51.8	56.0	60.2	64.5	68.9	73.2	77.6
100	0.0	1.5	5.1	8.8	12.6	16.3	20.2	24.1	28.0	32.0	36.1	40.2	44.4	48.6	52.8	57.1	61.5	65.8	70.3	74.7	79.2

UOP PROCESS DIVISION - UOP INC
FUEL OIL EVALUATION P714.1B

% STACK LOSS VS FLUE GAS TEMPERATURE
AT 60. F AIR TEMPERATURE

(THIS TABLE DOES NOT ALLOW FOR RADIATION LOSSES)

FUEL OIL PROGRAM OUTPUT REPORT

EXC. AIR	FLUE GAS TEMPERATURE, F																			
	60.	100.	200.	300.	400.	500.	600.	700.	800.	900.	1000.	1100.	1200.	1300.	1400.	1500.	1600.	1700.	1800.	1900.	2000.
0	0.0	0.9	3.1	5.4	7.6	10.0	12.3	14.7	17.2	19.6	22.1	24.7	27.3	29.9	32.5	35.2	37.9	40.6	43.3	46.1	48.9
5	0.0	0.9	3.2	5.6	8.0	10.4	12.9	15.4	17.9	20.5	23.1	25.7	28.4	31.1	33.9	36.6	39.4	42.3	45.1	48.0	50.9
10	0.0	1.0	3.4	5.8	8.3	10.8	13.4	16.0	18.6	21.3	24.0	26.8	29.5	32.4	35.2	38.1	41.0	44.0	46.9	49.9	52.9
15	0.0	1.0	3.5	6.0	8.6	11.2	13.9	16.6	19.3	22.1	24.9	27.8	30.7	33.6	36.6	39.6	42.6	45.7	48.7	51.8	55.0
20	0.0	1.0	3.6	6.3	9.0	11.7	14.4	17.2	20.1	22.9	25.9	28.8	31.8	34.9	37.9	41.0	44.2	47.4	50.5	53.8	57.0
25	0.0	1.1	3.8	6.5	9.3	12.1	15.0	17.9	20.8	23.8	26.8	29.9	33.0	36.1	39.3	42.5	45.8	49.0	52.4	55.7	59.0
30	0.0	1.1	3.9	6.7	9.6	12.5	15.5	18.5	21.5	24.6	27.7	30.9	34.1	37.4	40.7	44.0	47.4	50.7	54.2	57.6	61.1
35	0.0	1.1	4.0	7.0	9.9	13.0	16.0	19.1	22.3	25.4	28.7	32.0	35.3	38.6	42.0	45.5	48.9	52.4	56.0	59.5	63.1
40	0.0	1.2	4.2	7.2	10.3	13.4	16.5	19.7	23.0	26.3	29.6	33.0	36.4	39.9	43.4	46.9	50.5	54.1	57.8	61.5	65.2
45	0.0	1.2	4.3	7.4	10.6	13.8	17.1	20.4	23.7	27.1	30.5	34.0	37.6	41.1	44.8	48.4	52.1	55.8	59.6	63.4	67.2
50	0.0	1.3	4.4	7.7	10.9	14.2	17.6	21.0	24.4	27.9	31.5	35.1	38.7	42.4	46.1	49.9	53.7	57.5	61.4	65.3	69.2
55	0.0	1.3	4.6	7.9	11.2	14.7	18.1	21.6	25.2	28.8	32.4	36.1	39.9	43.7	47.5	51.4	55.3	59.2	63.2	67.2	71.3
60	0.0	1.3	4.7	8.1	11.6	15.1	18.6	22.2	25.9	29.6	33.4	37.2	41.0	44.9	48.9	52.8	56.9	60.9	65.0	69.1	73.3
65	0.0	1.4	4.8	8.4	11.9	15.5	19.2	22.9	26.6	30.4	34.3	38.2	42.2	46.2	50.2	54.3	58.4	62.6	66.8	71.1	75.3
70	0.0	1.4	5.0	8.6	12.2	15.9	19.7	23.5	27.3	31.3	35.2	39.2	43.3	47.4	51.6	55.8	60.0	64.3	68.6	73.0	77.4
75	0.0	1.5	5.1	8.8	12.6	16.4	20.2	24.1	28.1	32.1	36.2	40.3	44.4	48.7	52.9	57.3	61.6	66.0	70.4	74.9	79.4
80	0.0	1.5	5.2	9.0	12.9	16.8	20.7	24.7	28.8	32.9	37.1	41.3	45.6	49.9	54.3	58.7	63.2	67.7	72.2	76.8	81.4
85	0.0	1.5	5.4	9.3	13.2	17.2	21.3	25.4	29.5	33.7	38.0	42.4	46.7	51.2	55.7	60.2	64.8	69.4	74.1	78.7	83.5
90	0.0	1.6	5.5	9.5	13.5	17.6	21.8	26.0	30.3	34.6	39.0	43.4	47.9	52.4	57.0	61.7	66.4	71.1	75.9	80.7	85.5
95	0.0	1.6	5.6	9.7	13.9	18.1	22.3	26.6	31.0	35.4	39.9	44.4	49.0	53.7	58.4	63.1	67.9	72.8	77.7	82.6	87.5
100	0.0	1.6	5.8	10.0	14.2	18.5	22.8	27.2	31.7	36.2	40.8	45.5	50.2	54.9	59.8	64.6	69.5	74.5	79.5	84.5	89.6

UOP PROCESS DIVISION - UOP INC
FUEL GAS EVALUATION P714.1A

% STACK LOSS VS FLUE GAS TEMPERATURE
AT 60. F AIR TEMPERATURE

(THIS TABLE DOES NOT ALLOW FOR RADIATION LOSSES)

NATURAL GAS

% EXC. AIR	FLUE GAS TEMPERATURE, F																				
	60.	100.	200.	300.	400.	500.	600.	700.	800.	900.	1000.	1100.	1200.	1300.	1400.	1500.	1600.	1700.	1800.	1900.	2000.
0	0.0	0.9	3.2	5.5	7.9	10.3	12.7	15.2	17.7	20.2	22.8	25.4	28.0	30.7	33.4	36.1	38.9	41.7	44.5	47.4	50.2
5	0.0	1.0	3.3	5.6	8.2	10.7	13.3	15.8	18.4	21.1	23.7	26.5	29.2	32.0	34.8	37.7	40.5	43.4	46.4	49.3	52.3
10	0.0	1.0	3.5	6.0	8.6	11.2	13.8	16.5	19.2	21.9	24.7	28.5	30.4	33.3	36.2	39.2	42.2	45.2	48.2	51.3	54.4
15	0.0	1.0	3.6	6.2	8.9	11.6	14.3	17.1	19.9	22.8	25.7	28.6	31.6	34.6	37.6	40.7	43.8	46.9	50.1	53.3	56.5
20	0.0	1.1	3.8	6.5	9.2	12.0	14.9	17.7	20.7	23.6	26.6	29.7	32.7	35.8	39.0	42.2	45.4	48.7	51.9	55.2	58.6
25	0.0	1.1	3.9	6.7	9.6	12.5	15.4	18.4	21.4	24.5	27.6	30.7	33.9	37.1	40.4	43.7	47.0	50.4	53.8	57.2	60.7
30	0.0	1.1	4.0	7.0	9.9	12.9	15.9	19.0	22.1	25.3	28.5	31.8	35.1	38.4	41.8	45.2	48.7	52.1	55.6	59.2	62.7
35	0.0	1.2	4.2	7.2	10.2	13.3	16.5	19.7	22.9	26.2	29.5	32.8	36.3	39.7	43.2	46.7	50.3	53.9	57.5	61.1	64.8
40	0.0	1.2	4.3	7.4	10.6	13.8	17.0	20.3	23.6	27.0	30.4	33.9	37.4	41.0	44.6	48.2	51.9	55.6	59.3	63.1	66.9
45	0.0	1.3	4.4	7.7	10.9	14.2	17.6	20.9	24.4	27.9	31.4	35.0	38.6	42.3	46.0	49.7	53.5	57.3	61.2	65.1	69.0
50	0.0	1.3	4.6	7.9	11.3	14.7	18.1	21.6	25.1	28.7	32.4	36.0	39.8	43.6	47.4	51.2	55.1	59.1	63.1	67.1	71.1
55	0.0	1.3	4.7	8.1	11.6	15.1	18.6	22.2	25.9	29.6	33.3	37.1	41.0	44.9	48.8	52.8	56.8	60.8	64.9	69.0	73.2
60	0.0	1.4	4.9	8.4	11.9	15.5	19.2	22.9	26.6	30.4	34.3	38.2	42.1	46.1	50.2	54.3	58.4	62.6	66.8	71.0	75.3
65	0.0	1.4	5.0	8.6	12.3	16.0	19.7	23.5	27.4	31.3	35.2	39.3	43.3	47.4	51.6	55.8	60.0	64.3	68.6	73.0	77.4
70	0.0	1.5	5.1	8.8	12.6	16.4	20.3	24.2	28.1	32.1	36.2	40.3	44.5	48.7	53.0	57.3	61.7	66.0	70.5	74.9	79.3
75	0.0	1.5	5.3	9.1	12.9	16.8	20.8	24.8	28.9	33.0	37.2	41.4	45.7	50.0	54.4	58.8	63.3	67.8	72.3	76.9	81.5
80	0.0	1.5	5.4	9.3	13.3	17.3	21.3	25.4	29.6	33.8	38.1	42.5	46.8	51.3	55.8	60.3	64.9	69.5	74.2	78.9	83.6
85	0.0	1.6	5.5	9.6	13.6	17.7	21.9	26.1	30.4	34.7	39.1	43.5	48.0	52.6	57.2	61.8	66.5	71.3	76.0	80.9	85.7
90	0.0	1.6	5.7	9.8	13.9	18.1	22.4	26.7	31.1	35.5	40.0	44.6	49.2	53.9	58.6	63.3	68.2	73.0	77.9	82.8	87.8
95	0.0	1.7	5.8	10.0	14.3	18.6	22.9	27.4	31.8	36.4	41.0	45.7	50.4	55.1	60.0	64.9	69.8	74.7	79.8	84.8	89.9
100	0.0	1.7	6.0	10.3	14.6	19.0	23.5	28.0	32.6	37.2	41.9	46.7	51.5	56.4	61.4	66.4	71.4	76.5	81.6	86.8	92.0

UOP PROCESS DIVISION - UOP INC
FUEL GAS EVALUATION P714.1A

% STACK LOSS VS FLUE GAS TEMPERATURE
AT 400. F AIR TEMPERATURE

(THIS TABLE DOES NOT ALLOW FOR RADIATION LOSSES)

IATURAL GAS

(C.
1

	FLUE GAS TEMPERATURE, F																			
	60.	100.	200.	300.	400.	500.	600.	700.	800.	900.	1000.	1100.	1200.	1300.	1400.	1500.	1600.	1700.	1800.	1900.	2000.
)	0.0	0.9	3.0	5.2	7.4	9.6	11.9	14.2	16.6	18.9	21.3	23.8	26.3	28.8	31.3	33.9	36.5	39.1	41.7	44.4	47.1
)	0.0	0.9	3.1	5.4	7.7	10.0	12.4	14.8	17.2	19.7	22.2	24.7	27.3	29.9	32.5	35.2	37.9	40.6	43.3	46.1	48.9
)	0.0	0.9	3.2	5.6	8.0	10.4	12.8	15.3	17.8	20.4	23.0	25.6	28.3	31.0	33.7	36.5	39.3	42.1	44.9	47.8	50.7
)	0.0	1.0	3.4	5.8	8.3	10.8	13.3	15.9	18.5	21.1	23.8	26.5	29.3	32.1	34.9	37.8	40.6	43.6	46.5	49.5	52.4
)	0.0	1.0	3.5	6.0	8.6	11.1	13.8	16.4	19.1	21.9	24.6	27.4	30.3	33.2	36.1	39.0	42.0	45.0	48.1	51.1	54.2
)	0.0	1.0	3.6	6.2	8.8	11.5	14.2	17.0	19.7	22.6	25.4	28.3	31.3	34.3	37.3	40.3	43.4	46.5	49.6	52.8	56.0
)	0.0	1.1	3.7	6.4	9.1	11.9	14.7	17.5	20.4	23.3	26.2	29.2	32.3	35.3	38.4	41.6	44.7	47.9	51.2	54.4	57.7
)	0.0	1.1	3.8	6.6	9.4	12.2	15.1	18.0	21.0	24.0	27.0	30.1	33.2	36.4	39.6	42.8	46.1	49.4	52.7	56.1	59.4
)	0.0	1.1	3.9	6.8	9.7	12.6	15.6	18.6	21.6	24.7	27.8	31.0	34.2	37.5	40.8	44.1	47.4	50.8	54.3	57.7	61.2
)	0.0	1.2	4.1	7.0	9.9	13.0	16.0	19.1	22.2	25.4	28.6	31.9	35.2	38.5	41.9	45.3	48.8	52.3	55.8	59.3	62.9
)	0.0	1.2	4.2	7.2	10.2	13.3	16.4	19.6	22.8	26.1	29.4	32.8	36.1	39.6	43.0	46.6	50.1	53.7	57.3	60.9	64.6
)	0.0	1.2	4.3	7.4	10.5	13.7	16.9	20.1	23.4	26.8	30.2	33.6	37.1	40.6	44.2	47.8	51.4	55.1	58.8	62.5	66.3
)	0.0	1.2	4.4	7.6	10.8	14.0	17.3	20.7	24.0	27.5	31.0	34.5	38.0	41.7	45.3	49.0	52.7	56.5	60.3	64.1	68.0
)	0.0	1.3	4.5	7.7	11.0	14.4	17.7	21.2	24.6	28.2	31.7	35.3	39.0	42.7	46.4	50.2	54.0	57.9	61.8	65.7	69.6
)	0.0	1.3	4.6	7.9	11.3	14.7	18.2	21.7	25.2	28.8	32.5	36.2	39.9	43.7	47.6	51.4	55.3	59.3	63.3	67.3	71.3
	0.0	1.3	4.7	8.1	11.6	15.1	18.6	22.2	25.8	29.5	33.2	37.0	40.9	44.7	48.7	52.6	56.6	60.7	64.7	68.8	73.0
	0.0	1.4	4.8	8.3	11.8	15.4	19.0	22.7	26.4	30.2	34.0	37.9	41.8	45.8	49.8	53.8	57.9	62.0	66.2	70.4	74.6
	0.0	1.4	4.9	8.5	12.1	15.8	19.5	23.2	27.0	30.9	34.8	38.7	42.7	46.8	50.9	55.0	59.2	63.4	67.6	71.9	76.2
	0.0	1.4	5.0	8.7	12.4	16.1	19.9	23.7	27.6	31.5	35.5	39.5	43.6	47.8	51.9	56.2	60.4	64.7	69.1	73.5	77.9
	0.0	1.5	5.1	8.9	12.6	16.4	20.3	24.2	28.2	32.2	36.2	40.4	44.5	48.8	53.0	57.3	61.7	66.1	70.5	75.0	79.5
	0.0	1.5	5.3	9.0	12.9	16.8	20.7	24.7	28.7	32.8	37.0	41.2	45.4	49.7	54.1	58.5	62.9	67.4	71.9	76.5	81.1

Problem 1

UOP

TECHNICAL SERVICES DEPARTMENT				Date:	2/26/98
Refinery Projects & Services				Project:	
				By:	SAV
Customer:	ABC Refining				
Location:	Des Plaines, IL				
Comments:	Problem 3 - High Hydrogen Fuel Gas				
	Fuel GAS Program Version 1.11, 8/4/94				
FUEL GAS DATA	Refinery fuel gas		FLUE GAS INFO		Btu/lb
Adiabatic flame temp, °F	3552		15%xs air	bwt	enthalpy
LHV of fuel, BTU/lb	34190			1400	406.32
LHV, Btu/SCF	379.27			1500	439.52
lbs of wet air/lb fuel	27.97				
(Flowing @ Amb Air) Ft3 of wet air/lb fuel	369.87		BW temp, °F =	1350	390
Air preheat temp, °F	60		% Radiant Efficiency =	65.11%	
Air preheat content, btu/lb	0.00		FG out stack, °F =	300	67.85
Corrected heat content fuel, btu/lb	34190		Heat Loss =	2.00%	
Corrected heat content fuel, Btu/SCF	379.3				
Note: "P air side burner " (Mass Flow) "2 " Specific Volume (ft3/lb)			% Total Efficiency =	92.28%	
Site Elevation, feet	0				
Calc. Site Pressure, psig	14.7		Fuel MW =	4.20	
Component	mole%		Excess Air		
Hydrogen H2	92.300		%		
Nitrogen N2	0.000		15		
Oxygen O2	0.000				
Water H2O	0.000		Relative		
Carbon monoxide CO	0.000				
Carbon dioxide CO2	0.000		Humidity %		
Hydrogen sulfide H2S	0.000		100		
Sulfur dioxide SO2	0.000				
Ammonia NH3	0.000		Ambient		
Methane C1	2.900		Temp °F		
Ethane C2	2.700		80		
Propane C3	1.400				
n-Butane nC4	0.400		Combustion		
2-Methylpropane iC4	0.000		Air Temp °F		
n-Pentane nC5	0.300		60		
2-Methylbutane iC5	0.000				
22-Dimethylpropane iC5	0.000		Fuel Gas		
n-Hexane nC6	0.000		Temp °F		
2-Methylpentane iC6	0.000		60		
3-Methylpentane iC6	0.000				
22-Dimethylbutane iC6	0.000		lue Gas MW =	26.02	
23-Dimethylbutane iC6	0.000			ON A WET BASIS	
n-Heptane nC7	0.000		IN FLUE GAS	WT %	VOLUME %
Ethene C2 =	0.000		CO2	5.64	3.33
Propene C3 =	0.000		SO2	0.00	0.00
Isobutene iC4	0.000		H2O	18.15	26.22
1-Butene nC4	0.000		O2	2.96	2.40
	100.000		N2	73.25	68.05
			Sum =	100.00	100.00
				ON A DRY BASIS	
			IN FLUE GAS		
			CO2	6.89	4.52
			SO2	0.00	0.00
			H2O	0.00	0.00
			O2	3.61	3.25
			N2	89.50	92.23

Problem 2

UOP

TECHNICAL SERVICES DEPARTMENT				Date:	2/26/98
Refinery Projects & Services				Project:	
				By:	SAV
Customer:	ABC Refining				
Location:	Des Plaines, IL				
Comments:	Problem 2 - Fuel Oil				
	Fuel OIL Program Version 1.11, 5/26/95				
FUEL OIL DATA				FLUE GAS INFO	Btu/lb
Adiabatic flame temp, °F	2956	35%xs air		bwt	enthalpy
LHV of fuel, BTU/lb	17586			1400	371.6
				1500	401.9
lbs of wet air/lb fuel	18.99		BW temp, °F =	1450	386.7
(Flowing @ Amb Air) Ft3 of wet air/lb fuel	3.83		% Radiant Efficiency =	53.51%	
Air preheat temp, °F	60		FG out stack, °F =	500	114.5
Air preheat content, btu/lb	0.0		Heat Loss =	2.00%	
Corrected heat content fuel, btu/lb	17616		% Total Efficiency =	84.83%	
Site Elevation, feet	0				
Calc. Site Pressure, psig	14.700				
API of Fuel Oil	11		Excess Air%	34.8	
Carbon to Hydrogen Ratio	8.00		Ambient Temp °F	60	
Weight % S	0.20		Relative Humidity %	100	
Weight % N	0.00		Combustion Air, °F	60	
Weight % Ash	0.00				
Weight % C	88.72		Atomizing Steam, Psig	139	
Weight % H	11.08		Atomizing Steam, °F	360	
Lbs steam/Lb of Fuel	0.30				
Fuel Oil Temp, °F	180				
Flue Gas MW =	28.71				
	ON A WET BASIS			DRY BASIS	
IN FLUE GAS	WT %	VOLUME %		WT %	VOLUME %
CO2	16.02	10.45		17.29	11.84
SO2	0.02	0.009		0.02	0.01
H2O	7.36	11.74		0.00	0.00
O2	5.57	5.00		6.01	5.66
N2	71.02	72.80		76.67	82.48
Sum =	100.00	100.00		100.00	100.00

Problem 3

UOP

TECHNICAL SERVICES DEPARTMENT				Date:	2/26/98
Refinery Projects & Services				Project:	
				By:	SAV
Customer:	ABC Refining				
Location:	Des Plaines, IL				
Comments:	Problem 3 - High Hydrogen Fuel Gas				
	Fuel GAS Program Version 1.11, 8/4/94				
FUEL GAS DATA	Refinery fuel gas				
Adiabatic flame temp. °F	3552		FLUE GAS INFO		Btu/lb
LHV of fuel, BTU/lb	34190		15%xs air	bwt	enthalpy
LHV, Btu/SCF	379.27			1400	406.32
lbs of wet air/lb fuel	27.97			1500	439.52
(Flowing @ Amb Air) Ft3 of wet air/lb fuel	369.87		BW temp, °F =	1350	390
Air preheat temp, °F	60		% Radiant Efficiency =	65.11%	
Air preheat content, btu/lb	0.00		FG out stack, °F =	300	67.85
Corrected heat content fuel, btu/lb	34190		Heat Loss =	2.00%	
Corrected heat content fuel, Btu/SCF	379.3				
Note: "P air side burner " (Mass Flow) "2 * Specific Volume (ft3/lb)			% Total Efficiency =	92.28%	
Site Elevation, feet	0				
Calc. Site Pressure, psig	14.7		Fuel MW =	4.20	
Component	mole%		Excess Air		
Hydrogen H2	92.300		%		
Nitrogen N2	0.000		15		
Oxygen O2	0.000				
Water H2O	0.000		Relative		
Carbon monoxide CO	0.000				
Carbon dioxide CO2	0.000		Humidity %		
Hydrogen sulfide H2S	0.000		100		
Sulfur dioxide SO2	0.000				
Ammonia NH3	0.000		Ambient		
Methane C1	2.900		Temp °F		
Ethane C2	2.700		60		
Propane C3	1.400				
n-Butane nC4	0.400		Combustion		
2-Methylpropane iC4	0.000		Air Temp °F		
n-Pentane nC5	0.300		60		
2-Methylbutane iC5	0.000				
22-Dimethylpropane iC5	0.000		Fuel Gas		
n-Hexane nC6	0.000		Temp °F		
2-Methylpentane iC6	0.000		60		
3-Methylpentane iC6	0.000				
22-Dimethylbutane iC6	0.000		lue Gas MW =	26.02	
23-Dimethylbutane iC6	0.000			ON A WET BASIS	
n-Heptane nC7	0.000		IN FLUE GAS	WT %	VOLUME %
Ethene C2 =	0.000		CO2	5.64	3.33
Propene C3 =	0.000		SO2	0.00	0.00
Isobutene iC4	0.000		H2O	18.15	26.22
1-Butene nC4	0.000		O2	2.96	2.40
	100.000		N2	73.25	68.05
			Sum =	100.00	100.00
				ON A DRY BASIS	
			IN FLUE GAS		
			CO2	6.89	4.52
			SO2	0.00	0.00
			H2O	0.00	0.00
			O2	3.61	3.25
			N2	89.50	92.23

uop

ENGINEERING DEPARTMENT		date:	9/15/97
Equipment Skill Center		Project:	56000
		By:	FRB
customer:	ABC Oil Refining Company		
location:	Des Plaines, IL		
comments:	Case 1 Fuel		

FUEL GAS PROGRAM
Version 1.11, 8/4/94

FUEL GAS DATA	Refinery fuel gas			
adiabatic flame temp, °F	3354	FLUE GAS INFO:		Btu/lb
LHV of fuel, BTU/lb	21495	15%xs air	bwt	enthalpy
LHV, Btu/SCF	909.88		1400	383.27
lbs of wet air/lb fuel	19.70		1500	414.61
owing @ Amb Air) Ft3 of wet air/lb fuel	262.57	BW temp, °F=	**1450**	399
air preheat temp, °F	60	% Radiant Efficiency=	59.51%	
air preheat content, btu/lb	0.00	FG out stack, °F=	**400**	90.79
Corrected heat content fuel, btu/lb	21495	Heat Loss=	**2.00%**	
Corrected heat content fuel, Btu/SCF	909.9			
Note: ~P air side burner ~ (Mass Flow)^2 * Specific Volume (ft3/lb)		% Total Efficiency=	89.24%	

Site Elevation, feet	**385**		
Calc. Site Pressure, PSIG	14	Fuel MW=	16.04

Component		mole%	Excess Air
Hydrogen	H2	**0.000**	%
Nitrogen	N2	**0.000**	**15**
Oxygen	O2	**0.000**	
Water	H2O	**0.000**	Relative
Carbon monoxide	CO	**0.000**	
Carbon dioxide	CO2	**0.000**	Humidity %
Hydrogen sulfide	H2S	**0.000**	**0**
Sulfur dioxide	SO2	**0.000**	
Ammonia	NH3	**0.000**	Ambient
Methane	C1	**100.000**	Temp °F
Ethane	C2	**0.000**	**60**
Propane	C3	**0.000**	
n-Butane	nC4	**0.000**	Combustion
2-Methylpropane	iC4	**0.000**	Air Temp °F
n-Pentane	nC5	**0.000**	**60**
2-Methylbutane	iC5	**0.000**	
22-Dimethylpropane	iC5	**0.000**	Fuel Gas
n-Hexane	nC6	**0.000**	Temp °F
2-Methylpentane	iC6	**0.000**	**60**
3-Methylpentane	iC6	**0.000**	

uop

ENGINEERING DEPARTMENT				date:	9/15/97
Equipment Skill Center				Project:	56000
				By:	FRB
customer:	ABC Oil Refining Company				
location:	Des Plaines, IL				
comments:	Case 1 Fuel				
22-Dimethylbutane	iC6	0.000	Flue Gas MW=	27.78	
23-Dimethylbutane	iC6	0.000		ON A WET-BASIS	
n-Heptane	nC7	0.000	IN FLUE GAS	WT %	VOLUME %
Ethene	C2=	0.000	CO2	13.25	8.37
Propene	C3=	0.000	SO2	0.00	0.000
Isobutene	iC4=	0.000	H2O	10.85	16.73
1-Butene	nC4=	0.000	O2	2.89	2.51
		100.00	N2	73.00	72.39
			Sum=	100.00	100.00
				ON A DRY BASIS	
			IN FLUE GAS		
			CO2	14.87	10.05
			SO2	0.00	0.00
			H2O	0.00	0.00
			O2	3.24	3.01
			N2	81.89	86.94
			Sum=	100	100

Heater Efficiency and Duty

GAS FIRING (Data August 31, 1995)

$$\text{Total Efficiency} = \frac{(\text{Enthalpy in} * (1 - \text{fraction loss}) - \text{Enthalpy out})}{\text{Enthalpy in}}$$

89.24 % Total Efficiency

Flue Gas

400 DegF
91 Btu/lb — Enthalpy Out

3.01 %O2 Dry Basis

1450 DegF
399 Btu/lb — Flue gas enthalpy after radiant coil

Enthalpy in

3354 DegF
1036 Btu/lb

59.51 % Radiant Efficiency

0.02 Fraction Losses

15 % Excess Air
60 DegF
125 Btu/lb
19.70 Lbs air per Lb of fuel
12059 SCF air per mm Btu heat Release

Fuel
LHV= 21495 Btu/lb of fuel
910 Btu/SCF

Heater Efficiency and Duty
GAS FIRING

Fuel flow rate=	14.47 MSCFH 613 lb/h	Flue gas rate=	12677 lb/h

Radiant Duty=	8.08 mm btu/h
Convection Duty=	3.91 mm btu/h
Total Heater Duty=	11.99 mm btu/h

UOP

ENGINEERING DEPARTMENT
Equipment Skill Center

date: 9/15/97
Project: 56000
By: FRB

Customer: ABC Oil Refining Company
location: Des Plaines, IL
comments:

FUEL OIL PROGRAM
Version 1.11, 5/26/95

FUEL OIL DATA	Refinery fuel gas		FLUE GAS INFO	Btu/lb
adiabatic flame temp, °F	3152	25%xs air	bwt	enthalpy
LHV of fuel, BTU/lb	17627		1400	373.3
			1500	403.9
lbs of wet air/lb fuel	17.48	BW temp, °F=	**1500**	403.9
owing @ Amb Air) Ft3 of wet air/lb fuel	3.89	% Radiant Efficiency=	55.03%	
air preheat temp, °F	60	FG out stack, °F=	**400**	88.2
air preheat content, btu/lb	0.0	Heat Loss=	**2.00%**	
Corrected heat content fuel, btu/lb	17652	% Total Efficiency=	88.62%	
Site Elevation, feet	**385**			
Calc. Site Pressure, PSIG	14.487			
API of Fuel Oil	**12**	Excess Air%	**25**	
Carbon to Hydrogen Ratio	7.92	Ambient Temp °F	**60**	
Weight % S	**1.00**	Relative Humidity %	**100**	
Weight % N	**0.30**	Combustion Air, °F	**60**	
Weight % Ash	**0.00**			
Weight % C	87.64	Atomizing Steam, Psig	**50**	
Weight % H	11.06	Atomizing Steam, °F	298	
Lbs steam/Lb of Fuel	**0.30**			
Fuel Oil Temp, °F	**60**			
Flue Gas MW=	28.72			

	ON A WET BASIS		DRY BASIS	
IN FLUE GAS	WT %	VOLUME %	WT %	VOLUME %
CO2	17.10	11.15	18.55	12.75
SO2	0.11	0.048	0.12	0.05
H2O	7.86	12.53	0.00	0.00
O2	4.29	3.85	4.66	4.40
N2	70.65	72.42	76.67	82.79
Sum=	100.00	100.00	100	100

Heater Efficiency and Duty

OIL FIRING (Data August 31, 1995)

$$\text{Total Efficiency} = \frac{(\text{Enthalpy in} * (1 - \text{fraction loss}) - \text{Enthalpy out})}{\text{Enthalpy in}}$$

88.62 % Total Efficiency

Flue Gas

400 DegF
88 Btu/lb ← Enthalpy Out

4.40 %O2 Dry Basis

1500 DegF
404 Btu/lb ← Flue gas enthalpy after radiant coil

Enthalpy in

3152 DegF
940 Btu/lb

55.03 % Radiant Efficiency

0.02 Fraction Losses

25 % Excess Air
60 DegF
0 Btu/lb
17.48 Lbs air per Lb of fuel
13033 SCF air per mm Btu heat Release

Fuel

Corrected HV= 17652 Btu/lb of fuel

Heater Efficiency and Duty
OIL FIRING

Gravity=	0.986		
Volume correction factor=	1.000		
Volume Flow=	**49.0** BPSD	Flue gas rate=	13017 lb/h
Fuel flow rate=	704 lb/h		

Radiant Duty=	6.98 mm btu/h
Convection Duty=	4.11 mm btu/h
Total Heater Duty=	11.09 mm btu/h

UOP Fuels Programs

Fuels Programs Provided:

Two programs have been provided on a 3.5 inch, 1.44 MB, high density, IBM formated floppy disk (Macintosh formated also available on request). The first program, "Fuels.XLS" uses Microsoft Excel version 5.0. The second program, "Fuel_Gas.WK4" uses Lotus 123 release 5. UOP's engineering department has standardized on Microsoft Excel and therefore the fuel oil program which was written using Excel's Visual Basic could not be exported to the Lotus format.

Purpose of Fuels Programs:

The fuel gas and fuel oil programs provide a theoretical flame temperature as a function of composition (and API gravity for fuel oil), percent excess air, ambient air temperature, combustion air temperature, fuel temperature and in the case of fuel oil, steam requirements. Radiant heat transfer is proportional to theoretical flame temperature to the fourth power minus tube temperature to the fourth power, T^4flame-T^4tube. These programs have been used to provide design information but may also be used to monitor heater operations. In addition to the theoretical flame temperature, the fuels programs provide enthalpy of the flue gas leaving the radiant section and leaving the convection section. With this information an estimate of furnace efficiency can be made (see examples).

Flue Gas Enthalpy:

The flue gas specific enthalpy is determined by summing the heats of combustion of the individual components in the fuel, adding to that the fuel and air enthalpies (and steam enthalpy for oil firing, minus fuel oil latent heat) entering the furnace and dividing by the total mass entering the furnace:

$$SpecificEnthalpy = \frac{(NetHeating + FuelEnthalpy) * FuelFlowRate + AirEnthalpy * AirflowRate}{FlueGasFlowRate}$$

The products of combustion are: CO2, SO2, N2, O2, H2O with ppm amounts of NOx and CO.

The enthalpy of the fuel and products of combustion take the form:

$$H = \int_{Ref}^{T} B + C*T + D*T^2 + E*T^3 + F*T^4$$

OR

$$H = A + B*T + C*T^2 + D*T^3 + E*T^4 + F*T^5$$
$$-(A + B*T_{Ref} + C*T_{Ref}^2 + D*T_{Ref}^3 + E*T_{Ref}^4 + F*T_{Ref}^5)$$

where: H is enthalpy in btu/lb.

T is temperature in degrees rankin

A,B,C .. are constants

Net Heating Value:

The heating value of a fuel is equal to its heat of combustion but of opposite sign (positive). The Total or Gross heating value of the fuel is the heat released on complete combustion at a reference temperature of 60°F with all water condensed to the liquid state. The net heating value is similarly defined, except the final state of the water after combustion is taken as vapor.

Flame Temperature:

Calculation of theoretical flame temperature provided in this program is classical textbook. Theoretical flame temperature is the temperature attained when a fuel is burned in air (or oxygen) without gain or loss of heat. The maximum flame temperature is achieved when the fuel is burned with the stoiciometric amount of pure oxygen. Since pure oxygen is not used in heater design all further discussion will focus on flame temperture calculated at different percentages of excess air.

Specific enthalpy can be determined at several temperatures for a fixed percent excess air. Once these enthalpies are determined, a plot of temperature versus enthalpy can be drawn and theoretical flame temperature can be determined. Theoretical flame temperature can also be

determined by iterative computational methods. Iterative computational methods require specific enthalpy and products of combustion as input, see "Fuels Equations" Fuction(FlueGasTAtGivenEnt).

Thermal Efficiency:

Thermal efficiency is:

$$\%\,Efficiency = \frac{Total\_heat\_absorbed}{Total\_heat\_input} * 100$$

OR

$$\%\,Efficiency = \frac{(Enthalpy\_in * (1 - Fraction\_loss) - Enthalpy\_out)}{Enthalpy\_in} * 100$$

If the enthalpy in is determined on a per pound (per Kg) basis, this is the same as the specific enthalpy as described in the above section "Flue Gas Enthalpy". Therefore efficiency can be determined using specific enthalpies:

$$\%\,Efficiency = \frac{(Specific\_Enthalpy\_in * (1 - Fraction\_loss) - Specific\_Enthalpy\_out)}{Specific\_Enthalpy\_in} * 100$$

Problems:

Problem 1: ABC Oil Refining Co. has a vertical cylindrical reboiler which is fuel gas firing, 30 vol% H2 and 70 vol% propane. The percent excess air is 15%. They report a measured bridgewall temperature (flue gas exiting the radiant section) of 1350 °F. What is the adiabatic flame temperature? What is the radiant thermal efficiency of the furnace based on the flue gas side of this heater? Assume 2% heat losses.

Note:

1) Use heat of combustion for H2, 51623 Btu/lb.
2) Use heat of combustion for C3, 19937 Btu/lb.
3) Assume fuel temperature of 60°F. No enthalpy contribution from the fuel (reference temperature is 60°F).
4) Assume air temperature of 60°F. No enthalpy contribution from the air.
5) Assume that there is no moisture in the air, 0% humidity.
6) See section: "**Fuels Equations**" for enthalpy equations required.
7) Molecular weight of following components:

H2	2.016	C3H8	44.097
H2O	18.015	CO2	44.010
O2	31.999	N2	28.013

Hints:

Use 100 moles/hour of fuel gas fired as a basis:

One-half mole of O2 is used for each mole of H2 burned. $H2 + \frac{1}{2} * O2 \rightarrow H2O$

Five mole of O2 is used for each mole of propane burned.

$C3H8 + 5 * O2 \rightarrow 3 * CO2 + 4 * H2O$

N2 from air = $\frac{0.79}{0.21} * O2$ is required on a mole basis.

Remember total mass in (air and fuel) = total mass out (flue gas).

From section **Fuels Equations** the enthalpy of H2O (Btu/lb) is found from:

HVap = (-2.463 + 0.45739 * T1 - 0.52512 * T2 * 0.0001 + 0.64594 * T3 * 0.0000001 - 2.0276 * T4 * 0.00000000001 + 2.3631 * T5 * 0.000000000000001) * WTFracH2O

where T is in degrees R.

The enthalpy of H2O (Btu/lb) at it's reference temperature is:

HVap60 = (228.87) * WTFracH2O

Determine enthalpy at 3000 °F and 4000 °F then draw a straight line between them.

Problem 2: In August of 1995 a UOP heater specialist visited Client 1's refinery to trouble shoot a reboiler heater. Hysim was used to do a process side enthalpy balance around the column, receiver, condenser, exchangers and heater. From this process side balance it was found that the heater duty was 9.8 mm btu/h. From the fuels program and drawings included in Appendix B, what observations and conclusions can you make?

Fuels Equations:

```
'*************************************************************************************
PURPOSE OF THIS FUCTION:

'Given the temperature of flue gas in a furnace (Degrees F) and the composition of that flue
'gas as a weight fraction of CO2, O2, N2, SO2 and H2O this function returns the flue gas heat
'enthalpy in btu/lb.
'                              SOURCE OF INFORMATION:

'Constants for these equations were obtained from "Applied Hydrocarbon Thermodynamics" by
Wayne C.
'Edmister and Byung IK Lee, second edition 1984 pages 126 through 131. These are the
'Passut and Danner Equations presented by Edmister and Lee.
'
'                              LIMITATIONS / COMMENTS:
'
'These are ideal gas equations which apply to flue gases in furnaces without need to correct
'for deviation from ideal behaviour

'
```

VERIFICATION OF RELIABILITY

Results from this function give close agreement with UOP mainframe versions
Taken from previous spreadsheet version which is used heavily by UOP heater specialists

```
Function FlueGasEnthalpy(DegF As Single, WTFracCO2, WTFracO2, WTFracN2, WTFracSO2,
WTFracH2O)


  Dim T2 As Single
  Dim T3 As Single
  Dim T4 As Single
  Dim T5 As Single
  Dim HVap As Single
  Dim HVap60 As Single

  T1 = DegF + 460

  T2 = T1 * T1
  T3 = T2 * T1
  T4 = T3 * T1
  T5 = T4 * T1
  HVap = 0
  HVap60 = 0
                                        '{For CO2}
  HVap = HVap + (4.778 + 0.11443 * T1 + 1.0113 * T2 * 0.0001 - 0.2649 * T3 * 0.0000001 +
0.34706 * T4 * 0.00000000001 - 0.1314 * T5 * 0.000000000000001) * WTFracCO2 '{Btu/Lb}
  HVap60 = HVap60 + (88.151) * WTFracCO2 '{Btu/Lb}
                                        '{For O2}
  HVap = HVap + (-0.982 + 0.22749 * T1 - 0.3731 * T2 * 0.0001 + 0.4831 * T3 * 0.0000001 -
1.85243 * T4 * 0.00000000001 + 2.47488 * T5 * 0.000000000000001) * WTFracO2 '{Btu/Lb}
  HVap60 = HVap60 + (112.757) * WTFracO2 '{Btu/Lb}
                                        '{For N2}
  HVap = HVap + (-0.934 + 0.2552 * T1 - 0.1779 * T2 * 0.0001 + 0.1589 * T3 * 0.0000001 -
0.32203 * T4 * 0.00000000001 + 0.15893 * T5 * 0.000000000000001) * WTFracN2 '{Btu/Lb}
  HVap60 = HVap60 + (128.964) * WTFracN2 '{Btu/Lb}
                                        '{For SO2}
  HVap = HVap + (1.394 + 0.11026 * T1 + 0.33026 * T2 * 0.0001 + 0.0891 * T3 * 0.0000001 -
0.77313 * T4 * 0.00000000001 + 1.29287 * T5 * 0.000000000000001) * WTFracSO2 '{Btu/Lb}
  HVap60 = HVap60 + (68.396) * WTFracSO2 '{Btu/Lb}
                                        '{For H2O}
  HVap = HVap + (-2.463 + 0.45739 * T1 - 0.52512 * T2 * 0.0001 + 0.64594 * T3 * 0.0000001
- 2.0276 * T4 * 0.00000000001 + 2.3631 * T5 * 0.000000000000001) * WTFracH2O '{Btu/Lb}
  HVap60 = HVap60 + (228.87) * WTFracH2O '{Btu/Lb}
  FlueGasEnthalpy = HVap - HVap60
```

```
End Function

*****************************************************************************************
**********************************

*****************************************************************************************
**********************************
Function FlueGasTAtGivenEnth(Btu_LB As Single, WTFracCO2, WTFracO2, WTFracN2,
WTFracSO2, WTFracH2O)

  Dim CpVap As Single
  Dim T1 As Single
  Dim T2 As Single
  Dim T3 As Single
  Dim T4 As Single
  Dim T5 As Single
  Dim TotalEnth As Single
  Dim Enth60 As Single
  Dim DegF As Single

  T1 = Btu_LB / 0.25 + 460
  DegF = T1 - 460

  Do

  T2 = T1 * T1
  T3 = T2 * T1
  T4 = T3 * T1
  T5 = T4 * T1

      '{THE DERIVATIVE FOR NEWTON- RAPHSON CONVERGENCE}
  CpVap = 0

                                        '{For CO2}
  CpVap = CpVap + (0.11443 + 2 * 1.0113 * T1 * 0.0001 - 3 * 0.2649 * T2 * 0.0000001 + 4 *
0.34706 * T3 * 0.00000000001 - 5 * 0.1314 * T4 * 0.000000000000001) * WTFracCO2
'{Btu/Lb}

                                        '{For O2}
  CpVap = CpVap + (0.22749 - 2 * 0.3731 * T1 * 0.0001 + 3 * 0.4831 * T2 * 0.0000001 - 4 *
1.85243 * T3 * 0.00000000001 + 5 * 2.47488 * T4 * 0.000000000000001) * WTFracO2
'{Btu/Lb}

                                        '{For N2}
```

```
    CpVap = CpVap + (0.2552 - 2 * 0.1779 * T1 * 0.0001 + 3 * 0.1589 * T2 * 0.0000001 - 4 *
0.32203 * T3 * 0.00000000001 + 5 * 0.15893 * T4 * 0.000000000000001) * WTFracN2
'{Btu/Lb}

                                                    '{For SO2}
    CpVap = CpVap + (0.11026 + 2 * 0.33026 * T1 * 0.0001 + 3 * 0.0891 * T2 * 0.0000001 - 4 *
0.77313 * T3 * 0.00000000001 + 5 * 1.29287 * T4 * 0.000000000000001) * WTFracSO2
'{Btu/Lb}

                                                    '{For H2O}
    CpVap = CpVap + (0.45739 - 2 * 0.52512 * T1 * 0.0001 + 3 * 0.64594 * T2 * 0.0000001 - 4 *
2.0276 * T3 * 0.00000000001 + 5 * 2.3631 * T4 * 0.000000000000001) * WTFracH2O
'{Btu/Lb}

    TotalEnth = FlueGasEnthalpy(DegF, WTFracCO2, WTFracO2, WTFracN2, WTFracSO2,
WTFracH2O)
    DegF = T1 - 460 - (TotalEnth - Btu_LB) / CpVap
    T1 = DegF + 460
   Loop Until Abs(TotalEnth - Btu_LB) < 1
   FlueGasTAtGivenEnth = DegF
End Function

*****************************************************************************************
**********************************
```

APPENDIX A

PROBLEM 1 SOLUTION

UOP

ENGINEERING DEPARTMENT				date:	3/11/02	
Equipment Skill Center				Project:	**56000**	
				By:	**FRB**	
customer:	**ABC Oil Refining Company**					
location:	**Des Plaines, IL.**					
comments:	**Case 1 Fuel**					
	FUEL GAS PROGRAM					
	Version 1.11, 8/4/94					
FUEL GAS DATA	Refinery fuel gas					
adiabatic flame temp, °F	3466		FLUE GAS INFO		Btu/lb	
LHV of fuel, BTU/lb	20546		15%xs air		bwt	enthalpy
LHV, Btu/SCF	1706.16				1400	378.00
lbs of wet air/lb fuel	18.32				1500	408.91
owing @ Amb Air) Ft3 of wet air/lb fuel	244.26		BW temp, °F=		**1350**	363
air preheat temp, °F	60	% Radiant Efficiency=			63.76%	
air preheat content, btu/lb	0.00	FG out stack, °F=			**500**	116.52
Corrected heat content fuel, btu/lb	20546		Heat Loss=		**2.00%**	
Corrected heat content fuel, Btu/SCF	1706.2					
Note: ~P air side burner ~ (Mass Flow)^2 * Specific Volume (ft3/lb)			% Total Efficiency=		87.00%	
Site Elevation, feet	**385**					
Calc. Site Pressure, PSIG	14		Fuel MW=		31.47	
Component		mole%		Excess Air		
Hydrogen	H2	**30.000**		%		
Nitrogen	N2	**0.000**		**15**		
Oxygen	O2	**0.000**				
Water	H2O	**0.000**		Relative		
Carbon monoxide	CO	**0.000**				
Carbon dioxide	CO2	**0.000**		Humidity %		
Hydrogen sulfide	H2S	**0.000**		**0**		
Sulfur dioxide	SO2	**0.000**				
Ammonia	NH3	**0.000**		Ambient		
Methane	C1	**0.000**		Temp °F		
Ethane	C2	**0.000**		**60**		
Propane	C3	**70.000**				
n-Butane	nC4	**0.000**		Combustion		
2-Methylpropane	iC4	**0.000**		Air Temp °F		
n-Pentane	nC5	**0.000**		**60**		
2-Methylbutane	iC5	**0.000**				
22-Dimethylpropane	iC5	**0.000**		Fuel Gas		
n-Hexane	nC6	**0.000**		Temp °F		
2-Methylpentane	iC6	**0.000**		**60**		
3-Methylpentane	iC6	**0.000**				

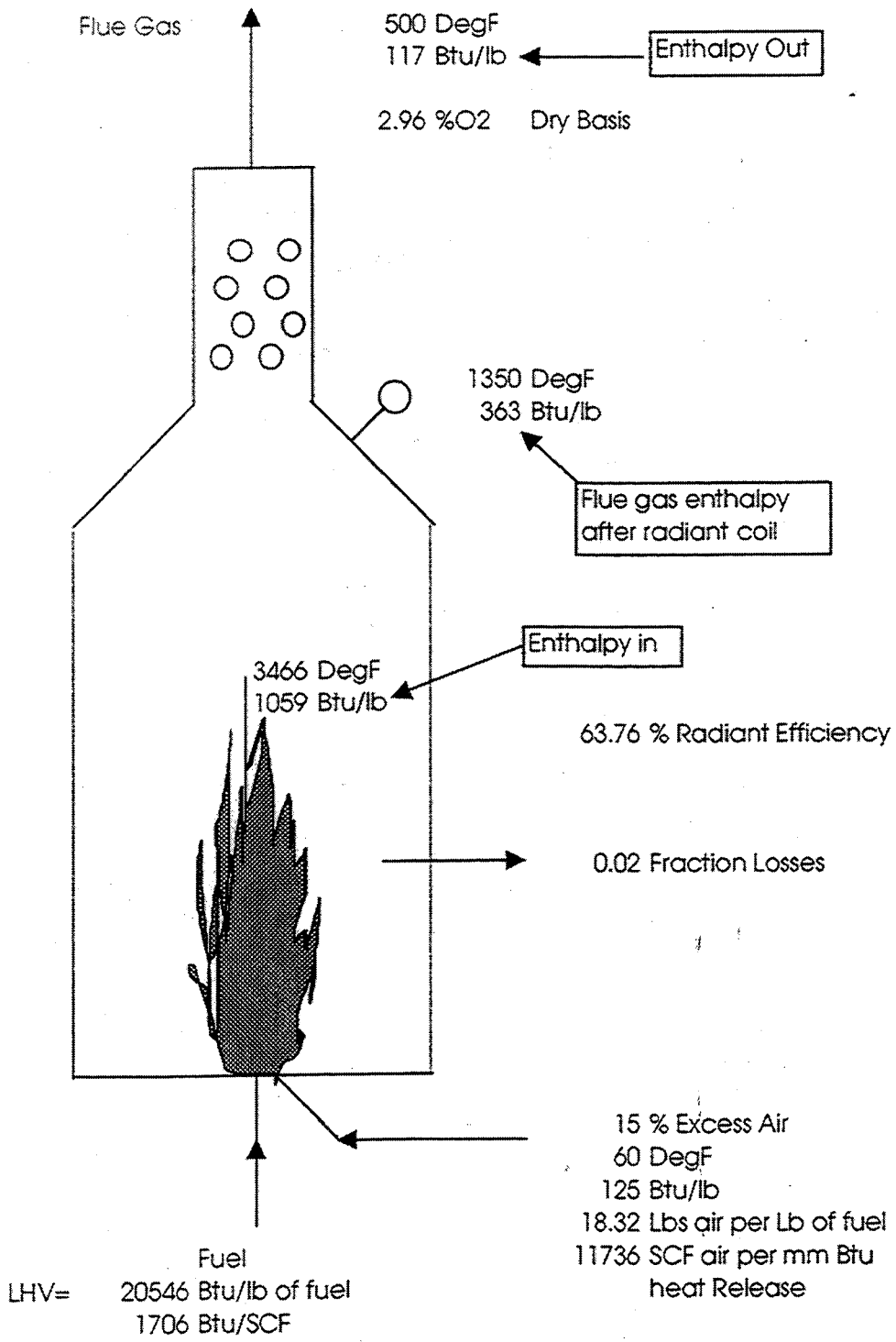
UOP						
ENGINEERING DEPARTMENT				date:	3/11/02	
Equipment Skill Center				Project:	**56000**	
				By:	**FRB**	
customer:	**ABC Oil Refining Company**					
location:	**Des Plaines, IL.**					
comments:	**Case 1 Fuel**					
22-Dimethylbutane	iC6	**0.000**		Flue Gas MW=	28.24	
23-Dimethylbutane	iC6	**0.000**			ON A WET BASIS	
n-Heptane	nC7	**0.000**		IN FLUE GAS	WT %	VOLUME %
Ethene	C2=	**0.000**		CO2	15.20	9.75
Propene	C3=	**0.000**		SO2	0.00	0.000
Isobutene	iC4=	**0.000**		H2O	9.18	14.39
1-Butene	nC4=	**0.000**		O2	2.88	2.54
		100.000		N2	72.74	73.32
				Sum=	100.00	100.00
					ON A DRY BASIS	
				IN FLUE GAS		
				CO2	16.73	11.39
				SO2	0.00	0.00
				H2O	0.00	0.00
				O2	3.17	2.96
				N2	80.09	85.65
				Sum=	100	100

Heater Efficiency and Duty

GAS FIRING (Data August 31, 1995)

$$\text{Total Efficiency} = \frac{(\text{Enthalpy in} * (1 - \text{fraction loss}) - \text{Enthalpy out})}{\text{Enthalpy in}}$$

87.00 % Total Efficiency



Heater Efficiency and Duty
GAS FIRING

Fuel flow rate=	**14.47** MSCFH 1202 lb/h	Flue gas rate=	23219 lb/h

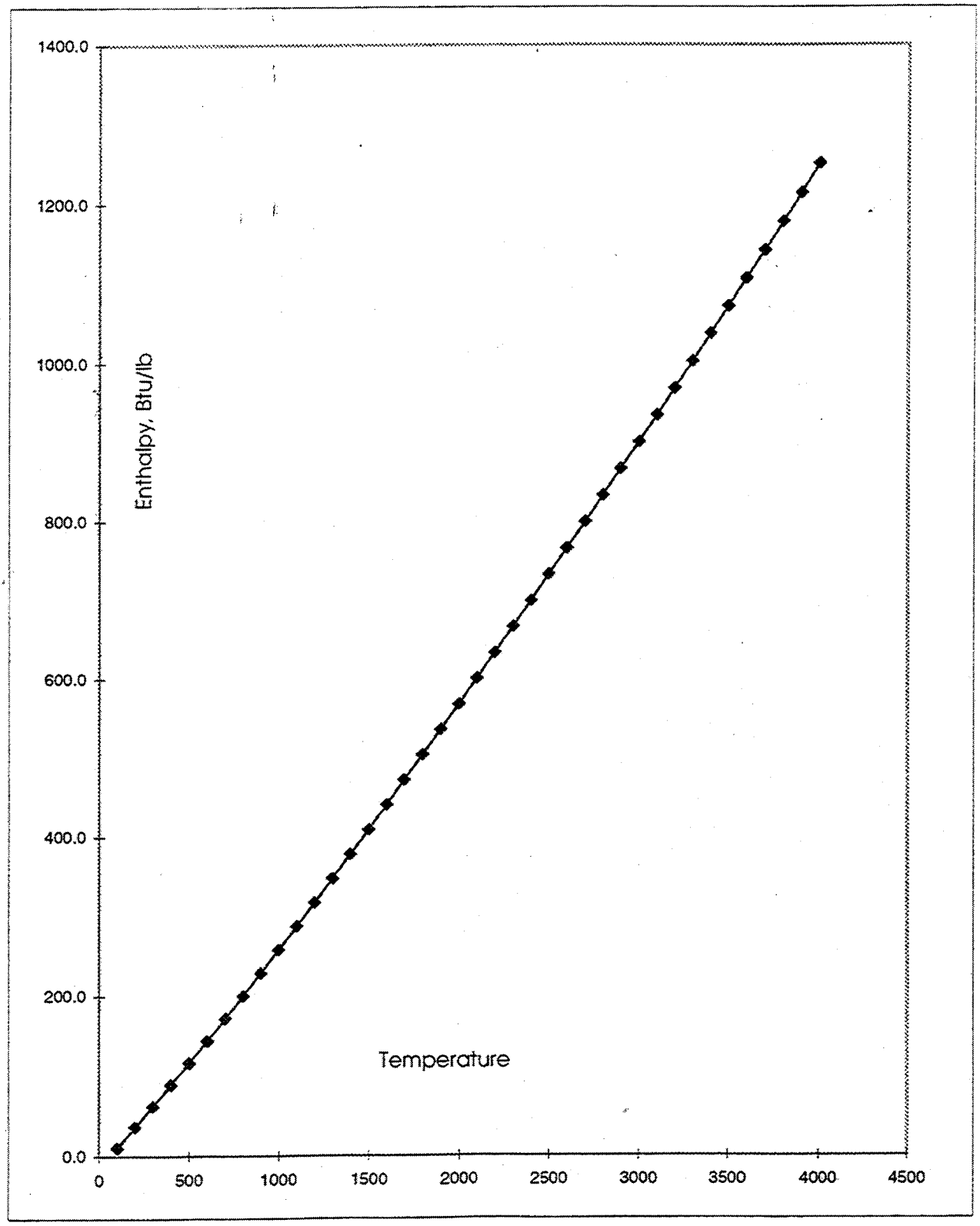
Radiant Duty=	16.17 mm btu/h
Convection Duty=	5.72 mm btu/h
Total Heater Duty=	21.89 mm btu/h

Heater Enthalpy
GAS FIRING

IN FLUE GAS	WT %
CO2	15.2
SO2	0.0
H2O	9.2
O2	2.9
N2	72.7
Sum=	100.0

Temperature	Enthalpy
100	10.3
200	36.4
300	62.7
400	89.4
500	116.5
600	144.0
700	171.9
800	200.1
900	228.8
1000	257.9
1100	287.3
1200	317.2
1300	347.4
1400	378.0
1500	408.9
1600	440.1
1700	471.6
1800	503.4
1900	535.5
2000	567.8
2100	600.2
2200	632.9
2300	665.7
2400	698.7
2500	731.9
2600	765.2
2700	798.6
2800	832.1
2900	865.7
3000	899.5
3100	933.5
3200	967.6
3300	1001.9
3400	1036.4
3500	1071.1
3600	1106.2
3700	1141.7
3800	1177.5
3900	1213.9
4000	1250.8

Ethalpy Profile



Problem Solutions:

Problem 1:

A) If done on a 100 lb-moles/hour basis: The fuel will contain 30 moles of H2 and 70 moles of C3H8. Changing this to weight:

H2 30 moles/h * 2.016 lb/mole = 60.48 lbs/h

C3H8 70 moles/h*44.097 lb/mole=3086.8 lbs/h

Total pounds of fuel = 3147.3 lb/h

B) Heat release from combustion:

60.48 lb/h * 51623 btu/lb = 3,122,159 btu/h

3086.8 lb/h * 19937 = 61,541,532 btu/h

Total heat release = 64,663,690 btu/h

Net heating value = 64,663,690 btu/h / 3147.3 lb/h = 20546 btu per pound of fuel fired.

Fuel enthalpy = 0 btu/lb

Air enthalpy = 0 btu/lb

C) Flue gas flow rate:

$H2+1/2*O2 \rightarrow H2O$

30 moles + 15 moles→ 30 moles

$C3H8+5*O2 \rightarrow 3*CO2+4*H2O$

70 moles + 350 moles→210 moles + 280 moles

365 moles of O2 required

79/21 * 365 = 1373.1 moles N2 required

at 15% excess air:

1.15 * 365 moles/h = 419.75 moles/h of O2 is required

1.15 * 1373.1 moles/h = 1579 moles/h of N2 passes through the heater

419.75 moles/h * 31.999 lbs/mole = 13428 lbs/h of O2

1579 moles/h * 28.013 lbs/mole = 44234 lbs/h of N2

Total lbs/h of flue gas =57662 lbs/h of dry air

D) Flue gas composition:

365 moles of O2 is consumed to create CO2 and H2O, therefore 420 - 365 = 55 moles of H2O remains in the flue gas.

	Moles/h	Molecular weight	lbs/h	Mole Fraction	Weight Fraction
CO2	210	44.010	9242	0.0975	0.1520
H2O	310	18.015	5585	0.1439	0.0918
N2	1579	28.013	44233	0.7331	0.7273
O2	55	31.999	1760	0.0255	0.0289
Total	2154		60820	1.00	1.00

E) At 3000 °F the enthalpy of the flue gas is: 899.5 btu/ lb:

HVap = (4.778 + 0.11443 * 3460 + 1.0113 * 3460^{2}* 0.0001 - 0.2649 * 3460^{3}* 0.0000001 + 0.34706 * 3460^{4} * 0.00000000001 - 0.1314 * 3460^{5}* 0.000000000000001) * 0.152 - (88.151) * 0.152
for CO2 at 3000°F=130.5

HVap = (-2.463 + 0.45739 * 3460 - 0.52512 * 3460^{2}* 0.0001 + 0.64594 * 3460^{3}* 0.0000001 - 2.0276 * 3460^{4} * 0.00000000001 + 2.3631 * 3460^{5}* 0.000000000000001) *0.0918 - (228.87) * 0.0918
for CO2 at 3000°F=152.8

HVap = (-0.934 + 0.2552 * 3460 - 0.1779 * 3460^{2}* 0.0001 + 0.1589 * 3460^{3}* 0.0000001 - 0.32203 * 3460^{4} * 0.00000000001 + 0.15893 * 3460^{5}* 0.000000000000001) * 0.7273 - (128.964) * 0.7273
for N2 at 3000°F=593.2

HVap = (-0.982 + 0.22749 * 3460 - 0.3731 * 3460^{2}* 0.0001 + 0.4831 * 3460^{3}* 0.0000001 - 1.85243 * 3460^{4} * 0.00000000001 + 2.47488 * 3460^{5}* 0.000000000000001) * 0.0289 - (112.757) * 0.0289
for O2 at 3000°F=23.1

At 4000 °F the enthalpy of the flue gas is: 1250.8 btu/ lb

$$SpecificEnthalpy = \frac{(NetHeating + FuelEnthalpy) * FuelFlowRate + AirEnthalpy * AirflowRate}{FlueGasFlowRate}$$

Specific enthalpy of the flue gas is: ((20546 + 0)*3147.3 + 0*57662) / 60820 = 1063 btu/lb

Using a linear interpolation between 3000 °F and 4000 °F:

$$\frac{4000-3000}{1250.8-899.5}=\frac{2.847\,DegF}{Btu}$$

(1063 - 899.5)*2.847=465°F Added to 3000°F

gives a theorectical flame temperature of 3465 °F.

F) At 1350 °F the enthalpy of the flue gas is: 363 btu/lb

$$\%\,Efficiency=\frac{(Specific\_Enthalpy\_in*(1-Fraction\_loss)-Specific\_Enthalpy\_out)}{Specific\_Enthalpy\_in}*100$$

Radiant section thermal efficiency = (1063*0.98 - 363) / 1063 *100 = 63.9%

APPENDIX B

PROBLEM 2 INFORMATION

Problem 2:

1. Since the duty based on flue gas temperatures and fuel flow rate is quite close to the duty obtained from the process side heat and weight balance, the temperatures and fuel flow rate appear to be quite reliable. In addition, the amount of excess O2 being measured appears to be correct.

2. The O2 measured is simulated by changing the percent excess air to achieve the O2 measured.

3. The measured temperature out the convection section, 880°F, is correct. This is 350°F above the original heater design

4. The radiant section flux is higher than the design duty flux yet the reboiler heat pickup is less than design. This means the convection section is not operating very effectively. There is probably fouling or dust collection on the tubes in the convection section. Possibly some of the convection tube extended surface has been damaged. History of unit operation from when this reboiler was started up should be checked.

5. The flue gas temperature leaving the radiant section is only slightly higher than design and the flux calculated from operating conditions is slightly higher than design. The radiant section of this heater appears to be working fine.

6. At the next turnaround the convection section of this heater should be opened and checked for damage and fouling. If there is no damage, all that would have to be done is tube cleaning with a steam lance.

UOP Fuels Programs

Case	Contractor Design	Using Measured Temperat-ures and flue gas enthalpy balances
Heater Duty, mm Btu/h	11.54	9.80
Convection Duty, mm Btu/h		2.52
Volume flow to heater, BPSD	19740	16000
Mass Flow to heater, Lbs/h	212150	169835
Inlet temperature, F	433	421
Outlet temperature, F	441	438
Outlet pressure, Psig	168	~165
Average Radiant flux, Btu/h-ft2	10000	10855
Peak flux, Btu/h-ft2		
Volumetric heat release, Btu/ft3-h	8880	9405
Calculated Tube wall, F		
Measured Tube wall, F		
Calculated burner heat release, mm Btu/h	13.19	13.97
Calculated fuel flow, SCFH	13491	13772
Measured fuel flow, SCFH		15000
Calculated flue gas flow, lbs/h		16990
Calculated Bridgewall Temperature, F	1330	
Measured Bridgewall Temperature, F		1380
Calculated Stack Temperature, F	530	
Measured Stack Temperature, F		880
% Excess Air	15	51
Mole% O2 on a dry basis	3.0	7.5
Measured O2, volume %		7.5

uop

ENGINEERING DEPARTMENT				date:	3/11/02	
Equipment Skill Center				Project:	**56000**	
				By:	**FRB**	
customer:	**ABC Oil Refining Company**					
location:	**Des Plaines, IL.**					
comments:	**Case 2 Fuel**					
	FUEL GAS PROGRAM					
	Version 1.11, 8/4/94					
FUEL GAS DATA	Refinery fuel gas					
adiabatic flame temp, °F	2713		FLUE GAS INFO			Btu/lb
LHV of fuel, BTU/lb	18647		51%xs air		bwt	enthalpy
LHV, Btu/SCF	910.32				1400	376.84
lbs of wet air/lb fuel	22.20				1500	407.57
owing @ Amb Air) Ft3 of wet air/lb fuel	297.90		BW temp, °F=		**1380**	371
air preheat temp, °F	60		% Radiant Efficiency=		51.63%	
air preheat content, btu/lb	0.00		FG out stack, °F=		**880**	222.64
Corrected heat content fuel, btu/lb	18647		Heat Loss=		**2.00%**	
Corrected heat content fuel, Btu/SCF	910.3					
Note: ~P air side burner ~ (Mass Flow)^2 * Specific Volume (ft3/lb)			% Total Efficiency=		70.15%	
Site Elevation, feet	**385**					
Calc. Site Pressure, PSIG	14		Fuel MW=		18.50	
Component		mole%	Excess Air			
Hydrogen	H2	**20.000**	%			
Nitrogen	N2	**3.000**	**51**			
Oxygen	O2	**0.000**				
Water	H2O	**0.000**	Relative			
Carbon monoxide	CO	**0.000**				
Carbon dioxide	CO2	**4.000**	Humidity %			
Hydrogen sulfide	H2S	**0.000**	**100**			
Sulfur dioxide	SO2	**0.000**				
Ammonia	NH3	**0.000**	Ambient			
Methane	C1	**51.000**	Temp °F			
Ethane	C2	**18.000**	**60**			
Propane	C3	**3.000**				
n-Butane	nC4	**1.000**	Combustion			
2-Methylpropane	iC4	**0.000**	Air Temp °F			
n-Pentane	nC5	**0.000**	**60**			
2-Methylbutane	iC5	**0.000**				
22-Dimethylpropane	iC5	**0.000**	Fuel Gas			
n-Hexane	nC6	**0.000**	Temp °F			
2-Methylpentane	iC6	**0.000**	**60**			
3-Methylpentane	iC6	**0.000**				

uop						
ENGINEERING DEPARTMENT				date:	3/11/02	
Equipment Skill Center				Project:	**56000**	
				By:	**FRB**	
customer:	**ABC Oil Refining Company**					
location:	**Des Plaines, IL.**					
comments:	**Case 2 Fuel**					
22-Dimethylbutane	iC6	**0.000**		Flue Gas MW=	27.94	
23-Dimethylbutane	iC6	**0.000**			ON A WET BASIS	
n-Heptane	nC7	**0.000**		IN FLUE GAS	WT %	VOLUME %
Ethene	C2=	**0.000**		CO2	10.66	6.77
Propene	C3=	**0.000**		SO2	0.00	0.000
Isobutene	iC4=	**0.000**		H2O	9.14	14.18
1-Butene	nC4=	**0.000**		O2	7.40	6.45
		100.000		N2	72.80	72.61
				Sum=	100.00	100.00
					ON A DRY BASIS	
				IN FLUE GAS		
				CO2	11.74	7.89
				SO2	0.00	0.00
				H2O	0.00	0.00
				O2	8.14	7.51
				N2	80.12	84.60
				Sum=	100	100

Heater Efficiency and Duty

GAS FIRING (Data August 31, 1995)

$$\text{Total Efficiency} = \frac{(\text{Enthalpy in} * (1 - \text{fraction loss}) - \text{Enthalpy out})}{\text{Enthalpy in}}$$

70.15 % Total Efficiency

Flue Gas

880 DegF
223 Btu/lb ← Enthalpy Out

7.51 %O2 Dry Basis

1380 DegF
371 Btu/lb

Flue gas enthalpy after radiant coil

Enthalpy in

2713 DegF
800 Btu/lb

51.63 % Radiant Efficiency

0.02 Fraction Losses

51 % Excess Air
60 DegF
126 Btu/lb
22.20 Lbs air per Lb of fuel
15668 SCF air per mm Btu heat Release

Fuel
LHV= 18647 Btu/lb of fuel
910 Btu/SCF

Heater Efficiency and Duty
GAS FIRING

Fuel flow rate=	**14.47** MSCFH 706 lb/h	Flue gas rate=	16390 lb/h

Radiant Duty=	7.03 mm btu/h
Convection Duty=	2.43 mm btu/h
Total Heater Duty=	9.45 mm btu/h

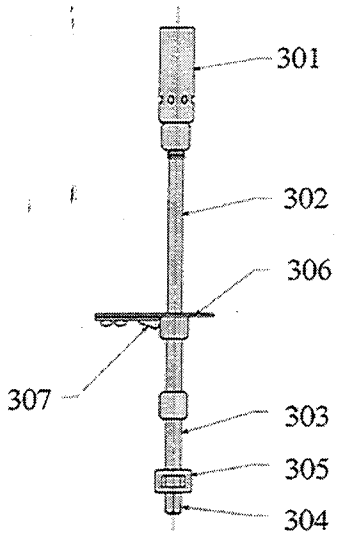
Training Services

Burners

Section Outline

- Pilot
- Gas
- Combination
- Low NO_x
- Emissions

Pilot - ST-1S Assembly



301 Pilot Tip
302 Pilot Nipple
303 Pilot Mixer
304 Orifice Spud
305 Air Door
306 Pilot Hub
307 Port Cover

FH 2004/7-3

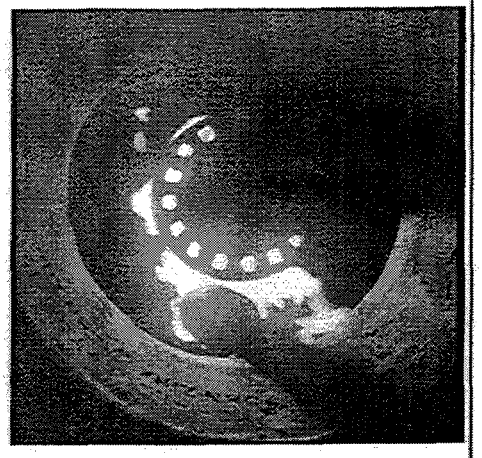
Pilot

- **UOP specifies continuous pilots for safety concerns**
- **Natural gas is recommended fuel**
- **John Zink ST-1S or equal**
- **Pilot is a small inspirated gas burner with glow plug head**
- **Works at lower fuel pressure than standard gas burners**
- **Air adjusted at inspirator to maintain good air mixing**

FH 2004/7-4

Pilot Burner Problems

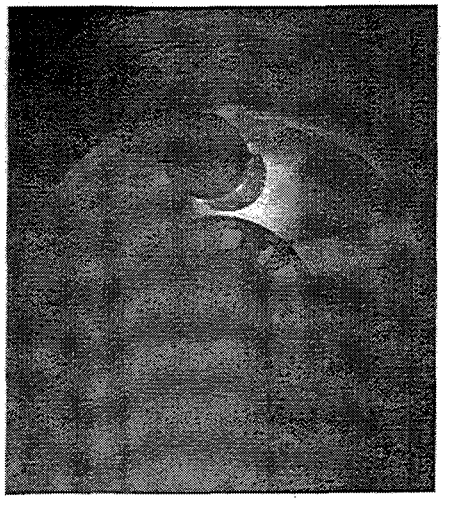
- Long Yellow Flames
- Prone to Blowout
- Insufficient Primary Air
- Open Air Register



FH 2004/7-5

Pilot Burners

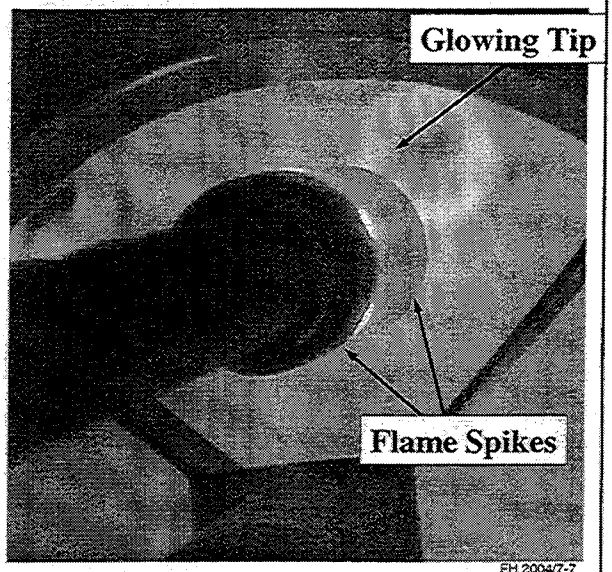
- Zink ST-1S
- Ignition Source
- Safety Device
- 90% Premix Burner
- Short Blue Flame
- Orange Color Tip
- Blue Flame Spikes on Side



FH 2004/7-6

Pilot Burners

- Callidus Pilot
- Similar to ST-1S



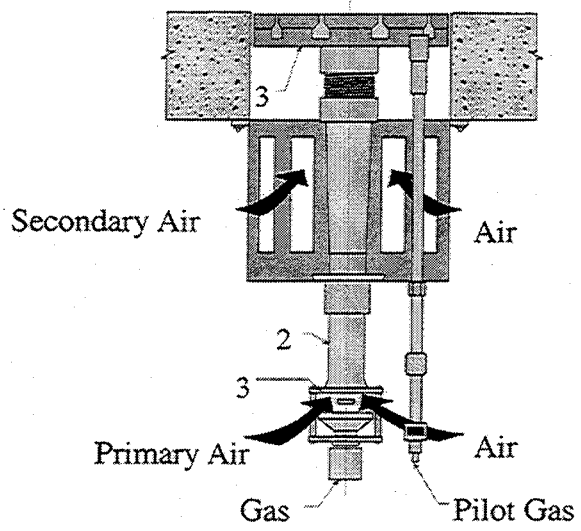
FH 2004/7-7

Premix Gas

- Inspirated Air Burners (Premix)
  - Advantages
    - Little fouling/plugging in tips
    - High efficiency air-fuel mixing
  - Disadvantages
    - Flashback (flame speed exceeds mixing speed)

FH 2004/7-8

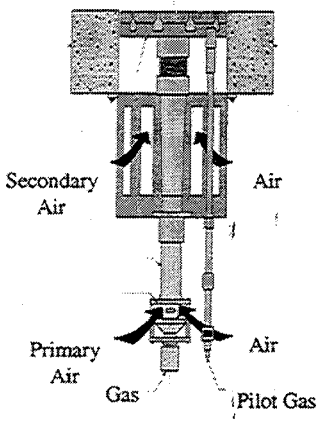
Premix Gas Burner



FH 2004/7-9

Premix Gas Burner

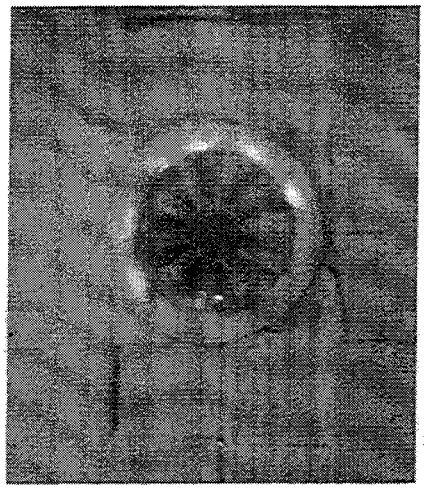
- **Oldest Burner Design**
- **Inspirates 30-50% of Combustion Air**
- **Self Correcting on Excess Air with Gas Pressure**
- **Large Gas Orifice - No Fouling**
- **High Noise**
- **Potential Flashback (Flame Speed Exceeds Mixing Speed)**
- **High NO_X**



FH 2004/7-10

Premix Gas Burner

- John Zink HEDV Burner
- Round Flame Premix
- Glowing Tile Indicates Improper Tip Position
- Gas Tips Should be 25/20 Cr/Ni



FH 2004/7-11

Premix Gas Burner

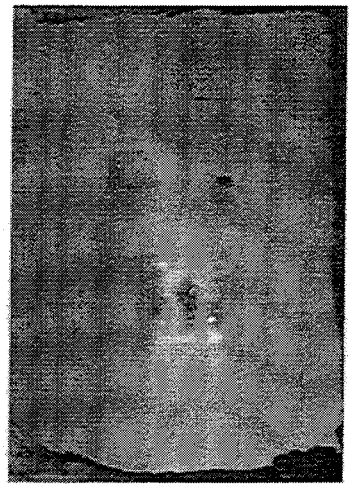
- John Zink PFH
- Flat Flame Premix
- Improper Round Tile
- Early 1970's Design



FH 2004/7-12

Premix Gas Burner

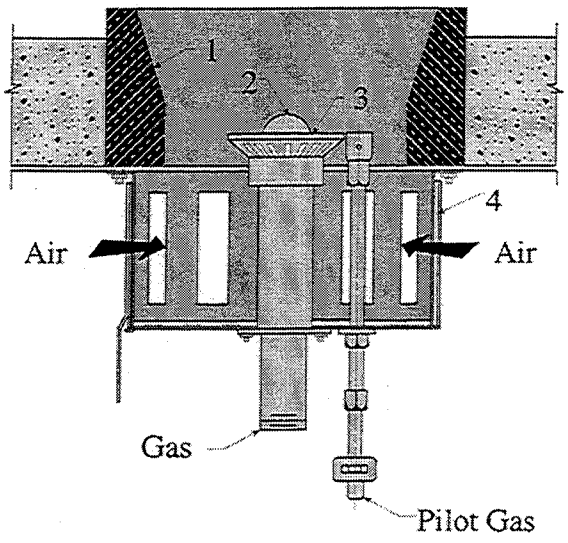
- John Zink PFH
- Flat Flame Premix
- Proper Rectangular Tile
- 1990's Revamp



Raw Gas

- Raw Gas or Nozzle Mix Burners
  - Advantages
    - High turndown ratio
    - No possibility of flashback
    - Can burn wide variety of fuels
  - Disadvantages
    - Fouling/plugging in tips
    - Lower efficiency air-fuel mixing

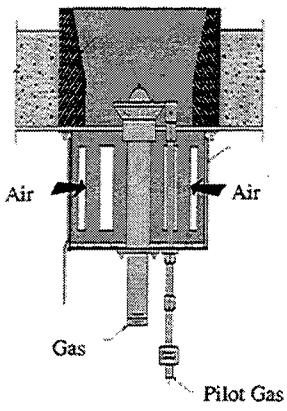
Raw Gas Burner



FH 2004/7-15

Raw Gas Burner

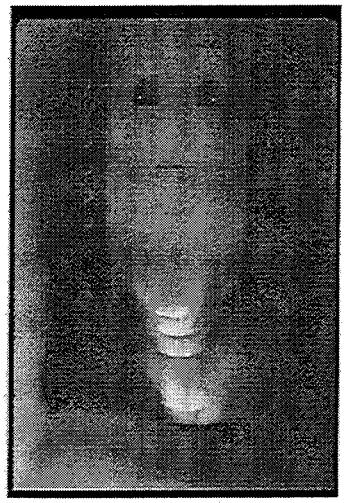
- **Simple Construction**
- **Can Burn Wide Range of Fuels**
- **Short Flames**
- **High Turndown Ratio**
- **No Possibility of Flashback**
- **Potential Fouling of Tips**



FH 2004/7-16

Raw Gas Burner

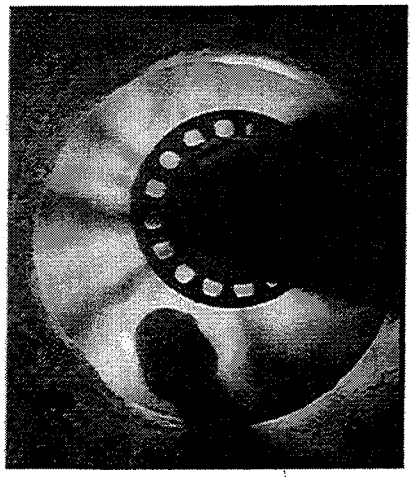
- 110 ppm NO_X
- Slight Haze on End of Flame
- Low Excess Air (2%)
- Hot Flame Temperature
- Bright Refractory Color



FH 2004/7-17

Raw Gas Burner

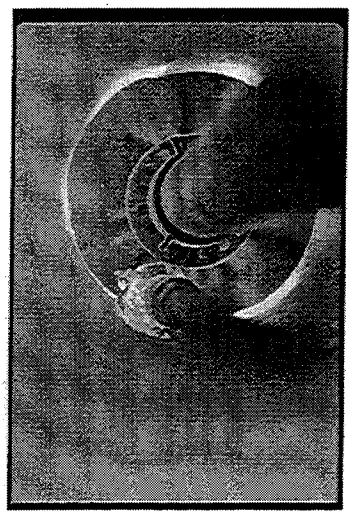
- Improper Tile Installation
- High Air Close to Pilot



FH 2004/7-18

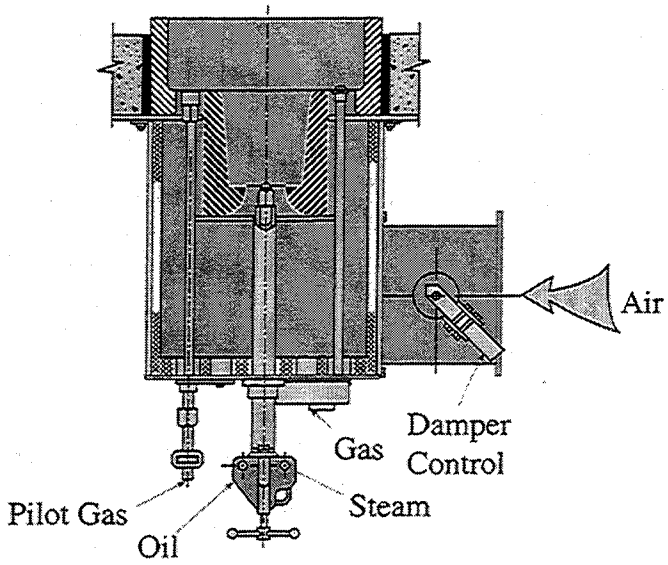
Raw Gas Burner

- **Flame Cone Located too Low**



FH 2004/7-19

Combination Burner

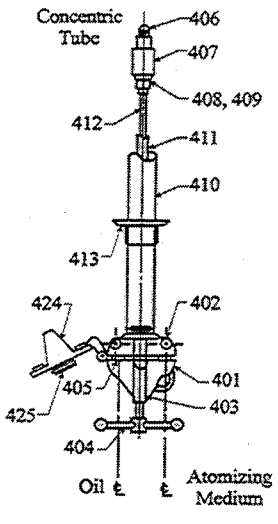


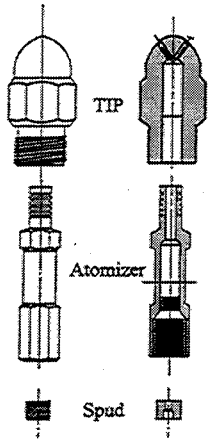
FH 2004/7-20

Oil Burners

- **Oil gun atomizes oil to fine spray of oil droplets that approximate gas firing**
- **Atomization done with steam, air, or mechanically (high pressure)**

Oil Burner

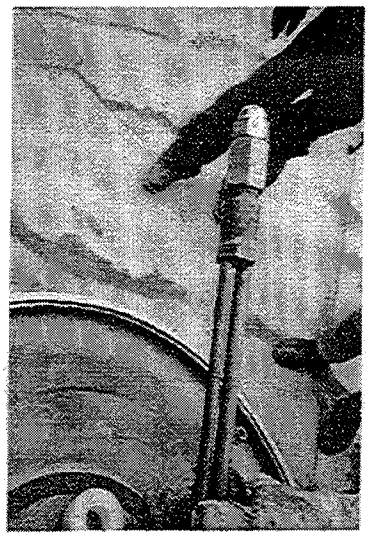




Item	Description
401	Oil Body
402	Oil Body Receiver
403	Clevis
404	Clevis Handle
405	Gasket (2 Req'd)
406	Oil Tip
407	Sleeve
408	Atomizer
409	Atomizer Bushing
410	Guide Tube
411	Steam Tube
412	Oil Tube
413	Hub
424	Pressure Seal (Opt)
429	O'Ring-Pressure Seal (2 Req'd)

Dual Tube Oil Gun

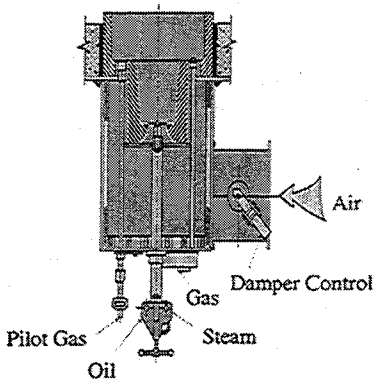
- Used for light oils
- Prevents flashing and pressure fluctuation with light oils and naphtha



FH 2004/7-23

Combination Burner

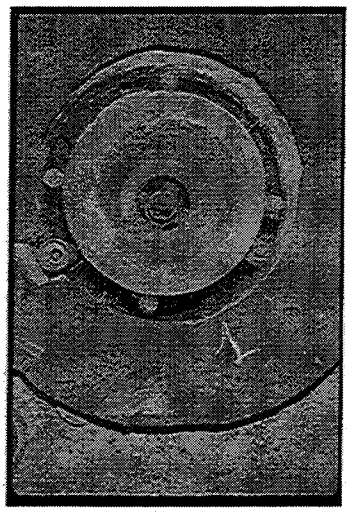
- Combines Raw Gas and Oil Burner
- Use Either Fuel When in Service
- Not Recommended to Run Both Fuels at Same Time
- Light Oil Gun Off Main Gas - Not Pilot
- Startup on Gas Until Firebox is Hot



FH 2004/7-24

Combination Burner

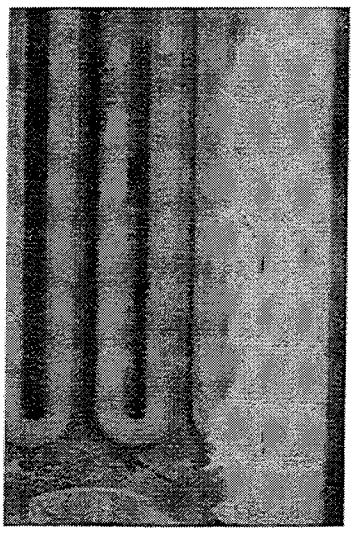
- Oil Gun in Center of Primary Tile, 1" (25 mm) Below Tile Ledge
- Gas Tips Located Between Primary and Secondary Tiles



FH 2004/7-25

Combination Burner

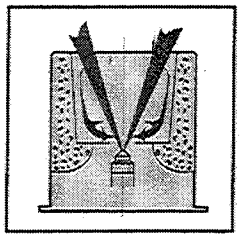
- Oil Firing on Right Burner
- High Emissivity Oil Flame
- Gas Firing on Left Burner
- Low Emissivity Gas Flame



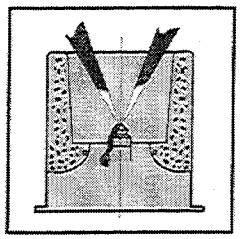
FH 2004/7-26

Oil Gun Location

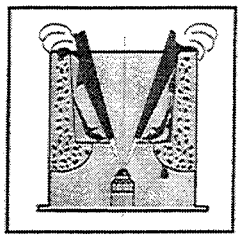
- **Proper position is critical to proper spray pattern and air mixing**



Proper location



Too far in

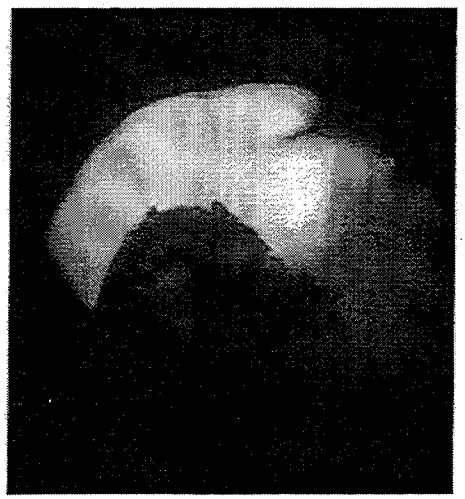


Too far back

FH 2004/7-27

Combination Burner

- **Looking at Oil Flame Through Primary Tile**
- **Flame Recirculation**



FH 2004/7-28

Combination Burner

- **Looking at Oil Flame Through Primary Tile**
- **Flame Recirculation with No.2 Oil with Staged Air Burner**



FH 2004/7-29

Combination Burner

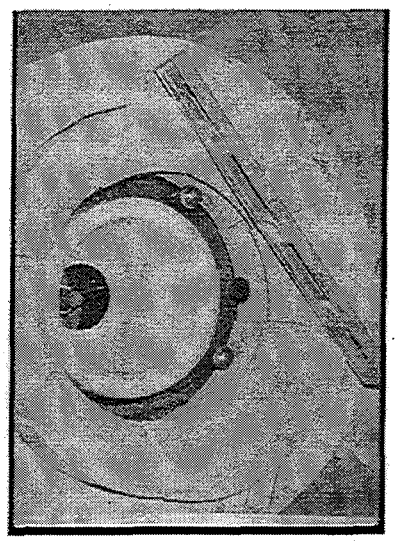
- **Oil Ash (Sodium Vanadates) Attack Refractory**
- **Use 94% Alumina on Primary Tile**



FH 2004/7-30

Combination Burner

- 94% Alumina on Primary Tile
- 60% Alumina on Secondary Tile
- Use Plywood Template to Align Secondary Tile Pieces



FH 2004/7-31

Burner Excess Air Adjustment

- Opening/closing air register or damper
- Registers - large clearances - will pass 50% of design air when fully closed
- Dampers - tighter clearances - will pass 12-25% of design air when fully closed
- Keep all burners lit if possible
- Minimum pressure drop 0.10-0.15 in H_2O
- 80% of burner pressure drop in throat - 20% in damper/register

FH 2004/7-32

Burner Air Controls

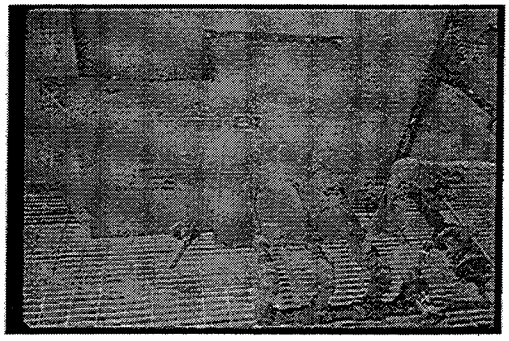
- Concentric Register
- Prone to Sticking
- Difficult to Adjust
- Leaks 50% Air when Closed



FH 2004/7-33

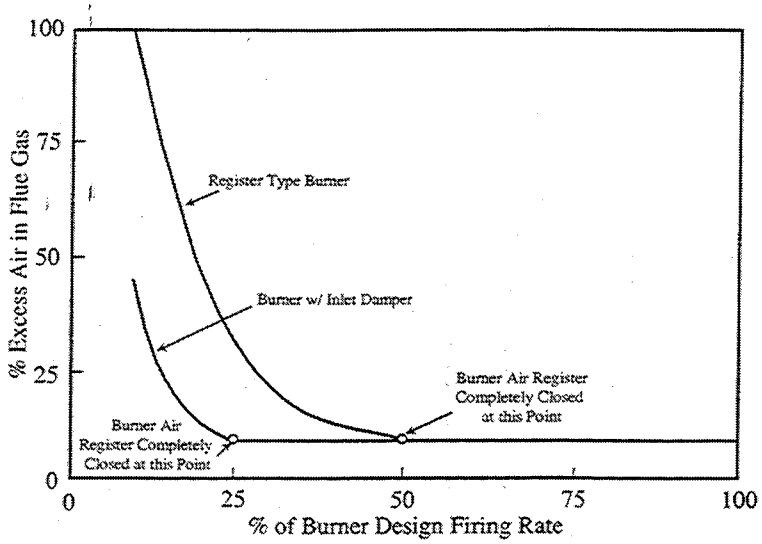
Burner Air Controls

- Damper Control
- Tight Shutoff
- Positive Position
- Good Excess Air Control



FH 2004/7-34

Burner Excess Air Adjustment



FH 2004/7-35

Emissions

- **National/local governments restrict emissions of:**
  - **NO_x**
  - **SO_x**
  - **CO**
  - **Particulate**
  - **Unburned hydrocarbons**
  - **Heavy metals**
  - **Others**

FH 2004/7-36

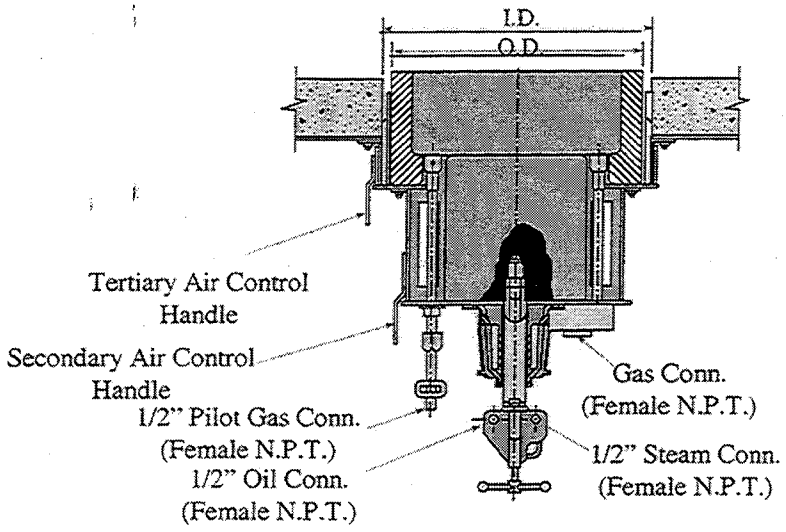
Training Services

Low NO_x Burners

Low NO_x

- Time - Temperature - Turbulence key to NO_x formation
- Flue Gas Recirculation
- Staged Air Burners
- Staged Fuel Burners

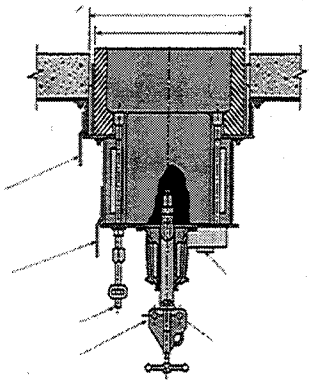
Staged Air Burner



FH 2004/7-39

Staged Air Burner

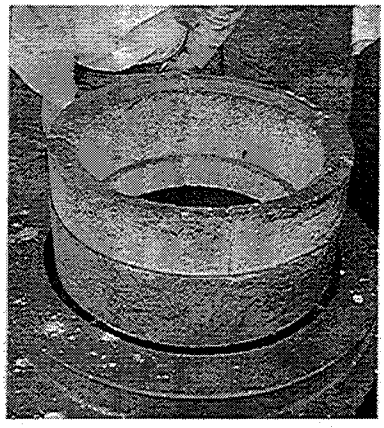
- **Slow Combustion**
- **50% Tertiary Air**
- **Excess Fuel in Primary Results in Lower Flame Temperature**
- **Very Long Flames**



FH 2004/7-40

Staged Air Burner

- 50% Combustion Air Outside Burner Tile
- Do Not Puck Annular Area



FH 2004/7-41

Staged Air Burner

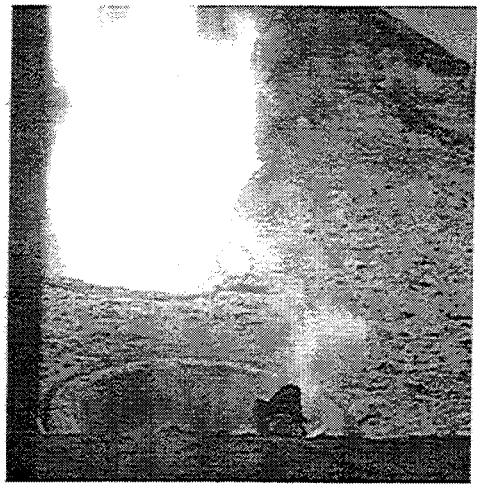
- Slow Burning
- Long Flames



FH 2004/7-42

Staged Air Burner

- Coking on Oil Tip



FH 2004/7-43

Staged Air Burner

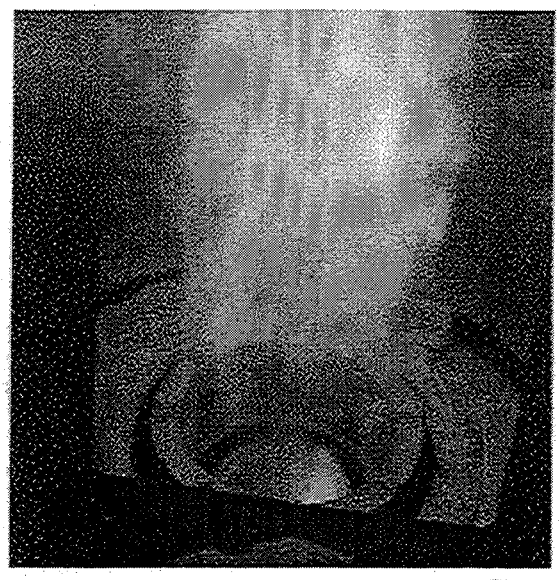
- Firing Gas
- Long Flames
- Cracking in Flame (Yellow Color)



FH 2004/7-44

Staged Air Burner

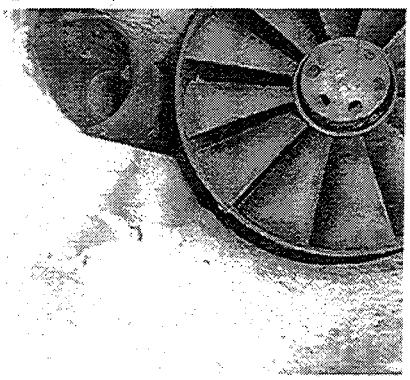
- Burning No. 2 Oil
- Bright color on primary tile indicates problem



FH 2004/7-45

Suspended Flame Burner

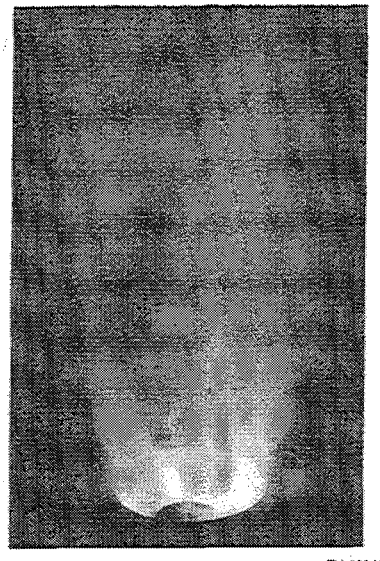
- Swirl Stabilizer replaces primary tile
- Ignition occurs above Swirler
- High air side design pressure drop
  6" H_20 (150 mm)



FH 2004/7-46

Suspended Flame Combination Burner

- John Zink PSFR
- Burning Methane in Test Furnace
- Staged fuel burner



FH 2004/7-47

Suspended Flame Combination Burner

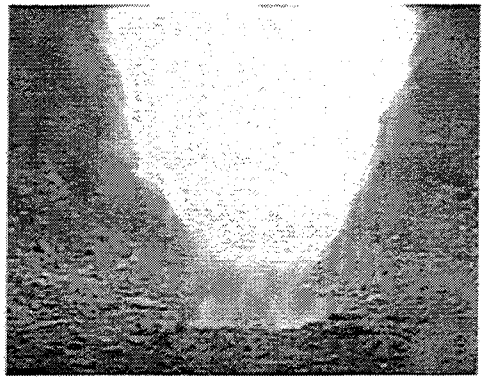
- John Zink PSFRC
- Burning No. 6 Oil in Test Furnace
- Ignition occurs above Swirler



FH 2004/7-48

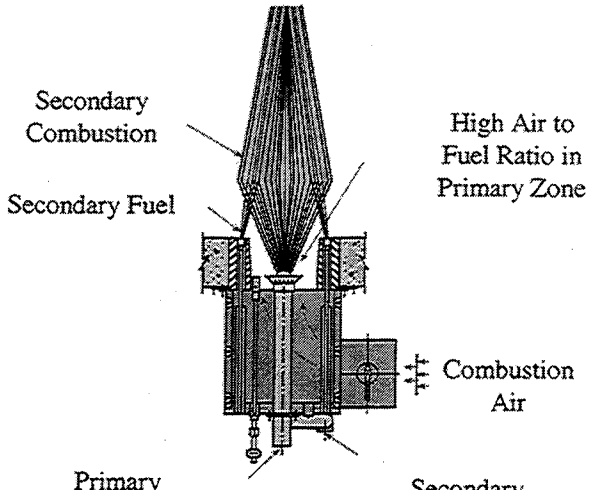
Suspended Flame Combination Burner

- **John Zink PSFRC**
- **Only Burner suitable for oil and gas firing simultaneously**
- **High air side pressure drop 3" H_20 (75 mm) makes up for low fuel pressures**
- **Short, Wide Flame**



FH 2004/7-49

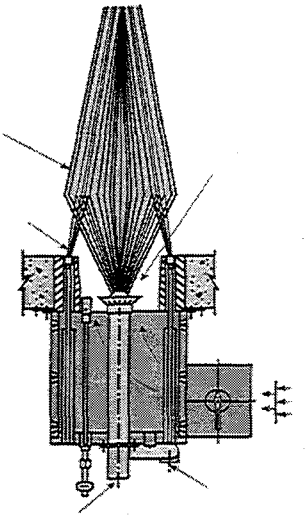
Staged Fuel Burner



FH 2004/7-50

Staged Fuel Burner

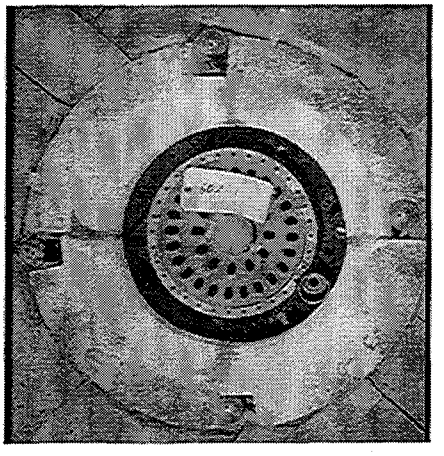
- Staged Combustion
- 30% Primary Fuel
- 70% Secondary Fuel
- High Excess Air in Primary Tip Results in Low Flame Temperature
- Low O_2 in Secondary Tip Results in Longer Combustion Time
- 50% Longer Flame
- 40 ppm NO_X



FH 2004/7-51

Staged Fuel Burner

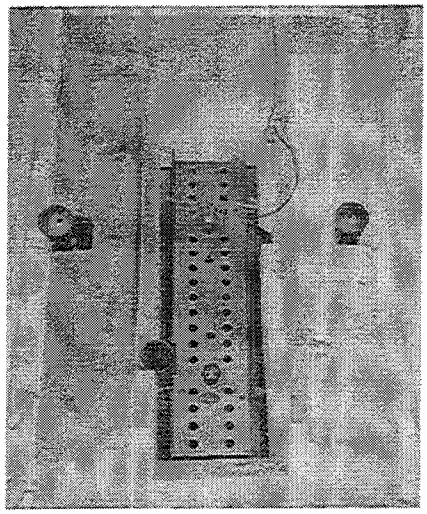
- Primary Tips Aligned to Secondary Tips
- Proper Tile Installation
- Uniform Annular Area



FH 2004/7-52

Staged Fuel Burner

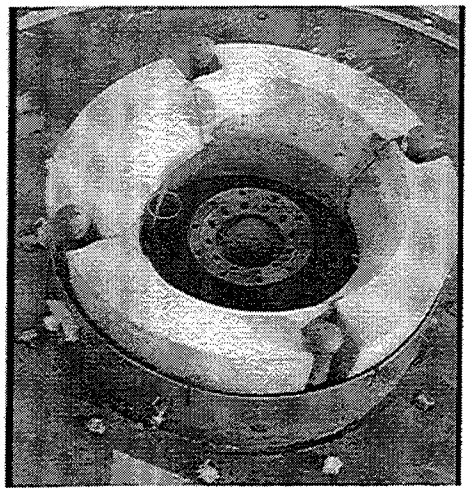
- Flat Flame Version- John Zink PSFFG
- Used for smaller heat releases, 3-5 MM Btu/h(0.9-1.5MW)
- Has two staged gas tips so holes can be larger to prevent plugging



FH 2004/7-53

Staged Fuel Burner

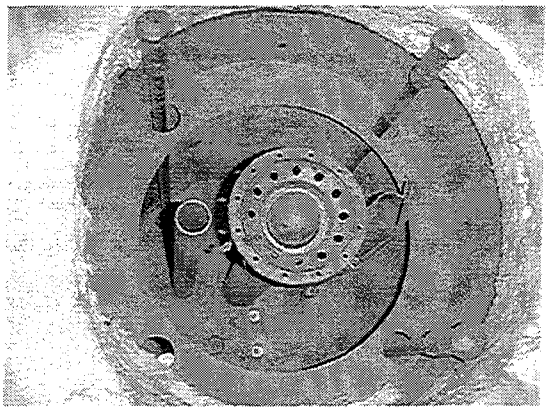
- Tile Alignment Ring Ensures Proper Installation
- John Zink PSFG



FH 2004/7-54

Staged Fuel Burner

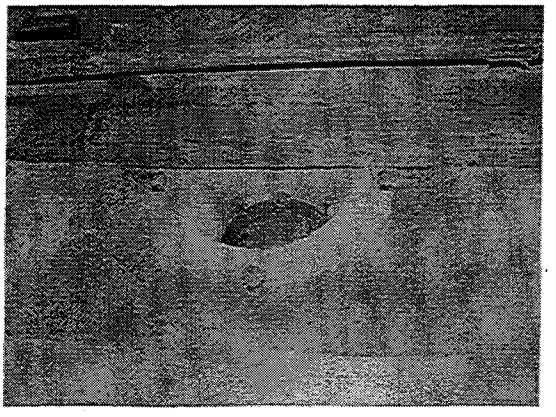
- Tile Alignment Ring Ensures Proper Installation
- 310 SS waste gas nozzle
- Limit waste gas to 3-5% of burner heat release per nozzle



FH 2004/7-55

Staged Fuel Burner

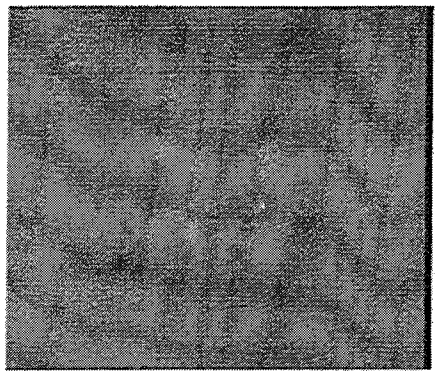
- John Zink PSFR- four individual primary gas tips
- Better turndown operations
- Mistake-tile is not level
- Mistake-expansion joint between burner tile and floor filled with castable - should be fiber



FH 2004/7-56

Staged Fuel Burner

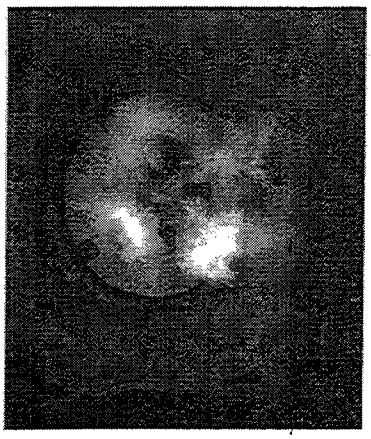
- Operating Burner at 20 psig (1.4 kg/cm^2)
- Blue Flame
- Methane Fuel



FH 2004/7-57

Staged Fuel Burner

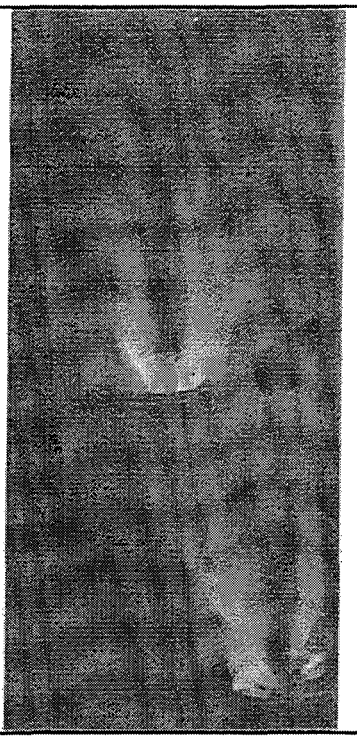
- Turndown Operation at 3 psig (0.21 kg/cm^2)
- Yellow Flame
- Cracking in Flame



FH 2004/7-58

Staged Fuel Burner

- Operating Burner at 32 psig (2.2 kg/cm²)
- Low excess air 2% O_2
- Soft Hazy Flame Ends at low excess air
- Blue flame (good fuel-air mixing)
- Refinery Fuel



Staged Fuel Gas Burner

- Improper Secondary Tip Location in center of burner



FH 2004/7-60

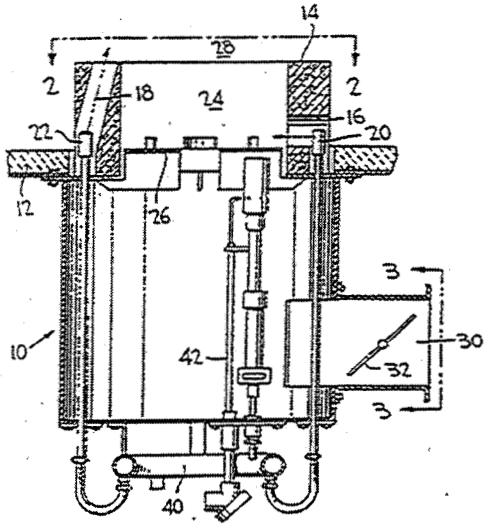
Staged Fuel Burner

- **Improper Secondary Tip Location**
- **Poor Combustion**



FH 2004/7-61

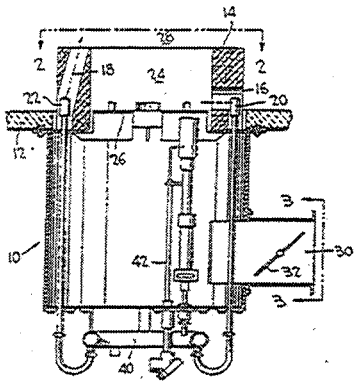
Ultra Low NO_x Burner



FH 2004/7-62

Ultra Low NO_x Burner

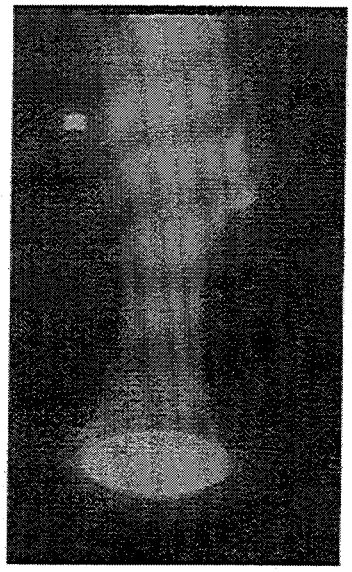
- Flue Gas Recirculation
- Lowest NO_X
- Very Long Flames
- Potential Stability Problems



FH 2004/7-63

Ultra Low NO_x Burner

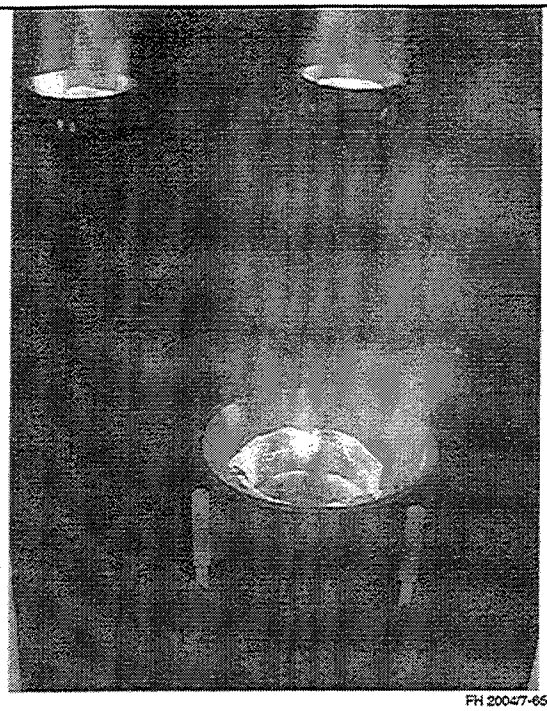
- Callidus LE Burner
- Hourglass Shaped Flame
- Low Temperature
- High Combustion Time
- 30 ppm NO_X



FH 2004/7-64

Ultra Low NO_x Burner

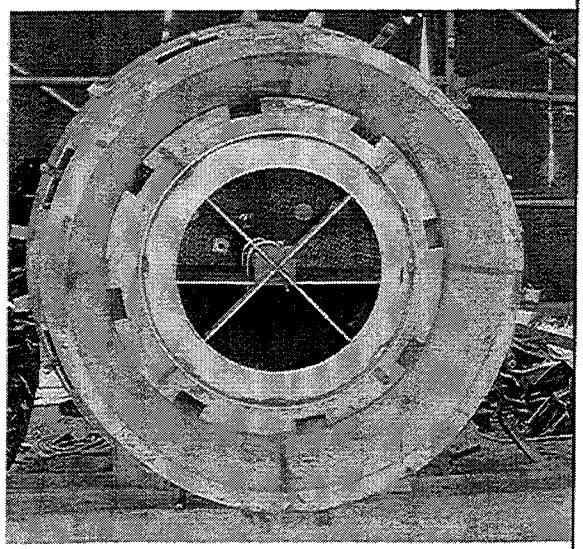
- Callidus LE Burner
- Damaged Flame Holder
- Prefer 1 diameter spacing between tiles
- NOx levels track floor temperatures



FH 2004/7-65

Ultra Low NO_x Burner

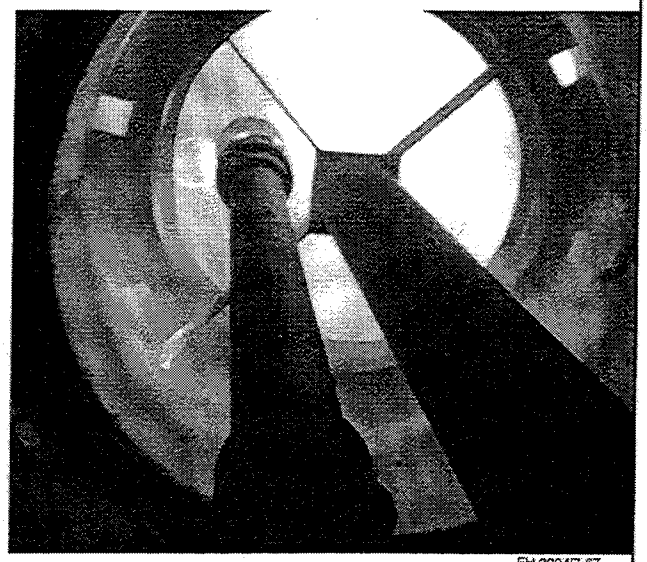
- Callidus LE Burner
- New Design Flame Holder, 8 Secondary Tips
- Tangential primary



FH 2004/7-66

Ultra Low NO_x Burner

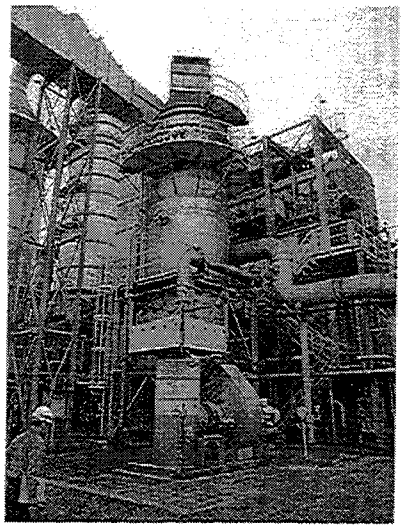
- Callidus LE Burner
- Flame is behind flameholder, It will burn up
- Flameholder needs to lowered



FH 2004/7-67

Catalytic De-NO_x

- Very Expensive
- Operates at 400-700°F
- Used in Retrofits when Ultra-Low NO_x cannot be used
- Coupled with air preheat system



FH 2004/7-68

NO_x Factors

1. Burner Style
2. Fuel Type
3. Excess Air
4. Air Temperature

US NOx STANDARD:	24-77 pp	(0.03-0.10 lb/MM Btu)
ECC STANDARDS	275 ppm	(450 mg/m^3)
ECC STANDARDS	354 ppm	(450 mg/m^3)
1995 ECC STANDAR	118 ppm	(150 mg/m^3)

Gas Firing @ 15% Excess Air

Standard Burner	110 ppm	(0.14 lb/MM Bt Ambient Air
	150 ppm	(0.20 lb/MM Bt 465*F Air Preheat
Staged Air Burner	80 ppm	(0.10 lb/MM Bt Ambient Air
	103 ppm	(0.13 lb/MM Bt 465*F Air Preheat
Staged Gas Burner	40 ppm	(0.05 lb/MM Bt Ambient Air
	64 ppm	(0.08 lb/MM Bt 465*F Air Preheat
Ultra Low NOx Burne	24 ppm	(0.03 lb/MM Bt Ambient Air
	38 ppm	(0.08 lb/MM Bt 465*F Air Preheat

Oil Firing @ 25% Excess Air

Standard Burner	280 ppm	(0.36 lb/MM Bt Ambient Air
	340 ppm	(0.44 lb/MM Bt 465*F Air Preheat
Staged Air Burner	210 ppm	(0.27 lb/MM Bt Ambient Air
	240 ppm	(0.31 lb/MM Bt 465*F Air Preheat

Burners

Efficient operation of the fired process heater is dependent upon the proper operation of the burner. A burner can be defined as a device that positions a flame in the desired location by delivering fuel and air with sufficient mixing energy to ensure continuous ignition and complete combustion.

For the purposes of this discussion, industrial burners have been classified as natural draft and forced draft. Premix gas burners, nozzle mix gas burners, fuel oil burners, and combination oil and burners will be considered.

Premix Gas Burners

In premix gas systems, the primary air and gas are mixed at somepoint upstream from the burner discharge ports by an inspirator mixer. The premix gas burner is then defined as a burner in which the nozzle or gas tip is supplied with gas and air from an upstream mixing device. Thus, the gas and air are mixed prior to reaching the burner nozzle. The burner tip or nozzle serves only as a flame holder, maintaining the flame in the desired location.

Figure 1
Premix Gas Burner

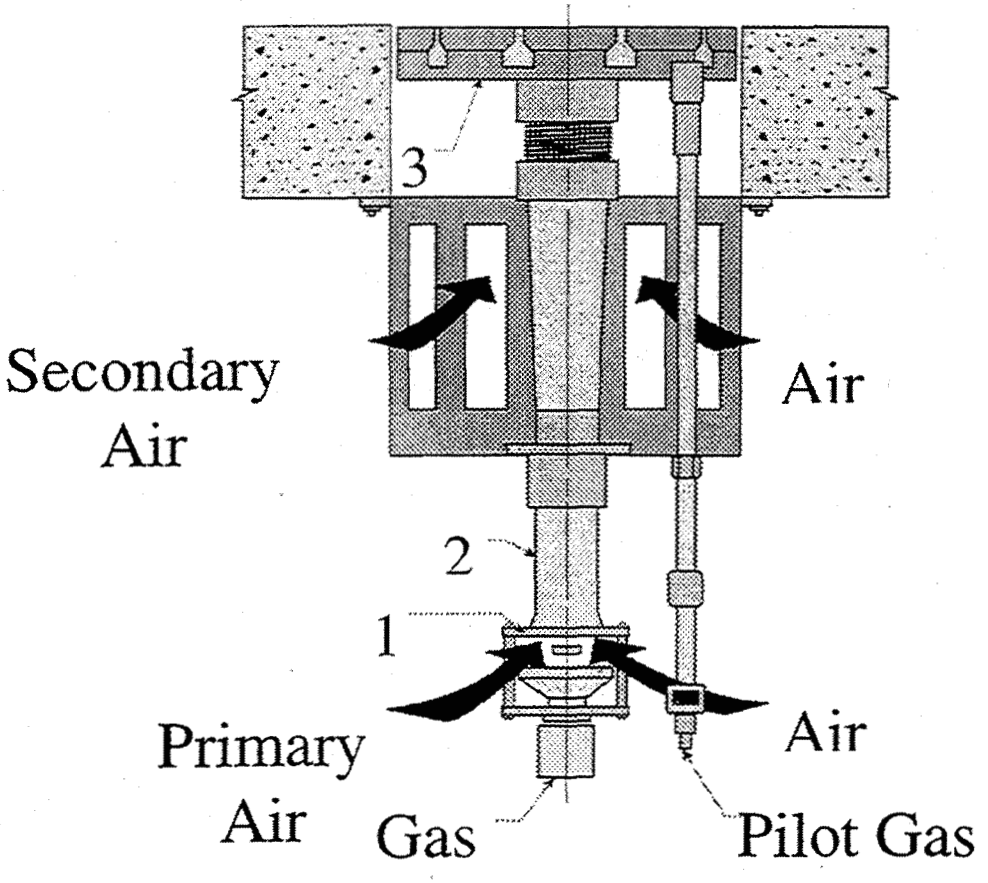


Figure 1 illustrates a typical premix burner assembly. Item 1 is the inspirator or mixer body. Item 2 is the fuel gas discharge orifice, and Item 3 is the burner tip or nozzle. Fuel gas is discharged across the orifice at a relatively high pressure. The low pressure zone created by the fuel gas velocity inspirates a portion of the combustion air called primary air. The primary air rate may range from 30 to 100% of the total combustion air depending upon the burner design and the firing rate. The fuel gas air mixture travels through the mixer body and is discharged through a series of ports in the burner nozzle. The ports in the burner nozzle must be designed to maintain flame stability, give the desired flame shape, and ensure that the fuel gas/air mixture exit velocity is always above the flame propagation speed. The stability of a fuel gas burner depends on having a point within the desired combustion zone with a fuel gas mixture that does not exceed the flame propagation speed. With a premix burner of this type this is accomplished by drilling ports such that the gas/air mixture streams impinge and give a low velocity turbulent zone to create flame stability.

Premix burners normally have both primary and secondary air control. Primary air control is accomplished by a primary air door at the entrance to the inspirator while secondary air control is accomplished by a secondary air register assembly located in the vicinity of the, discharge nozzle. The normal mode of operation is with the primary air door full open and the secondary air register adjusted to obtain the desired excess air. Other primary air door settings may be necessary to prevent flashback or flame liftoff, both of which will be discussed in more detail.

Premix burners have advantages and disadvantages when compared to natural draft raw gas burners.

Some advantages of premix burners include smaller flame volume, a large fuel gas orifice, change in airflow with varying fuel rates, and reduced top plugging when unsaturated or condensed hydrocarbons are present in the fuel gas.

Some advantages of premix burners include smaller flame volume, a large fuel gas orifice, change in airflow with varying fuel rates, and reduced top plugging when unsaturated or condensed hydrocarbons are present in the fuel gas.

One item that determines flame volume is the rate and degree of fuel mixing with the air. Because of the premixing of the fuel gas and a portion of the combustion air, a compact relatively high intensity flame is achieved. For a given burner, the portion of the combustion air that is inspirated depends to a great extent on the fuel gas pressure. Normally, a nominal pressure of approximately 15 psig is used. This pressure ensures that the maximum (sonic) velocity is reached, yet normally allows gas orifices to be used that are large enough to limit fouling or plugging. Because of the above, premix burners are normally considered the best burner for low combustion air pressure drops, since for this case one of the sources of mixing energy is relative low.

Premix burners provide some fuel air ratio control. The degree of fuel air ratio control is dependent, in part, upon the amount of air that is inspirated. For example, if the burner inspirates

5% of the combustion air, then that portion of the combustion air varies with fuel flow. If the burner inspirates 100% of the combustion air the total airflow varies with fuel flow.

A third advantage of the premix burner is the large fuel gas orifice. Flame shaping in low air pressure drop burners is normally achieved by the fuel nozzle (gas tip) drilling. Multiple ports are used in the fuel nozzle to achieve the appropriate flame pattern. The design of a raw gas or nozzle mix burner includes several small orifices that discharge high pressure gas into the combustion zone to achieve the desired flame pattern. Thus plugging problems are more common with the raw gas burner than with the premix burner.

The fuel/air mixture of the premix burner has a greater cooling effect on the nozzle than the fuel gas has on the nozzle of a raw gas -burner. If condensate is present in the fuel gas, tip fouling that normally occurs when the condensate contacts a hot surface is reduced because of a cool premix nozzle.

Disadvantages of the premix burner include limited turndown and the resulting flashback and the limited adaptability for use with multiple fuels. It is not normally used in forced draft applications.

The range of input rates within which the burner will operate is specified by the burner turndown ratio. A form of instability known as flame blow-off limits the maximum input rate (which results from the mixture velocity exceeding the flame velocity). Normally the selection of a premix burner is based on the maximum input rate. Once the burner has been selected the input rate is set. The minimum input rate is limited by the phenomenon known as flashback (which results from the flame velocity exceeding the mixture velocity). When the flame velocity exceeds the mixture velocity the flame burns back through the tip and combustion occurs at the fuel gas orifice. The minimum turndown for the premix burner is dependent upon the fuel gas pressure available and the type of fuel burned. The burner turndown is limited for fuels such as hydrogen that have high flame burning speeds.

The premix burner has a limited capability for utilization of multiple fuels with a wide range of heating values. The ability to inspirate air depends on the energy of the fuel gas stream. The different pressure requirements for various fuels greatly affects the ability of the premix burner to inspirate primary air.

Pilot Burners

The pilot burner is a safety device. It provides a stable source of ignition for the main burners. The pilot burners should be operated at a constant fuel gas pressure. A separate fuel gas regulator is required for the pilot burners. The pilot gas regulator should be set at 3-5 psig for fuels with low hydrogen content (under 30 volume %). For high hydrogen fuels (over 30 volume%), the fuel gas regulator is set at 10-15 psig to prevent flashback which results from the flame velocity exceeds the mixture velocity. When the flame velocity exceeds the mixture velocity the flame, combustion occurs at the mixer orifice.

The pilot burner is a premix burner designed to inspirate 100% of the combustion air. This enables the pilot to stay lit even if the combustion air blower fails. The pilot burner tip is designed with six ignition ports. The air mixer should be opened to the point where six spikes of flame are seen from these ignition ports. When this happens, there will be flame inside the tip from the ignition ports up and out the tip. The pilot tip may glow red. It is made out of high alloy. It will not burn up. It is important to have flame inside the tip because the tip acts as a shield to prevent gusts of wind or high flows of air from blowing the pilot flame out. The hot pilot tip also ensures reignition if the fuel supply is momentarily interrupted.

Nozzle Mix Burners

A second type of natural draft gas burner to be discussed is the nozzle Mix gas burner. As the name implies, gas and combustion air do not mix until they leave the nozzle discharge ports. The two fluids are kept separate in the burner itself, but the nozzle fuel orifices are designed to provide mixing with the air as the air discharges through the burner tile. Figure 2 illustrates a natural draft nozzle mix burner that is typically used in fired heaters. The important components are Item 1, the burner tile, Item 2, the burner tip, Item 3, the burner cone, and Item 4, the burner air register assembly.

The burner tile serves as a limiting orifice for the combustion air. The shape of the burner tile helps control the shape of the flame. It also radiates heat to the gas-air mixture to help maintain a stable flame. The size of the tile is determined by the air pressure drop or draft available. In sizing a burner, every attempt should be made to utilize all the air pressure drop that will be available. A natural draft burner of this type typically uses 0.2 to 0.3 in w.c. pressure drop across the burner tile.

Item 2, the gas tip, serves as a limiting orifice for the fuel gas. A normal design pressure of approximately 15 psig is used. Again, this pressure permits the gas to achieve maximum velocity yet prevents the port size from becoming extremely small. Flame shape is achieved by the gas tip drilling. Normally, the tip has one or two port circles with ten to twenty fuel gas discharge ports. The flame shape is determined by the included angle of discharge for the ports. For example, a 70° included angle gives a relatively long, narrow flame, while a 1000 included angle gives a relatively short wide flame pattern.

Item 3, the cone, serves as a flame holder for the burner. Normally, several small ports with an included angle of 1800 are drilled in the gas tip. These ports discharge fuel gas onto the face of the cone. The impingement of the fuel gas is in a low velocity zone on the face of the cone. The fuel air mixture begins to burn on the face of the cone and stabilizes the flame.

The air register assembly, Item 4, is used to control the total amount of air to the burner. Correct operation of the register assembly of a natural draft burner is imperative to achieve proper operation of the heater. The air register assembly is used to control excess air while the stack damper is used to control the draft. Normal accepted practice is to set the draft just below the convection section at approximately 0.1 in. w.c. negative pressure. The burner register is then

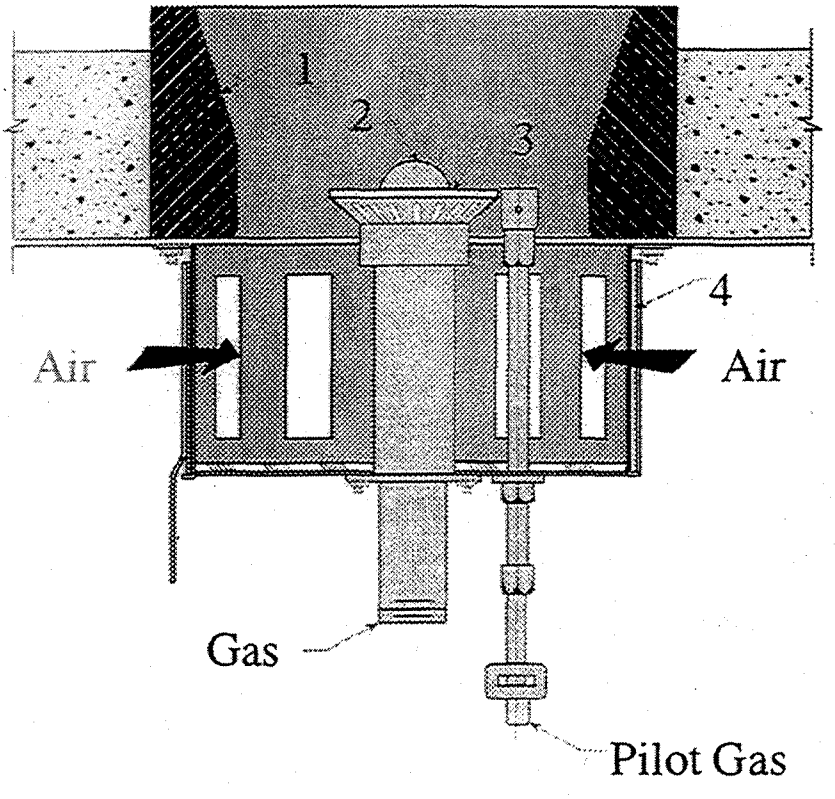
adjusted to control the excess air. Adjustment of the register assembly changes the combustion product flow rate through the heater and therefore changes the draft. With adjustment of the air register assembly, additional adjustment of the stack damper is required.

Nozzle mix or raw gas burners when compared to premix burners have several advantages. These advantages include a high turndown ratio, no possibility of flashback, and the ability to burn a wide variety of fuels.

The disadvantages of natural draft raw gas burners include tip fouling because of the small ports, and somewhat less efficient mixing with air than normally achieved with a premix gas burner.

Figure 2

Raw Gas or Nozzle Mix Burner



Cleaning Gas Burners

Gas burners will need cleaning during unit operations. When corrosion at the head of the burner does occur, the burner can be removed for cleaning or replacement.

1. Close gas firing valve at burner.

2. Disconnect gas line from gas manifold.

3. Remove the manifold bolts and drop out the burner.

   CAUTION: Use protective mittens, as burner will be HOT.

4. Clean burner head. Be careful not to enlarge the hole opening. Use no hard steel cleaning tools, as these may damage the burner.

5. Replace burner and hook up gas line to burner.

6. Put burner back in service by the burner lighting procedure.

Pilot Burners

UOP specifies continuous pilots for safety concerns. If the fuel valve were to dose, extinguish the flame and then suddenly reopen, the pilot would relight the flame and prevent an explosive mixture.

Pilots have a 1/16" fuel orifice and are prone to plugging. Strainers are a must. Natural gas is the best fuel.

UOP specifies two pilots, the J-3/4 and the ST-1S. Most new job should have the higher stability ST-1S. The J-3/4 operates at 3-5 psig gas pressure. The ST-1S operates at 10-15 psig.

Open the primary air register wide open. The ST-1S will have 4 horizontal flame spikes shooting from the side. The pilot head will glow red. This is normal. It will not hurt the head which is cast 310 SS. The glowing head acts as a glow plug in case pilot fuel is interrupted.

Hydrogen has a flame speed of 7 feet/sec versus 1 feet/sec for most hydrocarbons. Flashback or backfiring can occur when the pilot fuel has high hydrogen content (> 30%). Flashback occurs when the flame speed exceeds the mixture velocity and the flame bums back in the mixer. If backfiring occurs, raise the fuel pressure to 18-20 psig (ST-1S) and pinch down on the primary register.

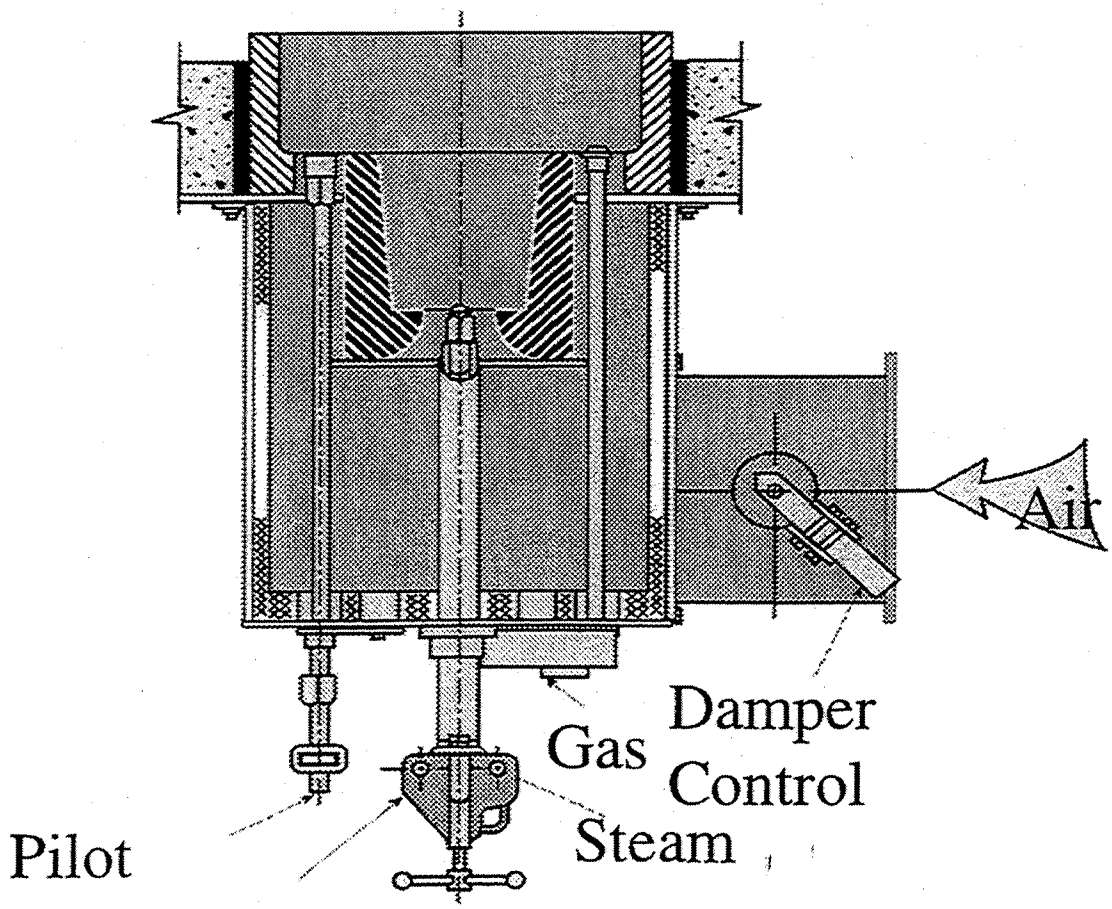
Some refiners have drilled the pilot fuel orifice to 0.10" to prevent plugging. Some stability loss can occur (blowoff). This should not be done without Zink's approval.

Combination Oil and Gas Burners

Typically in refinery operations most oil burners have the capability of firing gaseous fuels. Figure 3 illustrates a typical natural draft combination burner.

Figure 3

John Zink PDA
Combination Oil & Gas Burner



The burner shown by Figure 3 is a typical double block combination gas and oil burner that has been used in a natural draft application in the refinery industry for years. Item 1 is the secondary burner block, and Item 2 is the primary burner block or oil tile. As is the case for the raw gas burner, one of the purposes of the secondary block is to meter the combustion airflow. The other purpose of the secondary block is flame stabilization for fuel gas firing. This type of burner normally has four to six gas tips. The gas tips have an ignition port that impinges gas on a 1 edge that is part of the secondary tile. The fuel gas velocity is destroyed and the same time this portion of the fuel is mixed with combustion air. Ignition and burning then takes place on the tile ledge and this serves as the stabilizing ignition source for the burner. Item 3 is the gas tip. The

gas firing ports are so arranged on the tips to give the-desired flame pattern. Again, the drilling of the tips controls the length and diameter of the flame.

The oil tip is positioned in the primary tile in such a manner as to provide stability for fuel oil firing. The primary tile or block creates a low pressure zone in the vicinity of the oil tip which forces recirculation of oil mist into the hot combustion zone created by the primary tile. This provides a stabilizing effect for the burner and in addition, aids in vaporization of the fuel oil.

The position of the oil tip is critical. If the oil tip is raised too high in the primary tile, the recirculation effect is lost and the flame stability suffers. If the oil tip is too low in the primary tile, impingement of raw oil on the tile occurs and coking and an oil spill may, result.

The air required for primary combustion of the oil is admitted to the burner by Items 4 and 5, which make up the primary air register assembly.

As is the case for gas firing, the flame shape is determined by the oil tip drilling. With everything else being equal, a narrow angle gives a long narrow flame, while a wide angle gives a short wide flame.

Do not fire a burner simultaneously on gas and oil because it will starve the oil fire for air, and a puffing of the heater typical of air starvation may result. The only exception is lighting the oil burner with the gas burner. Once the oil flame is stable, turn the gas flame off.

When firing oil and gas, one fuel firing rate is held constant while the other fuel firing rate is varied to maintain process outlet temperature. The fuel connected to the temperature controller should be the fuel supplying the majority of the heat release. For example, if a heater has ten burners and eight are firing on gas and two on oil, the temperature controller should be connected to the gas supply. This allows the heater to better absorb fuel composition and process changes. Connecting the temperature controller to the fuel supplying the least heat release may result in burner flameout during upset conditions. If the temperature controller is connected to the two oil burners and the heat release load, increases quickly, the two oil burners will run out of air. The uncombusted oil will meet with excess air from the gas burners and combustion will result. This combustion may be in the convection section and equipment damage will occur.

Oil Burner Operation

How do oil burners operate?

Oil burning, generally, is more difficult than gas burning. Oil, as a liquid must be converted to oil as a gas (vapor) before it can mix with air and begin to burn. In typical operation, oil is delivered to the burner where it passes through an atomizer which converts the liquid to millions of tiny droplets which, hopefully, are not greater than 10-50 in size. Such conversion tremendously increases the ratio of surface-to-mass, permitting substantially instantaneous beat absorption the droplets, altering their physical state from liquid to gas in microseconds. Gas thus produced mixes freely with air and burning begins. For burning to occur, there must be an ample supply of heat relative to the atomizer discharge. Atomization may be through the action of steam, air, or high-pressure gas. Although there are several methods of atomizing the oil, steam is used in the refinery because it is readily available.

Steam-atomizing burners may be either the outside mixing type or the inside mixing type such as we use. The inside mixing type has the advantage of lower atomizing steam consumption. The steam and oil form a foam in the internal mixing chamber. This foam is directed by the design of the burner tip so that the flame is of proper size and pattern for the firebox.

Viscous fuels should be heated enough to become sufficiently fluid. very-high-viscosity fuels, the higher the temperature the better the performance. Temperatures should not be used which are above the flash point of the oil, or the flame will sputter. High fuel temperature along with a highly superheated atomizing steam may cause excessive coking at the burner top.

The recommended maximum atomization viscosity is 43 cS (200 SUS). Too a viscosity results in poor atomization with too large drops, and, too small a contact area for combustion air. This will effect a poor blending of oil and air, as well as unsatisfactory combustion. Moreover, oil drops are likely to spout against the burner elements. Too low a viscosity results in very fine atomization with rapid combustion so that more air is needed at the burner mouth and less on the periphery; since the burner fronts have not been designed for this function, this will lead to a greater excess air volume. An extreme case will be that the oil will be sprayed from the burner tip as a gas, and the flame will repeatedly fail and ignite again on the burner tile.

Cleaning Oil Burners

Oil burner tips will accumulate coke and other foreign particles, which will eventually plug the oil tip. The tips, therefore, should be cleaned occasionally to keep them operating at maximum efficiency.

1. Close oil firing valve and oil block valve to oil gun.

2. Open the steam purge valve and steam through the oil line and oil tube for five (5) minutes to clear the oil gun of oil.

3. Close steam purge, steam firing and steam block valves.

4. Disconnect oil and steam lines at oil gun.

5. Loosen clevis (receiver bolt), and remove oil gun from receiver.

   CAUTION: Use heat protective mittens, as oil gun will be HOT.

6. Remove oil tip and remove all foreign material from the atomizing chamber and orifice holes.

7. Blow the steam and oil tube with air, then replace oil tip.

8. Replace the copper gaskets in oil body, on oil and steam inlets.

9. Place oil gun back in oil gun receiver and tighten clevis.

10. Hook up steam and oil lines, then open steam firing, steam purge and steam block valves to check for leaks.

11. Put burner back in service by following the burner, lighting procedure.

Training Services

Heater Design Procedures

FH 2004/8-1

Section Outline

- Process
- Radiant Layout
- Firebox Geometry
- Burner to Tube Clearance
- Convection Layout
- Sulfur Dewpoint
- Coil Pressure Drop
- Draft
- Design Summary

FH 2004/8-2

Fired Heater Specification

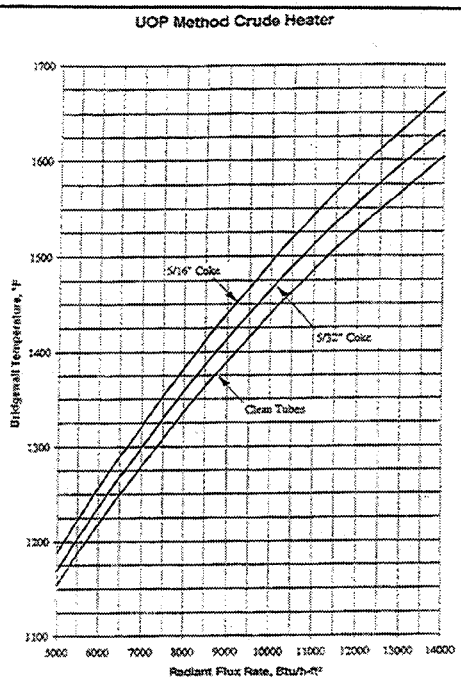
Process	Radiant Flux, Btu/h-ft2	Mass Velocity, lb/sec-ft2	Process ΔP,psi	Coil Metallurgy	Type of Heater
Crude	10,000	200 min	150	5 Cr or 9 Cr	Vertical Tube Box Cabin or V.C.
Vacuum	8,000	250 min	75	5 Cr or 9 Cr	Vertical Tube Box Cabin or V.C.
Visbreaking	7,000-8,000	350 min (5-6 fps cold oil)	250-300 clean	9 Cr	Cabin with brick centerwall
Coking	9,000	400 min (8 fps cold oil)	300 clean	9 Cr	Cabin with brick centerwall or double fired horizontal tube
Semi-Regn. Platformer	10,000	40-50	7	2-1/4 Cr	V.C., cabin, wicket
CCR Platformer	10,000 SFU 15,000 DFU	23-35	3 per cell	9 Cr	Single Fired U-tube Double Fired U-tube
Reboiler	10,000	150-300	50	C.S.	V.C.
Hot Oil	10,000	400-600	65	C.S.	V.C.
Naphtha Hydrotreater (max. 15 mils/yr)	10,000	70-180	50	9 Cr	V.C.
Naphtha Hydrotreater (with cracked stock)	10,000	70-180	50	347	V.C.
Kerosene Hydrotreater	10,000	150-250	50	347H SS	V.C.
Diesel Hydrotreater	10,000	150-250	50	347H SS	V.C. or double fired vertical tube
Hydrocracker Gas Only	10,000 - 13,000	50-80	30-50	347H SS	double fired vertical tube
Hydrocracker Gas&Liquid	15,000	180-300	50-80	347H SS	double fired vertical tube

Example Design Problem

- **Design a new vertical cylindrical crude heater**
- **Process Conditions**
  - **50,000 BPSD=632,624 lb/h**
  - **Process Inlet 500°F(260°C)**
  - **Process Outlet 7200°F(382°C)**
  - **Process Duty =127.7 MM Btu/h**
- **Low NOx –natural draft burners are required**

Select Radiant Flux Rate

- Design new units for conservative flux rates
- Consider future process cabability
- Example, new crude heater flux=10,000
- BWT=1440°F



FH 2004/8-5

Run Fuels Program

- Stack=Process Inlet +75°F
- BWT from UOP Charts
- Example
  - Crude heater BWT=1440°F
  - Stack=500+75=575°F
- Fuels program output
  - efficiency=84.5%
  - fuel fired=127.7/.845= 151.1 MM Btu/h
  - radiant convection split (59.6%/84.5%)=70.5% radiant
  - amount of flue gas

FH 2004/8-6

Run Fuels Program

FUEL GAS DATA

adiabatic flame temp, °F	3331		FLUE GAS INFO	Btu/lb
LHV of fuel, BTU/lb	23035	15%xs air	lbwt	enthalpy
LHV, Btu/SCF	719.29		1400	389.56
lbs of wet air/lb fuel	20.92		1500	421.41
(Flowing @ Amb Air) Ft3 of wet air/lb fuel	280.72	BW temp, °F=	1440	402
air preheat temp, °F	60	% Radiant Efficiency=	59.61%	
air preheat content, btu/lb	0.00	FG out stack, °F=	575	141.41
Corrected heat content fuel, btu/lb	23035	Heat Loss=	2.00%	
Corrected heat content fuel, Btu/SCF	719.3			
Note: °P air side burner – (Mass Flow)^2 * Specific Volume (ft3/lb)		% Total Efficiency=	84.50%	

FH 2004/8-7

Layout Radiant Section

- **Set number of passes to obtain desired mass velocity**
  - **Example, crude=6"Sch 40 tubes , minimum mass velocity=200 lb/sec-ft2**
- **Even number of tubes per pass**
- **Set Tube Length to obtain L/D=2**
- **Estimate number of burners and heat release**
  - **Flame length must be 1/3 to ½ radiant height**
  - **Gas flames -1 ft/mm btu/h**
  - **Gas Low NOx flames -1.5 ft/mm btu/h**
  - **Gas Ultra-Low NOx flames -2-3 ft/mm btu/h**
  - **Oil flames -2 ft/mm btu/h**
  - **Oil Low NOx flames -1.5 ft/mm btu/h**
- **Check burner to tube clearance**

FH 2004/8-8

Layout Radiant Section

- Number of passes=(632624 lb/h)/(3600sec/h)*200MV*0.2006Ft2/pass
  = **4.4 (Use 4 passes)**

- 127.7 mm Btu/h* (.705 radiant)/10,000= **9003 ft2 radiant surface**
- 9003/1.734ft2/ft=**5191 feet of 6" pipe required**
- **Try 88 tubes**
  - Length=5191/88=59'-1.6'return bend =**57.4'**
  - Tube Circle Diameter=88/3.147=28'
  - **L/D=57.4/28=2.05**

- **Try 12 Burners**
  - fuel fired= (151.1 MM Btu/h)/12=**12.6 MM Btu/burner**
  - Flame length= 2 feet*12.6= **25.2 feet**
  - % tube coverage=25.2/57.4= **44%**

FH 2004/8-9

Burner to Tube Clearance

- **Most heater failures are cause by flame impingement**
- **API clearances are not adequate**
- **Add an additional 12" for coking or fouling services**

Gas Firing

Maximum Heat Release per Burner, MM Btu/h	UOP Spacing horizontal to centerline wall tube from burner centerline, inches	API Spacing horizontal to centerline wall tube from burner centerline, inches
4	39	30
6	39	36
8	45	42
10	48	48
12	54	54
14	60	60

Oil Firing

Maximum Heat Release per Burner, MM Btu/h	UOP Spacing horizontal to centerline wall tube from burner centerline, inches	API Spacing horizontal to centerline wall tube from burner centerline, inches
4	48	33
6	48	39
8	54	45
10	57	51

Oil Firing- Staged Air Burner

Maximum Heat Release per Burner, MM Btu/h	UOP Spacing horizontal to centerline wall tube from burner centerline, inches	API Spacing horizontal to centerline wall tube from burner centerline, inches
4	54	33
6	54	39
8	60	45
10	63	51

FH 2004/8-10

Layout Radiant Section

- **Try 12 Burners at 72" burner to tube spacing**
  - Burner circle diameter= 28'-2(6')=16'
  - Space between burner= {16*12*3.147- 12(30")}/12=20"
  - Burner to burner= **50"= 1.66 tile OD**

  - **Low Nox burners require 1.5 tile OD**
  - **Ultra low NOx require 2.0 tile OD**

Layout Convection Section

- **Set Tube Length to cover radiant section(TCD)=28'**
- **Set number of tubes per row to obtain flue gas mass velocity =0.3-0.4 lb/sec/ft2**
- **Set number of tube rows to obtain 75°F cold end approach**
- **Use fins for extended surface**
  - **Use Maximum fin length=1"**
  - **Use Maximum fin density= 5fins/inch**

  Check convection flue gas pressure drop, target=0.2-0.3" H2O

Select Convection

- If 0.3MV=11 rows
- If 0.4MV=12 rows
- 12 tubes/row=0.25 MV
- 8 tubes/row=0.38 MV

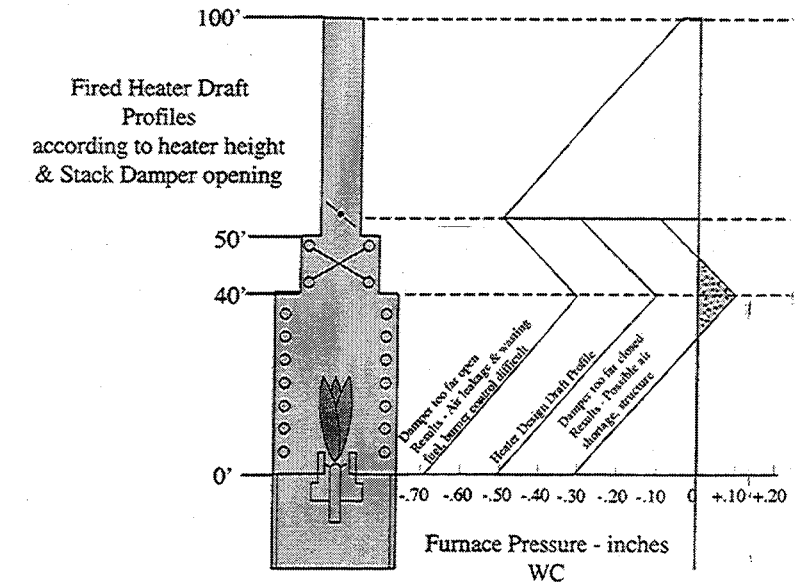
- Use 8 tubes per row at 12 tube rows to obtain a 71°F cold end approach with a 0.284" ΔP

Convection Section Layout
Gas Fired, 450°F Process Inlet
5 fins/inch, 0.75"high(4")\1.0"high(6"),0.05" thick

Number Tube Rows	4" Tube Cold End Approach, °F	Flue Gas ΔP, Inch H_2O	6" Tube Cold End Approach, °F	Flue Gas ΔP, Inch H_2O
		0.2 lb/sec/ft² Mass Velocity		
6	219	0.052	214	0.032
7	149	0.040	147	0.041
8	102	0.046	100	0.047
9	69	0.053	68	0.055
10	44	0.059	47	0.061
11	29	0.065	32	0.068
12	18	0.071	21	0.075
		0.3 lb/sec/ft² Mass Velocity		
6	298	0.072	298	0.073
7	218	0.089	220	0.092
8	160	0.104	163	0.108
9	117	0.119	121	0.123
10	83	0.132	89	0.139
11	60	0.147	66	0.154
12	43	0.159	48	0.167
		0.4 lb/sec/ft² Mass Velocity		
6	357	0.127	363	0.130
7	274	0.157	281	0.163
8	210	0.183	218	0.192
9	161	0.211	169	0.221
10	120	0.235	131	0.248
11	93	0.260	101	0.275
12	71	0.284	79	0.300

FH 2004/8-13

Draft Profile



FH 2004/8-14

Layout Stack Diameter

- **Calculate flue gas**
  - fuel fired= (151.1 MM Btu/h)/23035btu/lb=**6560 lb/hr**
  - Flue gas = 6560*(20.92+1) lb air/lb fuel= **143,792 lb/hr**

Calculate stack diameter(Size for 25 fps)

- Flue gas flow= 143792/(.0364lb/ft3)*3600=1097ft3/sec
- Area = 1097/25= **43.9ft2**
- Diameter= $(43.9ft2/.7854)^{0.5}$= **7.5'**

FH 2004/8-15

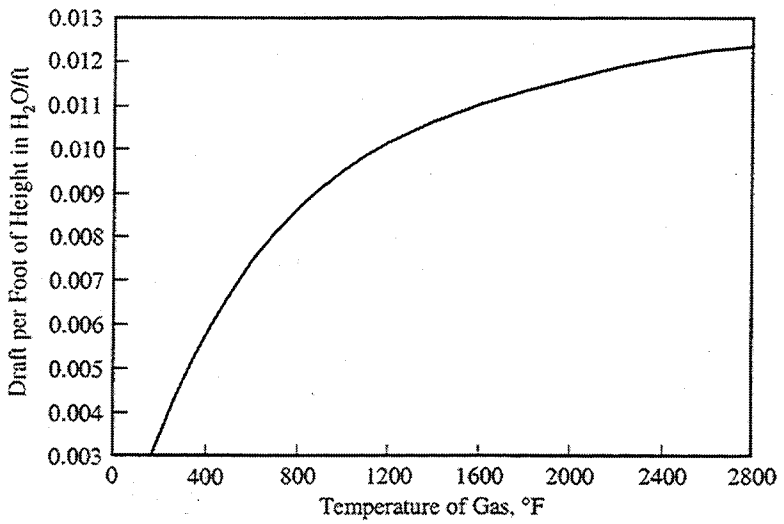
Run Fuels Program

FUEL GAS DATA

adiabatic flame temp, °F	3331		FLUE GAS INFO		Btu/lb
LHV of fuel, BTU/lb	23035	15%xs air		bwt	enthalpy
LHV, Btu/SCF	719.29			1400	389.56
lbs of wet air/lb fuel	20.92			1500	421.41
(Flowing @ Amb Air) Ft3 of wet air/lb fuel	280.72	BW temp, °F=		1440	402
air preheat temp, °F	60	% Radiant Efficiency=		59.61%	
air preheat content, btu/lb	0.00	FG out stack, °F=		575	141.41
Corrected heat content fuel, btu/lb	23035	Heat Loss=		2.00%	
Corrected heat content fuel, Btu/SCF	719.3				
Note: ~P air side burner ~ (Mass Flow)^2 * Specific Volume (ft3/lb)		% Total Efficiency=		84.50%	

FH 2004/8-16

Draft



FH 2004/8-17

Layout Stack Height

- **Calculate pressure drop in velocity heads**

- **Velocity Head (inch H20)=(0.003)(MV)^2(SV)**
  - Flue gas mass velocity= lb/sec-ft2
  - Flue gas specific volume= ft3/lb

Velocity Head Loss

- Bare Tubes =0.2VH (per row)
- Finned Tube =1.0VH (per row)
- Damper =1.5VH (per row)
- Stack Entrance =0.5VH
- Stack Exit =1.0VH

FH 2004/8-18

Layout Stack Length

- **Calculate pressure drop=0.574" H2O**
  - Bridgewall= 0.1" H20
  - Convection= 0.284" H20
  - Damper= 0.05" H20
  - Entrance and Exit (1.5VH)=0.10" H20
  - Stack Friction= 0.04" H20

Calculate stack length(Size for 0.0065" H2O/ft)

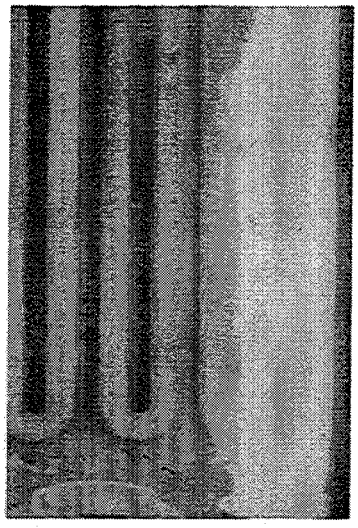
- Height = 0.574/0.0065= **88 ft**

Layout Comparison

	Hand Calc.	Computer
Radiant Flux Rate, btu/h-ft2	10,000	9,963
Radiant Surface,ft2	9,003	8,936
BWT,°F	1440	1464
Stack, °F	575	571
Fuel Efficiency, %	84.5	84.7
Fuel Fired, MM Btu/h	151.1	150.7
Stack Height, ft	88	94

Flame Emissivity

- Oil Firing on Right Burner
- High Emissivity Oil Flame
- Gas Firing on Left Burner
- Low Emissivity Gas Flame
- Oil flame requires larger clearance to tube



FH 2004/8-21

Flame Radiation

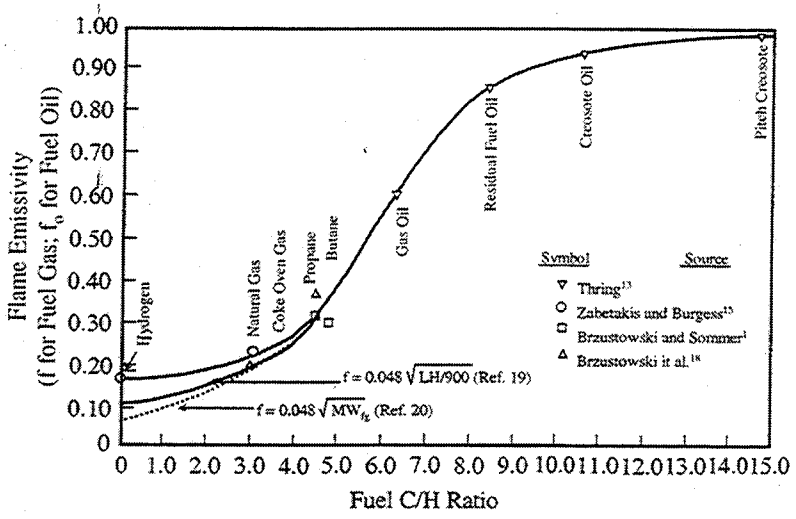
- Varies by square of distance to tube

$$= \frac{(\text{Fuel Emissivity})(\text{Burner Duty})}{(\text{Burner to Tube Distance})^2}$$

- Oil is twice that of gas
- Talmor Method

FH 2004/8-22

Flame Radiation - Talmor Method



FH 2004/8-23

Burner to Tube Clearance

- **Most heater failures are cause by flame impingement**
- **API clearances are not adequate**
- **Add an additional 12" for coking or fouling services**

Gas Firing

Maximum Heat Release per Burner, MM Btu/h	UOP Spacing horizontal to centerline wall tube from burner centerline, inches	API Spacing horizontal to centerline wall tube from burner centerline, inches
4	39	30
6	39	36
8	45	42
10	48	48
12	54	54
14	60	60

Oil Firing

Maximum Heat Release per Burner, MM Btu/h	UOP Spacing horizontal to centerline wall tube from burner centerline, inches	API Spacing horizontal to centerline wall tube from burner centerline, inches
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10	57	51

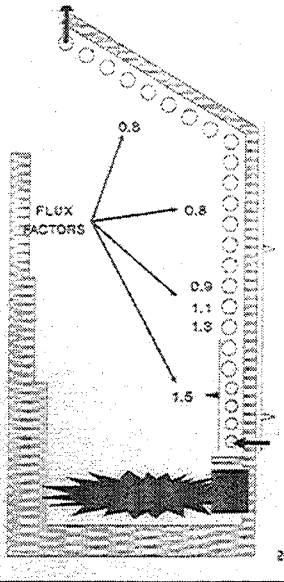
Oil Firing- Staged Air Burner

Maximum Heat Release per Burner, MM Btu/h	UOP Spacing horizontal to centerline wall tube from burner centerline, inches	API Spacing horizontal to centerline wall tube from burner centerline, inches
4	54	33
6	54	39
8	60	45
10	63	51

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Longitudinal Flux Factor

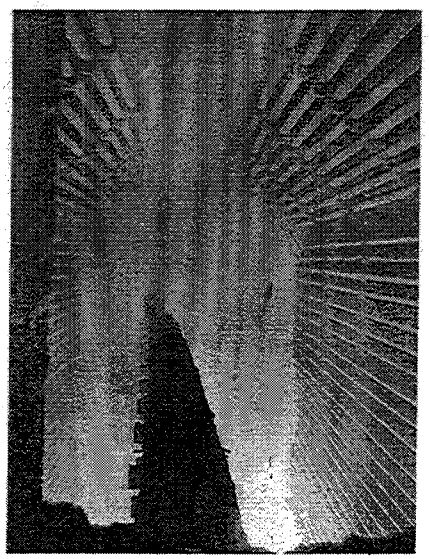
- Popular Design in 1960's
- Side Fired Cabin Heater
- Popular Design in 1960's
- Liquid from burner drips into box
- Overheat lower tube rows
- Locate inlet at bottom of firebox



2004/8-25

Longitudinal Flux Factor

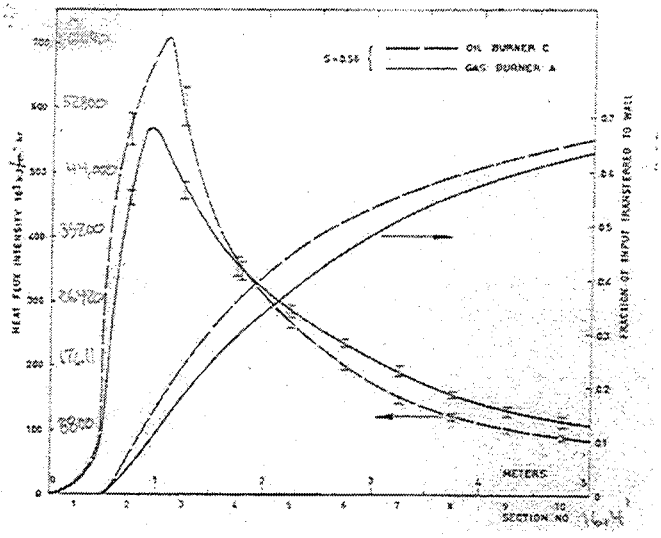
- High heat pickup in lower part of furnace
- Visually much brighter in bottom of furnace



FH 2004/8-26

Longitudinal Flux Factor

- Highest flux rate is 0.25-0.33 of firebox height for floor fired heater



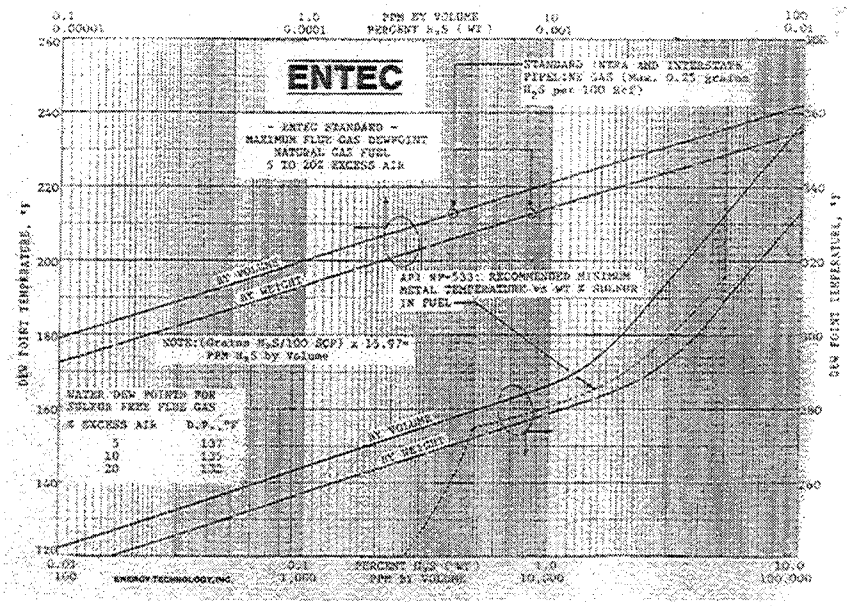
FH 2004/8-27

Flue Gas Sulfur Dewpoint

- Process Inlet-25°F (14 °C) Above Dewpoint
  - Prevents acid condensation on convection tubes
- Stack Inlet-75°F (42 °C) Above Dewpoint
  - Prevents acid condensation in stack

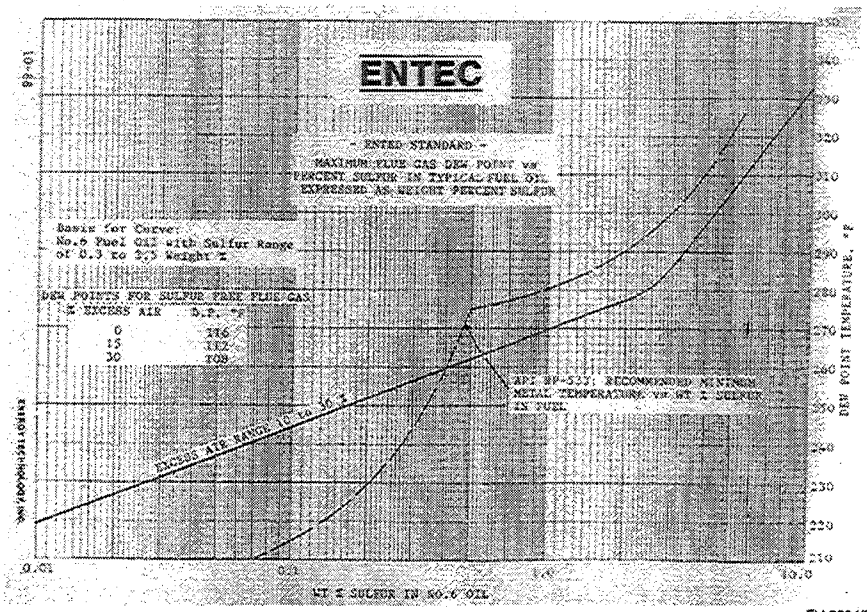
FH 2004/8-28

Sulfur Dewpoint- Fuel Gas



FH 2004/8-29

Sulfur Dewpoint- Fuel Oil

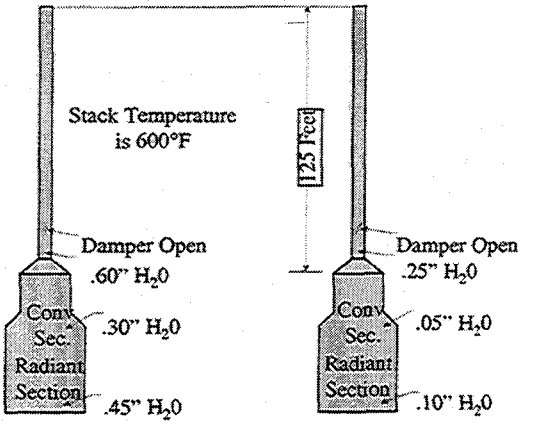


FH 2004/8-30

Draft

- Critical at Bridgewall
  - Avoid pressurizing heater casing
  - Avoid leakage of flue gases to atmosphere
- Critical at Burners
  - Insure sufficient air flow
  - Insure fuel-air mixing not limited
  - Normally acceptable if good value at bridgewall
- Stack Height and Diameter Sized to Obtain Necessary Draft

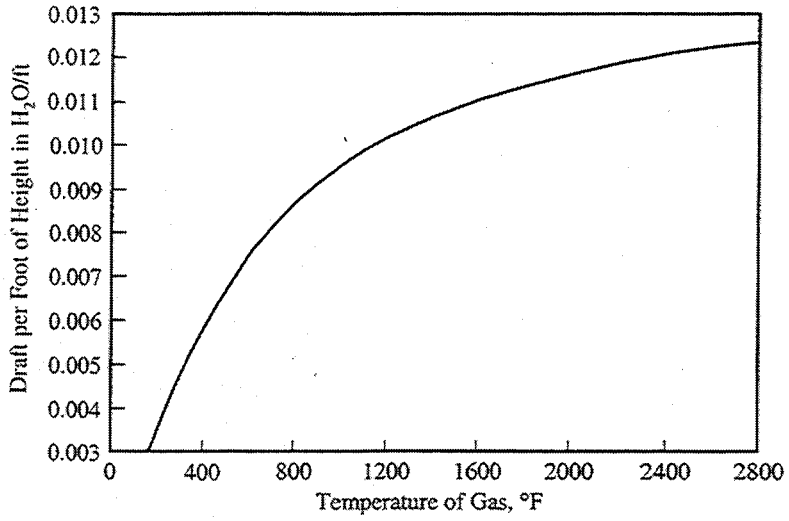
Draft



Damper Too Far Open
Too Much Draft
Result: Waste Fuel

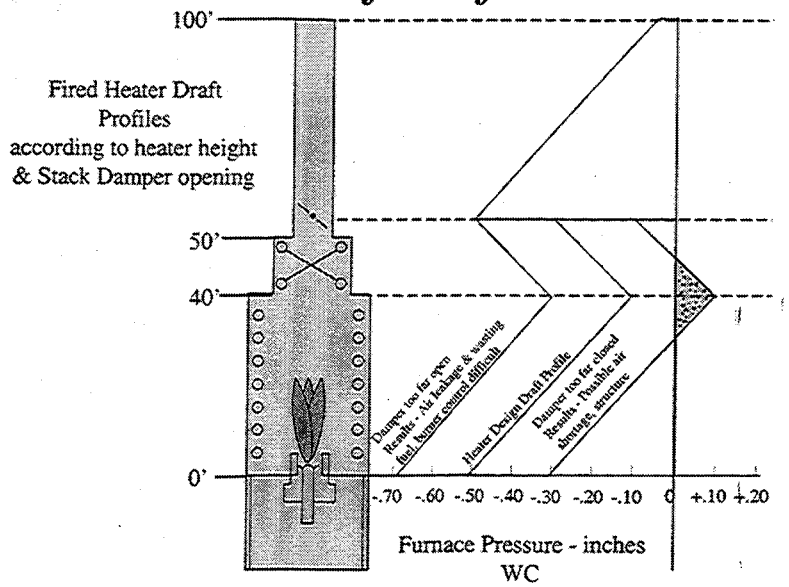
Damper Too Far Closed
Too Little draft. Positive
Pressure Below Convective Section
Result: Damage to Steel Structure

Draft



FH 2004/8-33

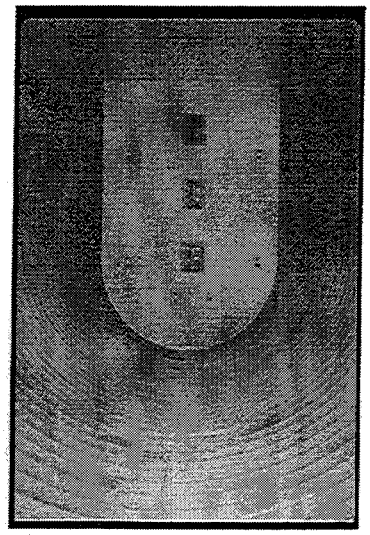
Draft Profile



FH 2004/8-34

Burner Design

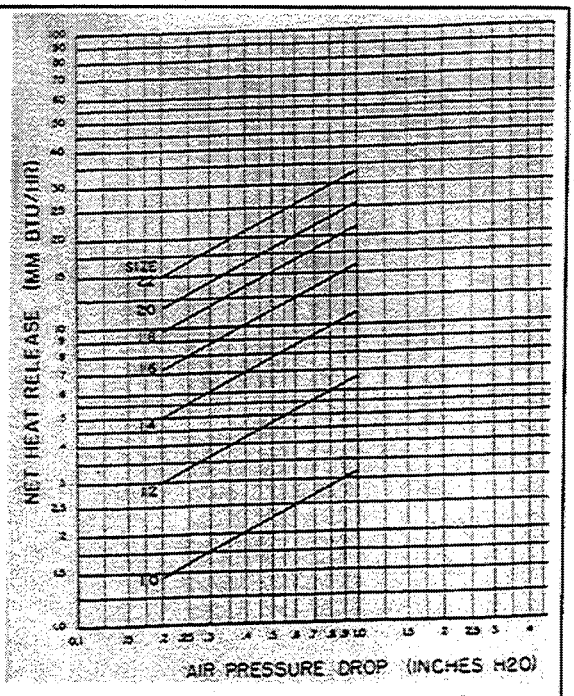
- **Different Draft at each burner**
- **Custom design to use 85% available draft**
- **Same excess air for damper setting**



FH 2004/8-35

Burner Heat Release Curve

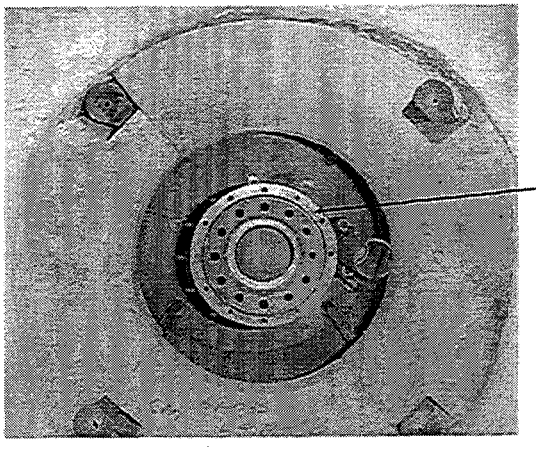
- **Standard Sizes**
- **Custom Design for 85% available draft**
- **Natural Draft burners have limited turndown**



FH 2004/8-36

Burner Design

- **Different Draft at each burner**
- **Custom design to use 85% available draft by welding strip steel to flame holder to obtain same excess air for damper setting**



Longitudinal Flux Factor

API 530 *Calculation of Heater- Tube Thickness in Petroleum Refineries Appendix C* provides a method of calculating TWTs. It list three flux multipliers:

Circumferential Heat Flux Factor
Longitudinal Heat Flux Factor
Tube Metal Factor

The Longitudinal Heat Flux Factor tends and Tube Metal Factor tend to cancel out each other. Most new units are designed using the Circumferential Heat Flux Factor only.

The Longitudinal Heat Flux Factor may be used to troubleshoot existing furnaces. Some major oil companies limit tube length to 50 feet to avoid this potential problem.

Data was plotted correlating L/D ratio versus Longitudinal Heat Flux Factor. High L/D ratios can produce localized overheating.

UOP usually assigns a longitudinal factor of 1.2 for gas firing and 1.3 for oil firing for heaters with an L/D ratio of 1.8-2.0. The peak flux location is 25-33% of the firebox height above the furnace floor

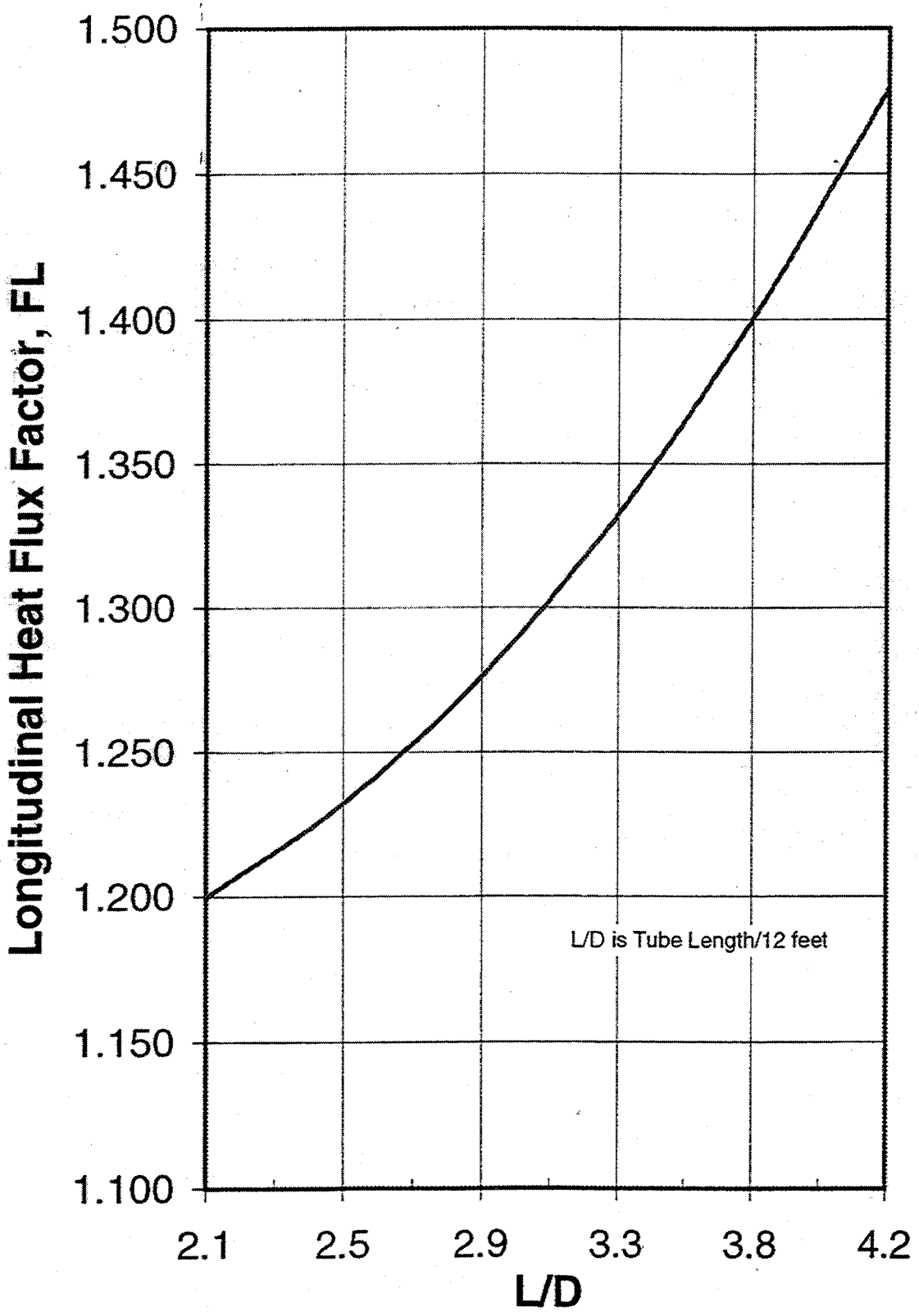
UOP feels that burner to tube spacing and flame lengths are the key variables in determining longitudinal flux maldistribution. For coking and fouling services, UOP increases the burner to tube clearance by at least 6" over the values reported in the *Burner to Tube Clearance Section of API 560*.

The flame length should cover 1/3-2/3 of the tube height. Short flames with long tubes will produce a high longitudinal factor. Longer flames give more even heating, but can result in high BWTs due to short tube length downstream of the flame for hot gas radiation heat transfer to occur.

UOP's design practice uses shorter flames than most designers. The burners are spaced further away from the tubes to compensate for the longitudinal flux maldistribution.

The Talmor method is the best method to determine longitudinal flux maldistribution.

Longitudinal Heat Flux Distribution



Burner to Tube Clearance

Burner to tube clearance is extremely important in fired heater design. The flame radiation is directly proportional to the square of the distance to the tube. Small burner to tube clearances result in flame impingement, hot spots and tube failures.

For example, consider a 5'-0" spacing versus a 3'-0" spacing. The smaller spacing results in $(5/3)^2$ = 2.8 times the flame radiation as the large spacing. The small burner to tube spacing is often a problem in vertical cylindrical heaters.

Few major oil companies follow the burner to tube clearances published in API-560 *Fired Heaters for General Refinery Services*. Most major oil companies heater standard specification have burner to tube clearances similar to UOP's Standard Specification.

In the past, UOP limited natural draft burner heat release to 12 MM Btu/h. Today, UOP limits natural draft gas firing to 14 MM Btu/h and natural draft oil firing to 10 MM Btu/h.

Most burners purchased today are low NO_x burners. There can be problems lighting large heat release natural draft staged gas burners due to the large space between the tips and the problem of filling this large space with fuel. Low NO_x oil burners have very long flames. At high heat releases, the fuel-air mixture runs out of mixing energy, results in uncontroled burning of fuel, poor flame quality, and potential flame impingment.

Burner to Tube Clearance

For critical service heaters (high temperature, high pressure, coking or fouling services), it is desirable to have additional burner to tube clearances (at least 6î additional).

Gas Firing

Maximum Heat Release per Burner, MM Btu/h	UOP Spacing horizontal to centerline wall tube from burner centerline, inches	API Spacing horizontal to centerline wall tube from burner centerline, inches
4	39	30
6	39	36
8	45	42
10	48	48
12	54	54
14	60	60

Oil Firing

Maximum Heat Release per Burner, MM Btu/h	UOP Spacing horizontal to centerline wall tube from burner centerline, inches	API Spacing horizontal to centerline wall tube from burner centerline, inches
4	48	33
6	48	39
8	54	45
10	57	51

Oil Firing- Staged Air Burner

Maximum Heat Release per Burner, MM Btu/h	UOP Spacing horizontal to centerline wall tube from burner centerline, inches	API Spacing horizontal to centerline wall tube from burner centerline, inches
4	54	33
6	54	39
8	60	45
10	63	51

Convection Section Layout

Calculations

The convection section calculations are complex and best handled by a computer program. A detailed set of calculation procedures are listed in Section 14, pp. 309-313.

Number of tubes per row

An even number of tubes per row per pass is selected. For example, a 4 pass heater may have 4,8 12, or 16 tubes per row. Select the number of tubes to give a flue gas mass velocity of 0.3 lb/sec-ft^2.

The convection section width = (No. of tubes + 0.5) x (tube C-C spacing). The width should not exceed 12'-6" to prevent excessive cost for intermediate tubesheet supports.

Tube Length

The tube length is set at the firebox length for cabin or vertical tube box heaters. Use a tube length equal to the radiant tube circle diameter on vertical cylindrical heaters.

Number of tube rows

Three shock tube rows are required due to radiation leakage from the radiant section. The shield rows have an overall heat transfer coefficient in the range of 8-9 Btu/h-ft^2. The flue gas exit temperature from the shock tubes is approximately 200°F lower than the BWT.

The finned rows have an overall heat transfer coefficient in the range of 4-5 Btu/h-ft^2. The number of fin tube rows (above the shock tubes)is set to obtain the desired cold end approach temperature. A gas fired heater will have 7-8 finned rows while an oil fired heater will have 9-10 finned rows.

Pressure drop

A small heater would be designed for 0.10-0.15 inch H_2O flue gas pressure drop. A large heater would be designed for 0.15-0.20 inch H_2O flue gas pressure drop.

Convection Section Layout

Typical Fin Density- Gas Firing

Nominal Tube Size	Fin Length, inches	Fins/inch	Fin Thichness,inches
3	0.50	5	0.05
4	0.75	5	0.05
5	1.0	5	0.05
6	1.0	5	0.05

Typical Fin Density- Oil Firing

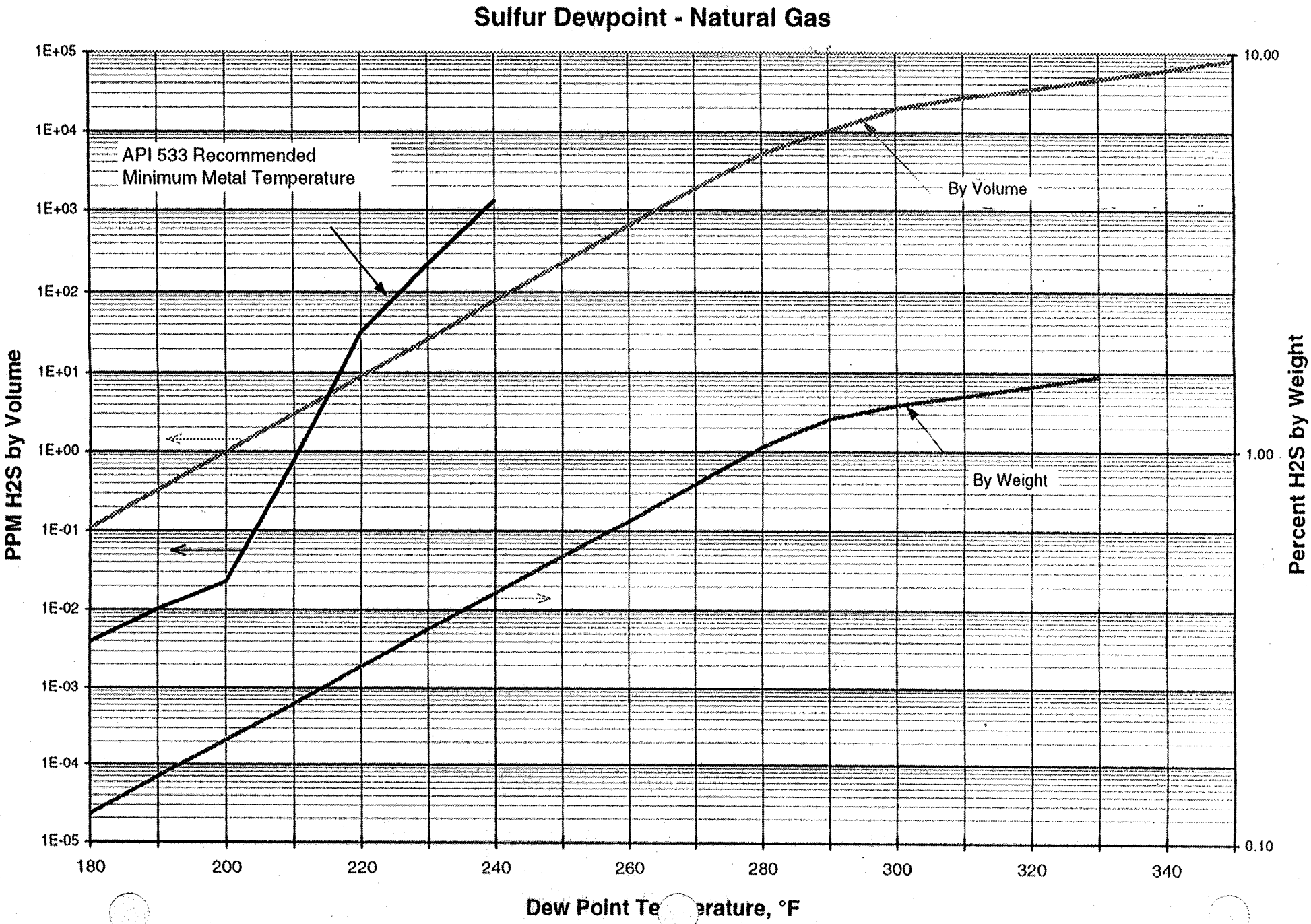
Nominal Tube Size	Fin Length, inches	Fins/inch	Fin Thichness,inches
3	0.5	3	0.10
4	0.75	3	0.10
5	0.75	3	0.10
6	0.75	3	0.10

Acid Dewpoint

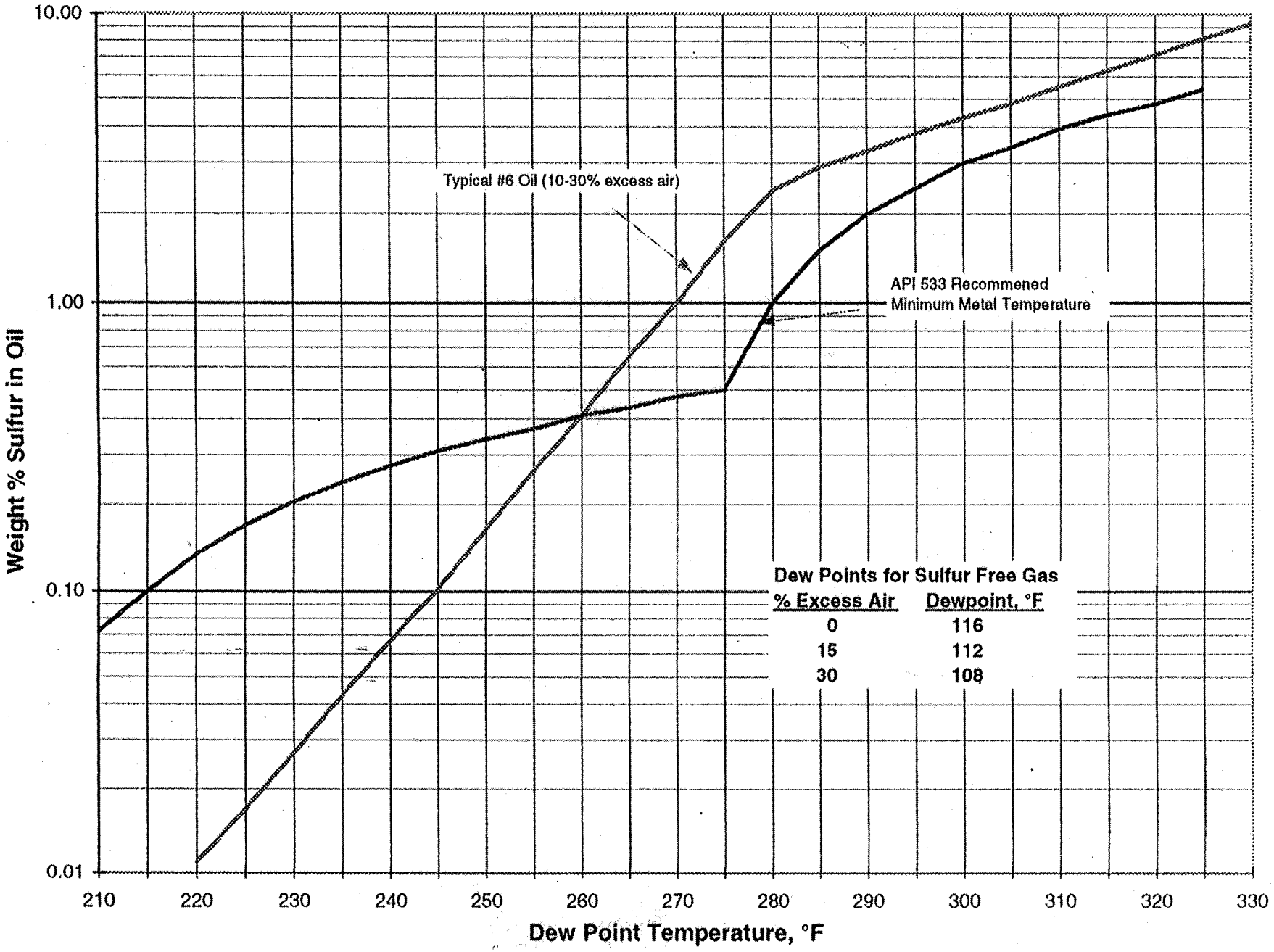
Operating below the acid dewpoint will result in a tube failure due to corrosion attack. The tube metal temperature is very close to the process fluid temperature. UOP recommends operating the process 25°F(14°C) above the acid dewpoint and operating the stack at 75°F(24°C) above the acid dewpoint. The typical stack heat loss is 40°F(22°C). It the margin is too small, acid will condense in the stack.

Use the more conservative dewpoint as calculated by the Entec method or the API method. The Entec data gives a higher dewpoint at low sulfur levels. This data compared very similar to data obtained from By-Cast. The API method gives higher dewpoint at higher sulfur levels.

These charts are used to keep acid from condensing in the air preheater. Condensing acid corrodes the air preheater, fan and duct.


Sulfur Dewpoint - Natural Gas
1E+05
1E+04
1E+03
1E+02
1E+01
1E+00
1E-01
1E-02
1E-03
1E-04
1E-05
PPM H2S by Volume
10.00
1.00
0.10
Percent H2S by Weight
API 533 Recommended
Minimum Metal Temperature
By Volume
By Weight
180
200
220
240
260
280
300
320
340
Dew Point Te erature, °F

Sulfur Dewpoint - #6 Fuel Oil



Pressure Drop through Pipe

The attached charts may be used to size lines or calculate pressure drop through a heater coil.

Line to Heater

The example is to size a line to the heater. The target is 1 psi/100 eq. Ft.

The flow rate is 29603 BPSD = 1104 GPM @ 0.594 specific gravity= 328229 lb/h.

A 8" Sch 40 line has a pressure drop of 1.4 X 0.596= 0.834 psi/100 eq. Ft.
A 6" Sch 40 line has a pressure drop of 6.5 X 0.596= 3.87 psi/100 eq. Ft
Use the 8" line.

Line from Heater

The example is to size a line from the heater. The target is 1 psi/100 eq. Ft.

The total flow rate is 328229 lb/h.
Vapor flow is 164115 lb/h at 0.932 lb/ft^3 vapor density.
Liquid flow is 164115 lb/h at 0.598 specific gravity = 37.32 lb/ft^3 liquid density

Vapor CFS (cubic feet per second) = 164115 lb/h divided by 3600sec/h divided by 0.932 lb/ft^3 vapor density = 48.91 CFS
Liquid CFS (cubic feet per second) = 164115 lb/h divided by 3600sec/h divided by 37.32 lb/ft^3 vapor density = 1.22 CFS

Two phase Specific Volume (SV)= CFS_{vapor} +CFS_{liquid} / Lb/sec
SV= 48.91 +1.22/91.175= 50.13/91.175= 0.550 ft^3/lb

A 12" Sch 40 line has a pressure drop of 2.6 X 0.550= 1.43 psi/100 eq. Ft.

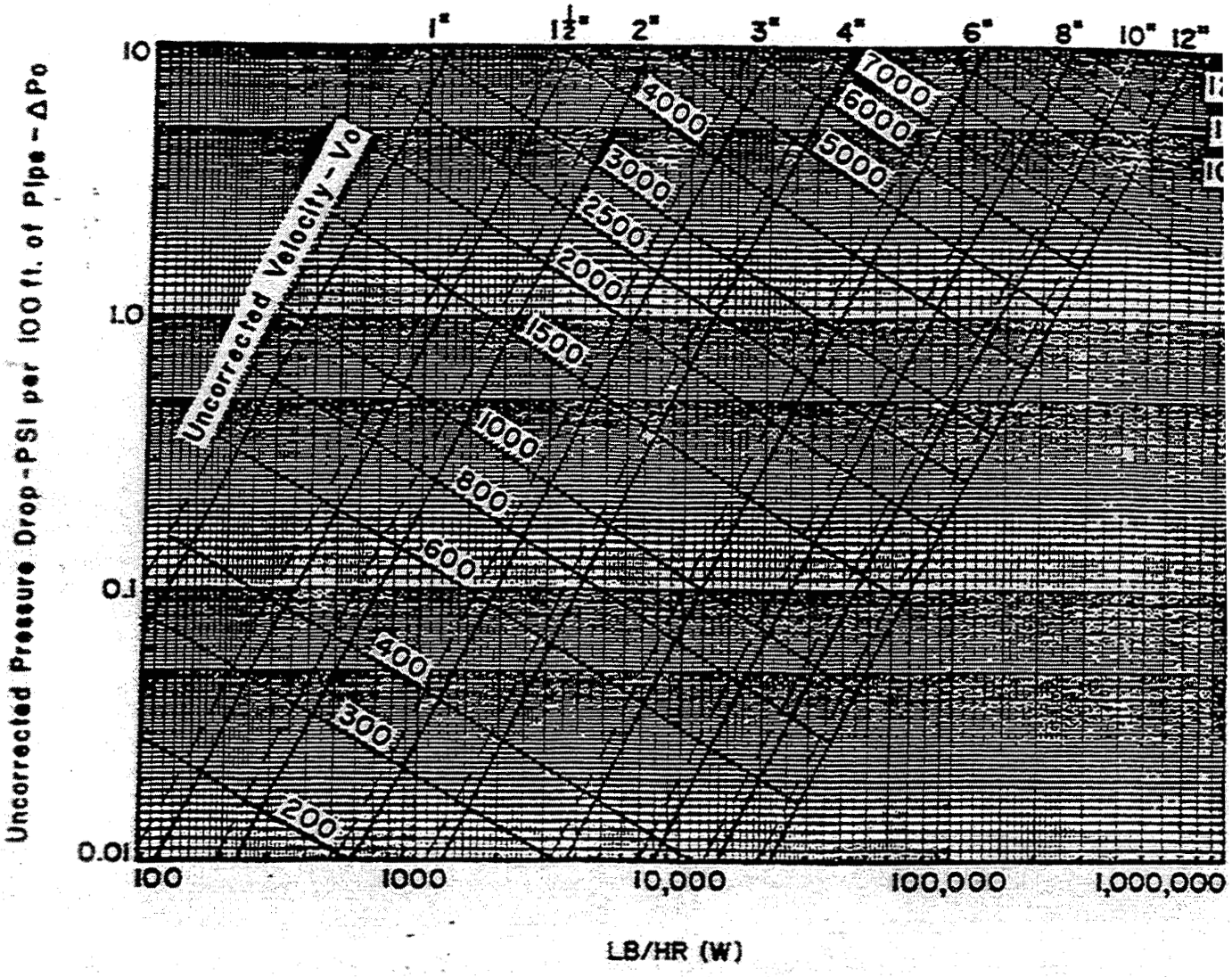
Heater Coil ΔP

Estimate the total coil SV:

$SV_{total\ coil} = 2/3\ SV_{coil\ outlet} + 1/3\ SV_{coil\ inlet}$

Use the total coil SV and the vapor line sizing chart to calculate the heater coil pressure drop.

Vapor Flow Thru Commercial Steel Pipe



$$\Delta Po = (f)\left(\frac{336}{d^5}\right)\left(\frac{W}{1000}\right)^2$$

Where

d = ID, inches

W = LB/HR

f =

0.027	for 1" Pipe
0.025	for 1½" Pipe
0.023	for 2" Pipe
0.022	for 3" Pipe
0.020	for 4" Pipe
0.018	for 6" Pipe
0.016	for 8" Pipe
0.015	for 10" Pipe
0.015	for 12" Pipe

Based on Viscosity of 0.015 Centipoise. Extrapolation Off Chart May Give Erroneous "f".

Corrected $\frac{\Delta PSI}{100'}$ = (ΔPo)(SV)

Corrected Vel. in FPM = (Vo)(SV)

Where

SV = Cu. Ft./Lb.

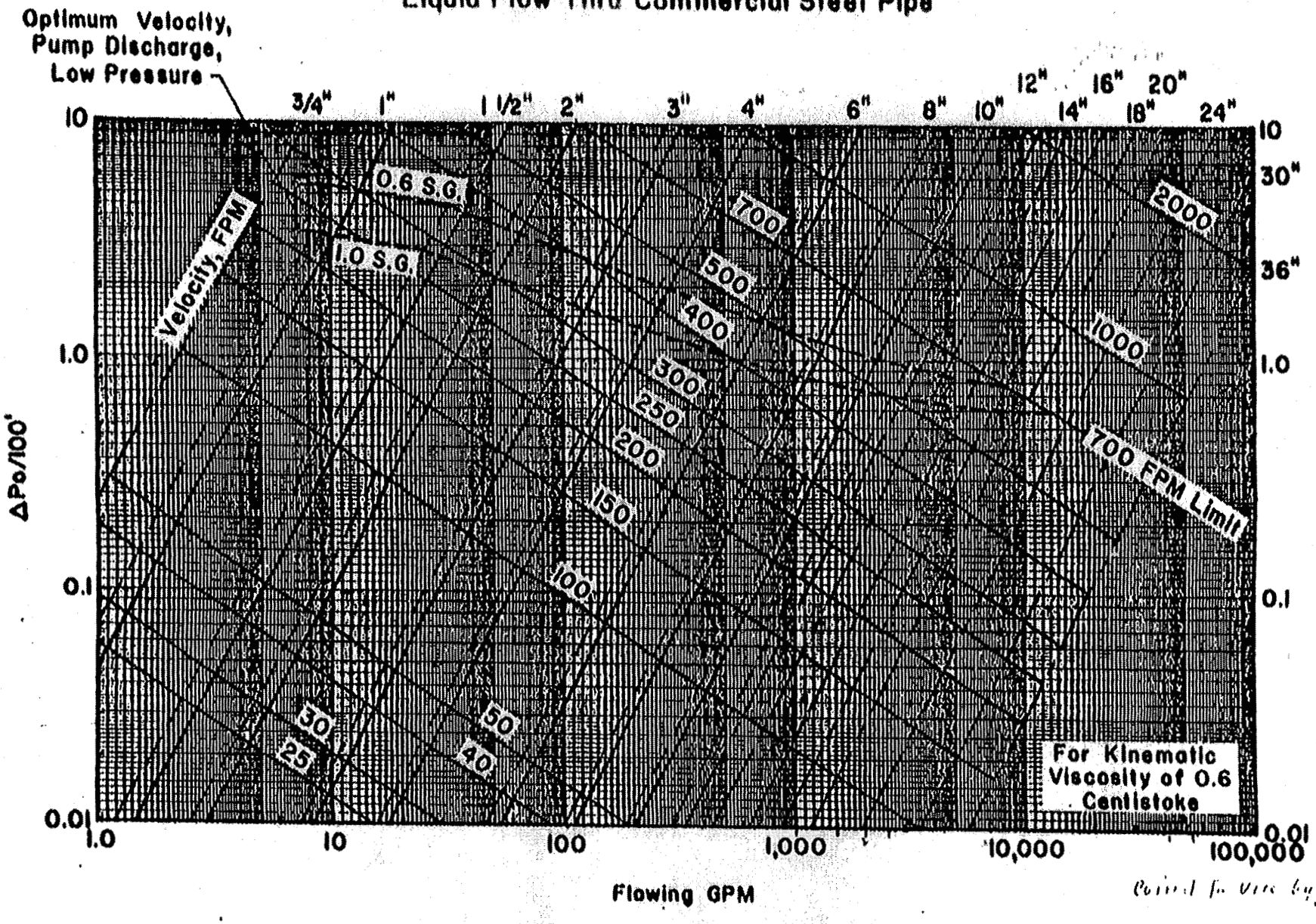
LEGEND

Sch. 80 — — —

Sch. 40 ———

PD 436
Redrawn 5-3-73

Liquid Flow Thru Commercial Steel Pipe



— — — — Sch. 80
——— Sch. 40
——— — — 3/8" Wall

Note:
To Find ΔP in PSI per 100 ft. of Pipe, Multiply ΔPo/100' by Specific Gravity.

5-3-78 REA

PD 435-C
(Supersedes PD 435-B)

Draft

Draft is negative air pressure generated by the buoyancy of hot gases inside the furnace. The pressure inside the furnace is negative because the hot gases are less dense than the outside air. These hot gases weigh less than the cooler air so they are buoyant inside the furnace. This buoyancy causes the hot gases to rise upward out of the stack, creating a slight vacuum inside the furnace. The vacuum causes the outside air to flow into the air registers. The pressure in the atmosphere is 14.7 pounds per square inch (PSI). A negative. pressure is any pressure below 14.7 PSI. The pressure inside the furnace is maintained just slightly below 14.7 PSI. This is called a negative pressure. The difference between the outside pressure and this negative pressure is what creates the draft. Draft is usually measured in three places: at the firebox floor, below the convection section, and below the stack damper. The most important draft reading is below the convection section because the negative pressure is smallest here. The small negative pressure is due to the tubes in the convection section which obstruct the flow of the upward moving gases. This resistance to flow can cause the pressure in the convection section to shift from slightly negative to slightly positive. When the pressure shifts positive, there is a loss of draft. With no draft, heat builds up just under the furnace arch and roof which can damage the structure of the furnace. A loss of draft also means that no air is pulled into the furnace so the burners will eventually go out. The furnace draft is usually controlled by positioning a damper in the stack. Opening the damper allows more flue gas to flow out the stack which, in turn, increases the draft throughout the entire furnace. The increase in draft is measured as an increase in negative pressure. When the damper is closed, the draft decreases. This is measured as a decrease in negative pressure. It's important to maintain the correct-furnace draft. Too little draft can damage the metal structure and snuff out the burners. Too much draft can pull excessive amounts of air into the furnace which wastes fuel.

Heater Design Procedure

1. Determine desired heater duty.

2. Assume radiant absorption equals 70-75% of total absorption.

3. Obtain a BWT from Section 11

4. Determine efficiency based on fuel properties, cold end approach temperature and anticipated excess air.

5. Determine final radiant- convection split using the UOP fuels program.

6. Determine fuel fired and flue gas rates.

7. Determine radiant section orientation
    a. Average radiant section heat flux.
    b. In-tube mass velocity.)
    c. Number of passes.
    d. Tube diameter and thickness.
    e. Number and length of tubes.
    f. Burner spacing.
    g. Box dimensions and refractory.

8. Determine the remaining convection section duty.

9. Calculated the radiant leakage into the convection section. Assume a 200° F drop in the shield tubes

10. Select the orientation of the shield bank (bare tubes only):
    a. Effective length of tubes.
    b. Number of tubes per row.
    c. Tube size.
    d. Number of passes

11. Determine applicable temperatures for shield section:

    a. Log mean temperature difference (LMTD).

    b. Average in-tube fluid temperature.

    c. Estimated average tube wall temperature (= average in-tube fluid temp. +100°F).

    d. Estimated average gas temperature (=average in-tube fluid temp. +LMTD).

    e. Estimated average gas film (=average in-tube fluid temp. +LMTD/2).

12. Calculate the maximum mass velocity per row through the convection section (evaluated at the plane intersecting the diameters of all the tubes in a row).

13. Calculate radiant, convection and wall coefficients.

14. Determine the wall radiation correction factor, and the outside coefficient.

15. Calculate the in-tube film coefficient. Determine the overall outside coefficient.

16. Determine the surface required and the number of rows.

17. Repeat the above sections 10-16 for any extended surface rows.

18. Steps 11-17 can be simplified by setting the flue gas mass velocity at 0.3 lb/ft^3-sec and estimating the required rows using the charts in Section 8.

19. Check the in-tube pressure differential across the heater. Redesign the heater if the pressure is unsatisfactory, adjust the number of passes and tube size if necessary.

20. Determine the stack height. Adjust the convection bank orientation if the draft across the convection bank is too high (stack becomes too high) or too low (flue gas distribution problems). Required pressure at the bridgewall = -0.10 in H_2O.

uop

Heater Rating Program

Customer: Training Problem
Location:
Project:

Unit: Crude
Item# 50,000 BPD
Comment: 31.5 API

Process Heat Absorption (MMBtu/hr)	
Radiant Duty	89.03
Convection Duty	38.69
Process Duty	127.72
Air Preheater Duty	0.00

Flow Rates (lb/hr)	
Process Fluid	632624
Combustion Air Rate	137007
Fuel Flow Rate	6550
Flue Gas Flow Rate	143557

Calculated Delta P (psi)	
Radiant	92.21
Convection	16.54
Total	108.75

Heat Flux (Btu/hr-ft^2)	
Avg Radiant Heat Flux	9963.4
Max Radiant Heat Flux	18819.3

	Heater Inlet Conditions	Crossover Zone Conditions	Heater Outlet Conditions
Temperature, °F	485.6	564.0	720.0
Pressure, psia	153.4	136.9	44.70
Liquid Flow, lb/hr	502223	540862	303633
Vapor Flow, lb/hr	30401	91773	328991
Liquid Viscosity, cp	0.372	0.310	0.491
Liquid Specific Heat, Btu/lb-°F	0.667	0.695	0.737
Liquid Thermal Cond, Btu/hr-ft-°F	0.039	0.038	0.054
Vapor Viscosity, cp	0.013	0.013	0.013
Vapor Specific Heat, Btu/lb-°F	0.638	0.664	0.702
Vapor Thermal Cond, Btu/hr-ft-°F	0.021	0.022	0.024
Tubeside Film Coefficient, Btu/hr-ft^2-°F	170.09	206.60	306.91

Combustion Design Conditions	
Fuel Type	Fuel Gas
Excess Air, %	15.0
Fuel Efficiency, %	84.7
Radiant Loss as a % of Heat Release	2.00
Flue Gas T Leaving Rad Section, °F	1464
Flue Gas T Leaving Conve. Section, °F	571
Max Flue Gas Mass Velocity, lb/ft^2-s	0.35
Air Temp at Burners, °F	60.0
Calc Heat Release Fuel only, MMBtu/hr	150.8
Calc Heat Release Air and Fuel, MMBtu/hr	150.8
Fuel Lower Heating Value, Btu/lb	23034.6
Fuel Mod. Heating Value, Btu/lb	23034.6
Adiabatic Flame Temp, °F	3340.8
Flue Gas Temp to Stack, °F	571.2
Lbs of Air per Lbs of Fuel	20.62

Mechanical Design Conditions	
Number of Passes at Radiant Outlet	4
Rad. Section Actual Surface Area, ft^2	8936
Rad Section Flat Plane Area, ft^2	5152
Rad Section Effective Plane Area, ft^2	4716
Avg. Rad Film Coeff, Btu/hr-ft^2-°F	252.5
Avg. Film Temp Difference, °F	44.0
Max Film Temp Difference, °F	100.1
Avg. Tube Wall T Difference, °F	15.8
Max. Tube Wall T Difference, °F	29.9
Avg. Rad Bulk Fluid Temp, °F	650.3
Avg Rad Tube Wall Temp, °F	706.8
Max Calc Rad Tube Wall Temp, °F	816.5
Front Face Tube Wall Temp, °F	759.9
Adjusted FF Tube Wall Temp, °F	759.9

Outlet Radiant Tube Thickness Calculations	
Required Minimum Thickness, in	0.2450
Available Minimum Thickness, in	1100
Max. Allowable Tube Wall Temp, °F	235.3
Design Pressure, psig	0.0625
Corrosion Allowance, in	

UOP

ENGINEERING DEPARTMENT
Equipment Skill Center

Customer: Training Problem	Unit: Crude
Location:	Item# 50,000 BPD
Project:	Comment: 31.5 API

Pressure Drop Calculations: Radiant Section — UOP Mainframe Method

Radiant Section 1

Tube No.	Temp °F	Press psia	Enthalpy Btu/lb	Wt% Vapor -	Velocity Ft/s	Sonic Vel Ft/s	Sp Vol ft³/lb	Avg TWT °F	Max TWT °F	Heat Transfer Coefficient Btu/hr-ft²-°F
1	720.00	44.70	-490.08	52.00	216.04	632.80	0.99	769.78	818.48	306.91
2	716.23	57.16	-496.20	47.99	159.03	643.59	0.73	766.40	816.14	298.70
3	711.47	68.13	-502.31	44.83	130.18	652.12	0.59	762.13	812.64	292.49
4	706.22	73.35	-508.43	42.09	111.50	659.65	0.51	756.96	808.55	287.32
5	699.77	79.45	-514.55	40.27	99.85	663.63	0.46	751.14	803.41	282.10
6	693.16	84.66	-520.67	38.53	90.63	667.33	0.41	745.18	798.12	277.05
7	686.45	89.79	-526.79	36.87	83.01	671.01	0.38	739.14	792.74	272.20
8	679.67	94.26	-532.91	35.27	76.57	674.60	0.35	733.04	787.29	267.52
9	672.83	98.36	-539.03	33.72	70.99	678.12	0.32	726.89	781.79	262.99
10	665.95	102.16	-545.16	32.21	66.07	681.59	0.30	720.70	776.26	258.58
11	659.03	105.68	-551.26	30.74	61.69	685.02	0.28	714.47	770.69	254.29
Inlet	652.05	108.96	-557.38							

Radiant 1 Duty, MMBtu/hr	42.58	Radiant 1 Hottel Factor	0.92
Radiant 1 Pressure Drop, psi	64.26	Radiant 1 Mass Velocity, lb/s-ft²	218.97

Radiant Section 2

Tube No.	Temp °F	Press psia	Enthalpy Btu/lb	Wt% Vapor -	Velocity Ft/s	Sonic Vel Ft/s	Sp Vol ft³/lb	Avg TWT °F	Max TWT °F	Heat Transfer Coefficient Btu/hr-ft²-°F
1	652.05	108.96	-557.38	29.30	57.73	686.43	0.26	708.17	765.07	250.09
2	644.99	112.02	-563.50	27.91	54.15	691.77	0.25	701.79	759.38	246.04
3	637.84	114.89	-569.62	26.56	50.88	695.08	0.23	695.35	753.61	242.09
4	630.66	117.59	-575.74	25.23	47.84	698.38	0.22	688.89	747.82	238.25
5	623.45	120.12	-581.86	23.92	45.01	701.68	0.21	682.41	742.02	234.50
6	616.22	122.51	-587.98	22.64	42.35	704.99	0.19	675.91	736.20	230.84
7	608.97	124.75	-594.10	21.37	39.85	708.30	0.18	669.41	730.37	227.27
8	601.70	127.02	-600.21	20.12	37.42	711.66	0.17	662.83	724.47	223.75
9	594.21	129.14	-606.33	18.97	35.27	714.69	0.16	656.19	718.60	219.92
10	586.67	131.14	-612.45	17.84	33.25	717.75	0.15	649.54	712.78	216.01
11	579.10	133.01	-618.57	16.73	31.31	720.86	0.14	642.90	706.97	212.18
Inlet	571.53	134.77	-624.69							

Radiant 2 Duty, MMBtu/hr	42.58	Radiant 2 Hottel Factor	0.92
Radiant 2 Pressure Drop, psi	25.81	Radiant 2 Mass Velocity, lb/s-ft²	218.97

Radiant Section 3

Tube No.	Temp °F	Press psia	Enthalpy Btu/lb	Wt% Vapor -	Velocity Ft/s	Sonic Vel Ft/s	Sp Vol ft³/lb	Avg TWT °F	Max TWT °F	Heat Transfer Coefficient Btu/hr-ft²-°F
1	571.53	134.77	-624.69	15.64	29.44	724.02	0.13	637.94	700.81	208.43
2	567.76	135.86	-627.75	15.07	28.46	725.72	0.13	634.66	697.78	206.50
Inlet	563.99	136.90	-630.81							

Radiant 3 Duty, MMBtu/hr	3.87	Radiant 3 Hottel Factor	0.92
Radiant 3 Pressure Drop, psi	2.13	Radiant 3 Mass Velocity, lb/s-ft²	218.97

UOP

ENGINEERING DEPARTMENT
Equipment Skill Center

Customer: Training Problem	Unit: Crude
Location:	Item# 50,000 BPD
Project:	Comment: 31.5 API

Pressure Drop Calculations: Convection Section — UOP Mainframe Method

Convection Section 1

Service: Radiant/Convective Stream

Tube No.	Temp, °F	Press, psia	Enthalpy, Btu/lb	Wt% Vapor	Vel, Ft/s	Sonic Vel, Ft/s	Sp Vol, ft³/lb	HT Coeff, Btu/hr-ft²-°F
1	563.99	136.90	-630.81	14.51	27.49	727.43	0.13	204.60
2	561.15	137.91	-633.13	14.06	26.70	728.84	0.12	203.09
Inlet	558.31	138.89	-635.45					
Row 1 Duty	2.94 MMBtu/hr							
1	558.31	138.89	-635.45	13.61	25.93	730.25	0.12	201.59
2	555.75	139.85	-637.56	13.21	25.23	731.57	0.12	200.22
Inlet	553.18	140.77	-639.66					
Row 2 Duty	2.66 MMBtu/hr							
1	553.18	140.77	-639.66	12.80	24.53	732.90	0.11	198.86
2	550.84	141.67	-641.57	12.42	23.89	734.15	0.11	197.61
Inlet	548.46	142.55	-643.48					
Row 3 Duty	2.42 MMBtu/hr							

Convection 1 Duty, MMBtu/hr 8.02

Convection 1 Pressure Drop, psi 5.65 — Convection 1 Mass Velocity, lb/s-ft² 218.97

Convection Section 2

Service: Radiant/Convective Stream

Tube No.	Temp, °F	Press, psia	Enthalpy, Btu/lb	Wt% Vapor	Vel, Ft/s	Sonic Vel, Ft/s	Sp Vol, ft³/lb	HT Coeff, Btu/hr-ft²-°F
1	548.46	142.55	-643.48	12.07	23.29	735.44	0.11	196.31
2	540.50	143.41	-649.67	11.14	21.93	738.94	0.10	192.69
Inlet	532.49	144.22	-655.86					
Row 1 Duty	7.83 MMBtu/hr							
1	532.49	144.22	-655.86	10.21	20.58	742.56	0.09	189.14
2	526.44	144.97	-660.55	9.50	19.52	745.45	0.09	186.45
Inlet	520.38	145.69	-665.24					
Row 2 Duty	5.93 MMBtu/hr							

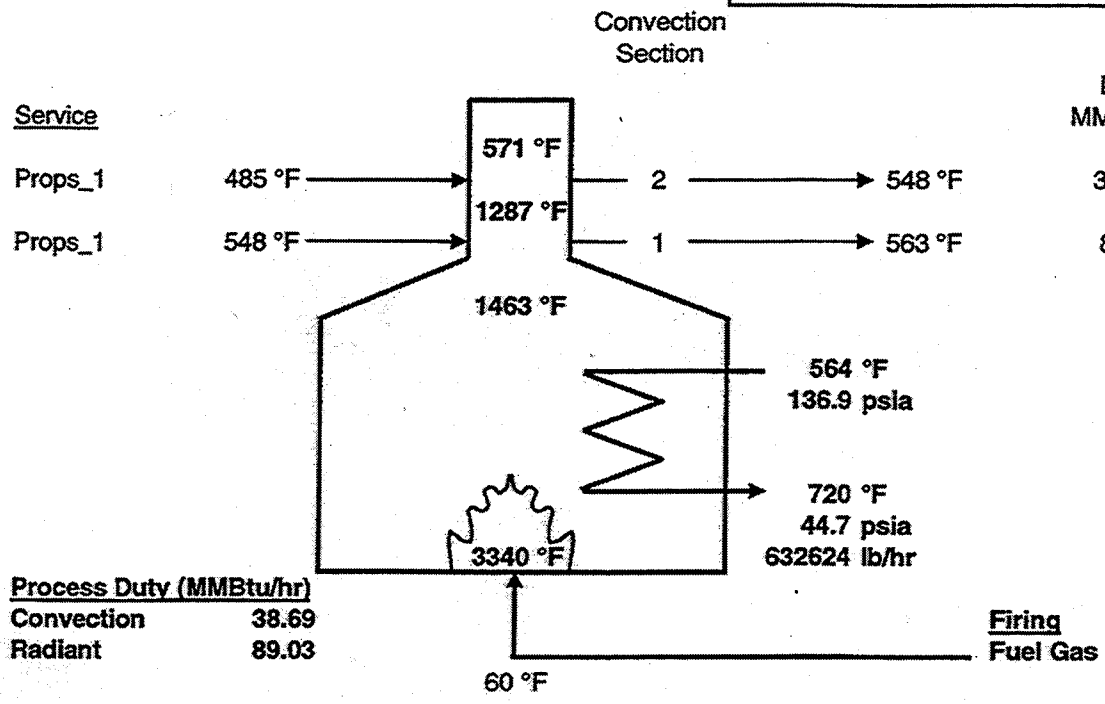
1	520.38	145.69	-665.24	8.79	18.48	748.41	0.08	183.80
2	515.76	146.38	-668.83	8.24	17.65	750.79	0.08	181.76
Inlet	511.13	147.03	-672.41					
Row 3 Duty		4.54 MMBtu/hr						
1	511.13	147.03	-672.41	7.70	16.82	753.22	0.08	179.76
2	507.57	147.65	-675.19	7.26	16.16	755.19	0.07	178.18
Inlet	504.01	148.26	-677.96					
Row 4 Duty		3.51 MMBtu/hr						
1	504.01	148.26	-677.96	6.84	15.51	757.19	0.07	176.63
2	501.23	148.83	-680.11	6.49	14.98	758.84	0.07	175.40
Inlet	498.42	149.39	-682.26					
Row 5 Duty		2.72 MMBtu/hr						
1	498.42	149.39	-682.26	6.18	14.51	760.68	0.07	174.26
2	496.22	149.94	-683.94	5.96	14.17	762.30	0.06	173.41
Inlet	493.99	150.47	-685.61					
Row 6 Duty		2.12 MMBtu/hr						
1	493.99	150.47	-685.61	5.73	13.83	763.96	0.06	172.56
2	492.24	150.99	-686.93	5.55	13.54	765.34	0.06	171.87
Inlet	490.49	151.50	-688.25					
Row 7 Duty		1.67 MMBtu/hr						
1	490.49	151.50	-688.25	5.36	13.26	766.73	0.06	171.19
2	489.13	152.00	-689.29	5.21	13.02	767.90	0.06	170.64
Inlet	487.76	152.49	-690.33					
Row 8 Duty		1.31 MMBtu/hr						
1	487.76	152.49	-690.33	5.06	12.78	769.08	0.06	170.09
2	486.69	152.97	-691.15	4.93	12.58	770.08	0.06	169.64
Inlet	485.62	153.45	-691.97					
Row 9 Duty		1.04 MMBtu/hr						

Convection 2 Duty, MMBtu/hr	30.67		
Convection 2 Pressure Drop, psi	10.89	Convection 2 Mass Velocity, lb/s-ft²	218.97

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ENGINEERING DEPARTMENT
Equipment Skill Center
9/9/2003 15:52

Customer: Training Problem
Location:
Project:
Unit: Crude
Item# 50,000 BPD
Comment: 31.5 API



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ENGINEERING DEPARTMENT
Equipment Skill Center

Customer: Training Problem	Unit: Crude
Location:	Item# 50,000 BPD
Project:	Comment: 31.5 API

Convection Section #	1
Width of Convection Section, In	102.4
Number of Passes	4
Metallurgy	5Cr
Extended Surface (Fins or Studs)	
Tubes/Row	8
Tube O.D., Inches	6.625
Tube I.D, Inches	6.045
Length, Feet	28
Horizontal Spacing, Inches	12
Diagonal Spacing, Inches	12
Extended Surface Metallurgy	5Cr
Fin or Stud Height, Inches	0
Fins Per Inch or Studs Per Plane	0
Fin Thickness, Inches	0
Corbel Height, Inches	0
Extension Ratio	1.00
Clearance To Wall	5.67
Section Duty, MMBtu/hr	8.02
Section Flue Gas, Inches of H2O	0.0000

R/C Stream Flow Rate, lb/hr	63262	**Inlet Specific Heat, Btu/lb/°F**	0.691
Inlet Process Temp,°F	548	**Outlet Specific Heat, Btu/lb/°F**	0.688
Process Out Temp,°F	564	**Inlet Heat Transfer, Btu/h/°F/ft**	197,610
FG Mass Velocity, lb/s-ft²	0.351	**Outlet Heat Transfer, Btu/h/°F/ft**	204,599
FG Temp In, °F	1464	**Inlet Specific Volume, ft³/lb**	0.122
FG Temp Out, °F	1287	**Outlet Specific Volume, ft³/lb**	0.109
FG Flow Rate, lbs/hr	14,557		
FG MW	27.34		

	Flue Gas Compostion (wet)		Flue Gas Compostion (dry)	
	Wt%	**Mol%**	**Wt%**	**Mol%**
CO2	11.88	7.38	13.63	9.16
SO2	0.00	0.00	0.00	0.00
H2O	12.83	19.48	0.00	0.00
O2	2.67	2.45	3.29	3.04
N2	72.42	70.69	83.08	87.79
	100.00	100.00	100.00	100.00

Row Number	**1**			
Flue Gas Mass Velocity, lb/hr-ft²	0.351		Flux (Bare Tube) Btu/hr-ft²	7616
5Cr Tubes				
	In	**Out**		
Flue Gas Temp °F	1463.9	1399.5		
Process Fluid Temp °F	558.3	564.0		
Process Film Temp °F	639.2	645.7		
Tube Metal Temp °F	652.9	659.4		
	This Row	**Cumulative**		
Duty MMBtu/hr	2.94	2.94		
Flue Gas Pressure Drop, in of H2O	0.00	0.00		
Surface (ft²)	390.9	390.9		

Row Number	2			
Flue Gas Mass Velocity, lb/hr-ft²	0.351		Flux (Bare Tube) Btu/hr-ft²	6805
5Cr Tubes				
	In	**Out**		
Flue Gas Temp °F	1399.5	1340.8		
Process Fluid Temp °F	553.2	558.3		
Process Film Temp °F	626.8	632.8		
Tube Metal Temp °F	639.2	645.2		
	This Row	**Cumulative**		
Duty MMBtu/hr	2.66	5.60		
Flue Gas Pressure Drop, in of H2O	0.00	0.00		
Surface (ft²)	390.9	781.7		

Row Number	3			
Flue Gas Mass Velocity, lb/hr-ft²	0.351		Flux (Bare Tube) Btu/hr-ft²	6186
5Cr Tubes				
	In	**Out**		
Flue Gas Temp °F	1340.8	1287.1		
Process Fluid Temp °F	548.5	553.2		
Process Film Temp °F	615.8	621.4		
Tube Metal Temp °F	627.0	632.6		
	This Row	**Cumulative**		
Duty MMBtu/hr	2.42	8.02		
Flue Gas Pressure Drop, in of H2O	0.00	0.00		
Surface (ft²)	390.9	1172.6		

UOP
ENGINEERING DEPARTMENT
Equipment Skill Center

Customer: Training Problem	Unit: Crude
Location:	Item# 50,000 BPD
Project:	Comment: 31.5 API

Convection Section #	2
Width of Convection Section, In	102
Number of Passes	2
Metallurgy	5Cr
Extended Surface (Fins or Studs)	Fins
Tubes/Row	8
Tube O.D., Inches	6.625
Tube I.D, Inches	6.065
Length, Feet	28
Horizontal Spacing, Inches	12
Diagonal Spacing, Inches	12
Extended Surface Metallurgy	5Cr
Fin or Stud Height, Inches	1
Fins Per Inch or Studs Per Plane	5
Fin Thickness, Inches	0.05
Corbel Height, Inches	0
Extension Ratio	12.57
Clearance To Wall	4.67
Section Duty, MMBtu/hr	30.67
Section Flue Gas, Inches of H2O	0.2774

R/C Stream Flow Rate, lb/hr	632624
Inlet Process Temp,°F	486
Process Out Temp,°F	548
FG Mass Velocity, lb/s-ft²	0.383
FG Temp In, °F	1287
FG Temp Out, °F	571
FG Flow Rate, lbs/hr	143557
FG MW	27.34

Inlet Specific Heat, Btu/lb/°F	0.687
Outlet Specific Heat, Btu/lb/°F	0.647
Inlet Heat Transfer, Btu/h/°F/ft	169,637
Outlet Heat Transfer, Btu/h/°F/ft	196,306
Inlet Specific Volume, ft³/lb	0.100
Outlet Specific Volume. ft³/lb	0.057

	Flue Gas Compostion (wet)		Flue Gas Compostion (dry)	
	Wt%	**Mol%**	**Wt%**	**Mol%**
CO2	10.88	7.38	13.63	9.16
SO2	0.00	0.00	0.00	0.00
H2O	12.03	19.45	0.00	0.00
O2	2.67	2.45	3.29	3.04
N2	72.42	70.69	83.08	87.79
	100.00	100.00	100.00	100.00

Row Number	1			
Flue Gas Mass Velocity, lb/hr-ft²	0.383			
1 Inch Fins, 5 Fins/Inch				
5Cr Fins			Extend Surf. Tip Temp °F	964.3
5Cr Tubes			Extension Efficiency	0.73
	In	**Out**	Flux (Bare Tube) Btu/hr-ft²	20027
Flue Gas Temp °F	1287.1	1110.8	****Flux Exceeds 20,000 Btu/hr-ft² for Row 1**	
Process Fluid Temp °F	532.5	548.5		
Process Film Temp °F	753.7	770.6		
Tube Metal Temp °F	790.5	807.4		
	This Row	**Cumulative**		
Duty MMBtu/hr	7.83	7.83		

Surface (ft²)	4915.2	4915.2

Row Number	**2**			
Flue Gas Mass Velocity, lb/hr-ft²	0.383			
1 Inch Fins, 5 Fins/Inch				
5Cr Fins			Extend Surf. Tip Temp °F	851.9
5Cr Tubes			Extension Efficiency	0.74
	In	**Out**	Flux (Bare Tube) Btu/hr-ft²	15173
Flue Gas Temp °F	1110.8	974.5		
Process Fluid Temp °F	520.4	532.5		
Process Film Temp °F	690.9	704.0		
Tube Metal Temp °F	718.6	731.8		
	This Row	**Cumulative**		
Duty MMBtu/hr	5.93	13.76		
Flue Gas Pressure Drop, in of H2O	0.04	0.08		
Surface (ft²)	4915.2	9830.3		

Row Number	**3**			
Flue Gas Mass Velocity, lb/hr-ft²	0.383			
1 Inch Fins, 5 Fins/Inch				
5Cr Fins			Extend Surf. Tip Temp °F	767.3
5Cr Tubes			Extension Efficiency	0.75
	In	**Out**	Flux (Bare Tube) Btu/hr-ft²	11615
Flue Gas Temp °F	974.5	868.4		
Process Fluid Temp °F	511.1	520.4		
Process Film Temp °F	643.3	653.7		
Tube Metal Temp °F	664.5	674.8		
	This Row	**Cumulative**		
Duty MMBtu/hr	4.54	18.30		
Flue Gas Pressure Drop, in of H2O	0.03	0.11		
Surface (ft²)	4915.2	14745.5		

Row Number	**4**			
Flue Gas Mass Velocity, lb/hr-ft²	0.383			
1 Inch Fins, 5 Fins/Inch				
5Cr Fins			Extend Surf. Tip Temp °F	703.1
5Cr Tubes			Extension Efficiency	0.75
	In	**Out**	Flux (Bare Tube) Btu/hr-ft²	8966
Flue Gas Temp °F	868.4	785.4		
Process Fluid Temp °F	504.0	511.1		
Process Film Temp °F	606.9	615.2		
Tube Metal Temp °F	623.2	631.5		
	This Row	**Cumulative**		
Duty MMBtu/hr	3.51	21.81		
Flue Gas Pressure Drop, in of H2O	0.03	0.14		
Surface (ft²)	4915.2	19660.7		

Row Number	**5**			
Flue Gas Mass Velocity, lb/hr-ft²	0.383			
1 Inch Fins, 5 Fins/Inch				
5Cr Fins			Extend Surf. Tip Temp °F	653.7

Surface (ft²)	4915.2	39321.3

Row Number	**9**			
Flue Gas Mass Velocity, lb/hr-ft²	0.383			
1 Inch Fins, 5 Fins/Inch				
5Cr Fins			Extend Surf. Tip Temp °F	544.7
5Cr Tubes			Extension Efficiency	0.77
	In	**Out**	Flux (Bare Tube) Btu/hr-ft²	2652
Flue Gas Temp °F	596.6	571.2		
Process Fluid Temp °F	485.6	487.8		
Process Film Temp °F	515.9	519.5		
Tube Metal Temp °F	520.7	524.3		
	This Row	**Cumulative**		
Duty MMBtu/hr	1.04	30.67		
Flue Gas Pressure Drop, in of H2O	0.03	0.28		
Surface (ft²)	4915.2	44236.5		

UOP

Heater Rating Program

Input Sheet

	Customer	Training Problem			Unit	Crude			
	Location				Item #	50,000 BPD			
	Project				Comment:	31.5 API			
					Comment:				

	Flow Rate	Radiant Outlet Temp	Radiant Outlet Press
	lb/hr	°F	psia

	Zones in Convection Sections	Radiant Sections	Total Duty	Fouling Temperature Difference	%Radiant Heat Loss	%Radiant of Total Duty (Initial Guess)	Tube Emissivity	Convection Beam Length
	#	#	MMBtu/hr	°F				
	2	3		0.0	2.0	70.00	0.85	0

	Mal Distribution Factor	Heat Loss Per Wall Area	Process Side Fouling Factor	Flue Gas Side Fouling Factor	Lower Rows Present?	Design Press	Corrosion Allowance	Fuel Fired Factor
		Btu/hr-ft²	1/(Btu/h-ft2-°F)	1/(Btu/h-ft2-°F)	1	psia	in	
	1.0	0.0	0.005	0.01	TRUE	250	0.0625	1.00

Radiant Section Input

	Section Number	Number of Tubes	Number of Passes	Tube OD	Tube ID	Tubes (AW or MW)	Straight Tube Length	Tube Metallurgy
	#			in	in		ft	
	1	44	4	6.625	6.065	AW	56.0	9Cr
	2	44	4	6.625	6.065	AW	56.0	9Cr
	3	8	4	6.625	6.065	AW	28.0	9Cr

	Section Number	Tube spacing CL to CL	Equivalent length of return bend	Return Bend in Firebox?	SingleFired=1 DoubleFired=2	Hottel Factor		
	#	in	L/D					
	1	12	50	FALSE	1			2
	2	12	50	FALSE	1			2
	3	12	50	FALSE	1			2

Convection Section

Section Number	Number of Passes	Number of Tubes Per Row	Number of Tube Rows	Tube OD	Tube ID	Tubes (AW or MW)	Effective Tube Length	Tube Metallurgy	Horizontal Tube Spacing	Diagonal Tube Spacing	Equivalent Length of Return Bend	Corbel Height	Wall to Wall Section Width
				in	in		ft		in	in	L/D	in	in
1	4	8	3	6.665	6.065	AW	28.0	SCr	12	12	50	0	
2	4	8	9	6.665	6.065	AW	28.0	SCr	12	12	50	0	

Section Number	Extended Surface Type	Fin or Stud Height	Pitch, fins/inch studs/plane	Metal	Fin Thickness		
		in			in		
1	None			SCr		1	4
2	Fins	1.0	5	SCr	0.05	2	4

Fuel Data - see Oil and Gas Sheets to enter in data

%Fuel Gas Fired		Soot Blower/ Open Lane, Row #:					
			3	10	0	0	
100.0							

Air Preheater Section Input

Steam Preheater Press	Steam Approach Temp	Ambient Air Temp
psig	°F	°F
150	10	60

SteamPreheat ☒ Yes ☐ No

Air Preheat
- ☒ None
- ☐ FG Acid DP Approach
- ☐ Hot End Approach Temp
- ☐ Air Combustion Temp

0

Stack Specification Input

Site Elevation	Ambient Air Temperature	Convection Height	Stack Temperature Drop	Bridgewall dP	Damper dP	Flue Gas Stack Velocity
ft	°F	ft	°F	in of H_2O	in of H_2O	ft/s
0	90	15	40	0.10	0.05	25.0

Stack Input Type:
- ☒ Stack Diameter 2
- ☐ Flue Gas Stack Velocity

Simultaneity Switches

Combined Flue Gas?
- ☐ Yes
- ☒ No

1.00

Steam Generation

☐ Non ☐ Typical ☒ Steam Superheat

Superheater Only

Guess Value

Steam Inlet Pressure	Steam Inlet Temperature	Steam Flow Rate	Steam Outlet Temperature
psia	°F	lb/hr	°F
50	350	2000	400

Alternate Process Stream in Convection Section 0

☒ Non ☐ One ☐ Two Streams ☐ Shielded Process

Training Services

Heater Instrumentation

Section Outline

- Process Flow
- Process Pressure
- Process Temperature
- Combustion Air Flow
- Pilot Gas
- Fuel Gas
- Fuel Oil

Section Outline
(continued)

- Atomizing Steam
- Fuel Input Control
- Draft
- Flue Gas Monitoring
- Flue Gas Temperature
- Fan Controls

Section Outline
(continued)

- Oxygen Control
- Combustible Control
- Air Preheater Instruments
- Steam System Instruments
- Miscellaneous Controls
- Burner Management Controls

Process Instrumentation - Reboiler

- **FRC flow control to each pass**
- **Low flow trips heater (FAL)**
- **Each pass outlet has high temperature alarm (TAH)**
- **TRC control if vaporizing curve slope greater than 0.2 (10°F/50% vapor)**
- **PDRC control if vaporizing curve slope less than 0.2**

Process Instrumentation - Reactor

- **FRC flow control not required**
- **Loss of recycle compressor trips heater (FAL)**
- **Each pass outlet has high temperature alarm (TAH)**

Firing Control - Fuel Gas

- PIC pressure control of fuel to each burner
- High and low pressure alarms (PAL/PAH)
- HIC control and board-mounted high signal selector act as minimum stop to keep control valve from closing
- HIC minimum firing set around 2 psig

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Firing Control - Fuel Oil

- PIC pressure control of fuel to each burner
- High and low pressure alarms (PAL/PAH)
- HIC control and board-mounted high signal selector act as minimum stop to keep control valve from closing
- HIC minimum firing set around 30 psig
- Atomizing steam flow set by differential pressure control (DPIC)
- DPIC minimum set around 25-30 psig
- Loss of steam stops oil flow

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Firing Control - Pilot Gas

- PCV pressure regulation of gas pressure at 10-12 psig
- Low pressure (<10 psig) trips pilots and main fuel

Flue Gas

- TI provided at radiant section exit (bridgewall) and convection section outlet (stack) to measure heater efficiency
- Oxygen analyzer (AI) provided at bridgewall to monitor excess air
- UOP specifies Thermox analyzer by Amtek (zirconium oxide-type sensing element)

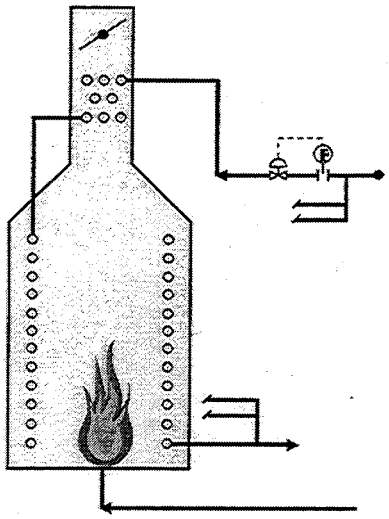
Forced Draft

- Two fans specified
- Manual variable inlet vanes provide turndown ratio for startup
- FIC venturi meters measure air flow; transmitter range 25 mm H_2O minimum
- Opposed blade dampers control air flow
- Burner air side pressure drop is 25-50 mm H_2O

Forced Draft
(continued)

- No natural draft drop out doors
- PIC pressure control at bridgewall to maintain 3 mm H_2O draft
- Time delay used on high firebox pressure shutdown
- Self-checking flame scanner used for each pilot for forced draft burners

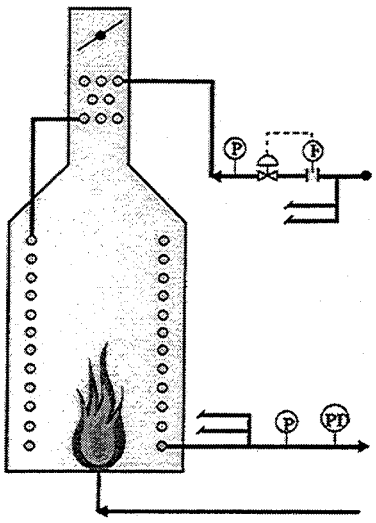
Process Flow



- Flow control to each pass in vaporizing service
- Low flow stops heater firing
- Minimum stop on valve prevents no flow
- Total flow rate used for heat balance
- Flow control not required for all vapor applications

FH 2004/9-13

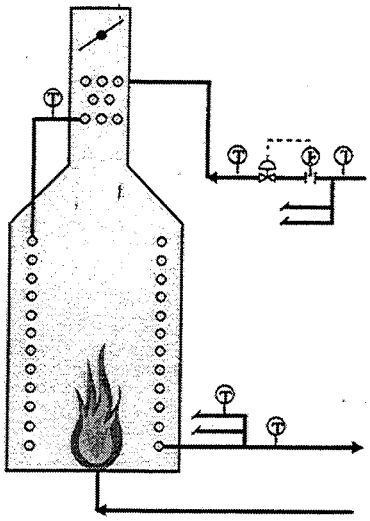
Process Pressure



- Measured on individual passes to check flow distribution
- Measured on individual passes to check for coking and/or fouling

FH 2004/9-14

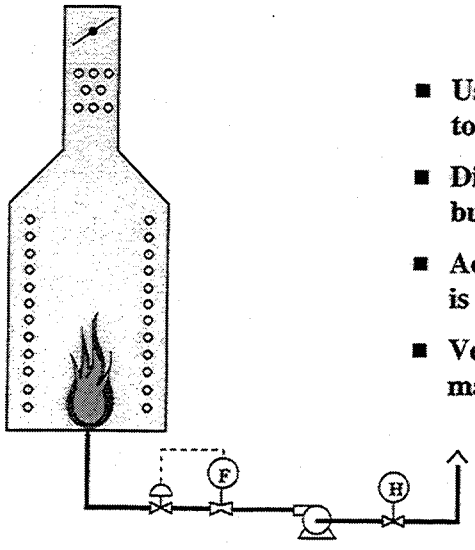
Process Temperature



- Inlet temperature used for heat balance
- Outlet temperature on each pass for flow distribution
- Total flow outlet temperature used for fuel control

FH 2004/9-15

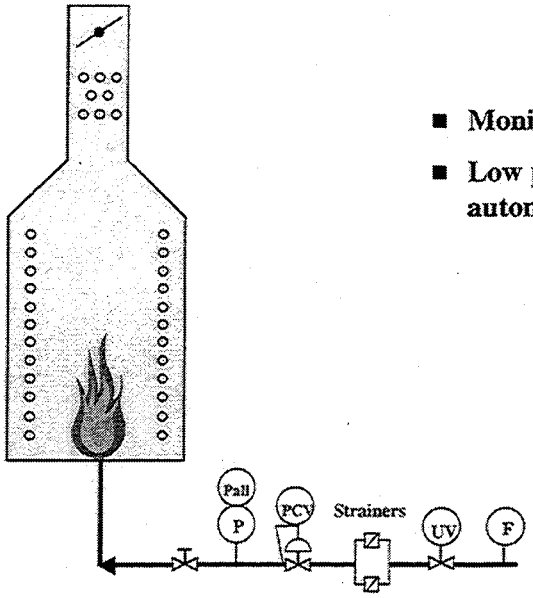
Combustion Air Flow



- Used in forced draft systems to assist combustion safety
- Differential pressure across burners can be used
- Accuracy of measurement is difficult
- Venturi meters, air foils, mass flow meters used

FH 2004/9-16

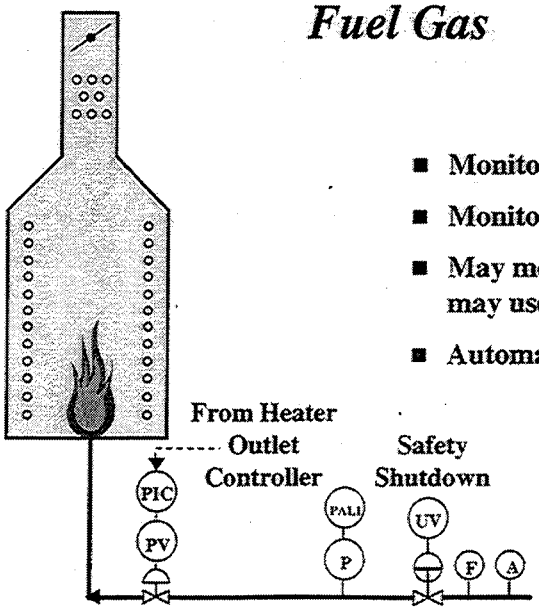
Pilot Gas



- Monitor pressure
- Low pressure leads to automatic shutdown

FH 2004/9-17

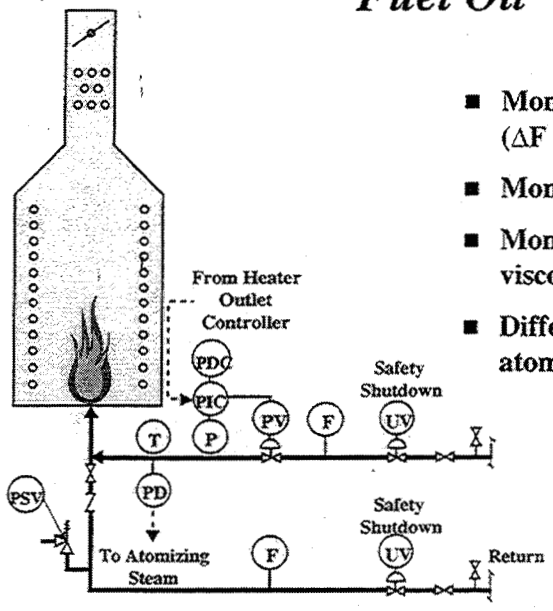
Fuel Gas



- Monitor flow for heat balance
- Monitor pressure
- May monitor heating value - may use gas density
- Automatic shutdown possible

FH 2004/9-18

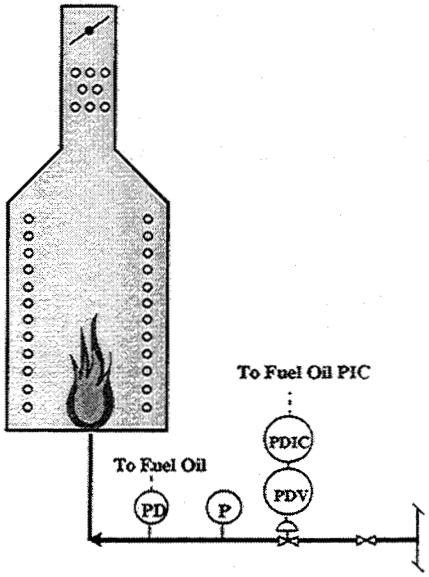
Fuel Oil



- Monitor flow for heat balance (ΔF for oil return)
- Monitor pressure
- Monitor temperature for viscosity control
- Differential pressure to atomizing steam

FH 2004/9-19

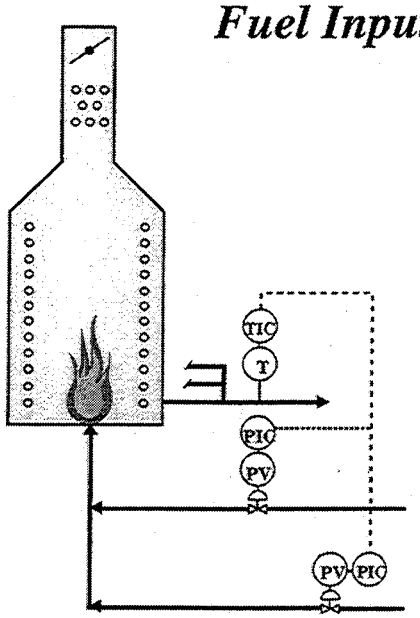
Atomizing Steam



- Pressure of supply
- Differential pressure to fuel oil (25 psi higher than oil typical)

FH 2004/9-20

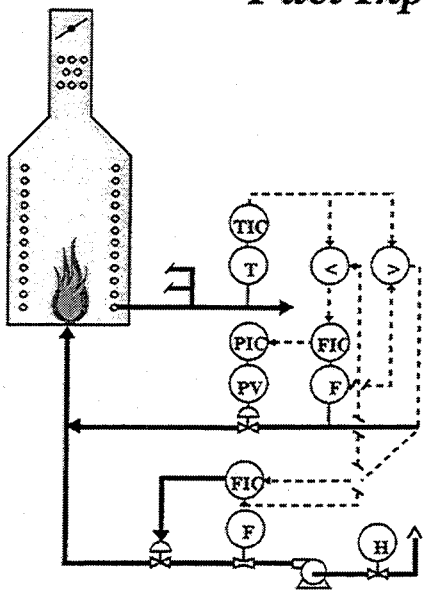
Fuel Input Control



- Measure heater outlet temperature to control fuel flow valve
- Split signal for multiple fuels

FH 2004/9-21

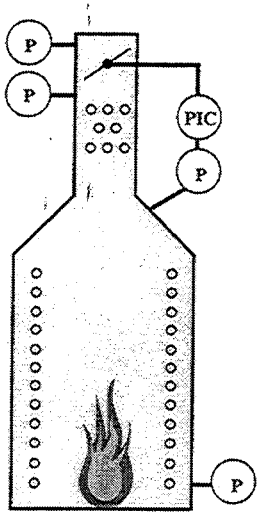
Fuel Input Control



- Lead-lag system for combustion air

FH 2004/9-22

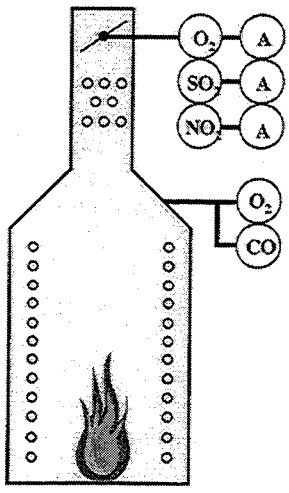
Draft



- Measure draft at burners, bridgewall, top convection, below damper and stack
- Use stack damper to control draft at bridgewall

FH 2004/9-23

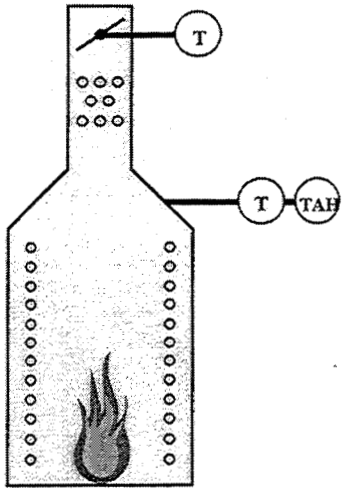
Flue Gas Monitoring



- Depends on local requirements
- Manual vs. automatic monitoring
- Oxygen monitoring at bridgewall for control, stack for emission monitoring
- Combustibles monitoring at bridgewall
- Emissions monitoring in stack
- Oxygen/CO reset of fuel control can be done but analyzers are not reliable enough

FH 2004/9-24

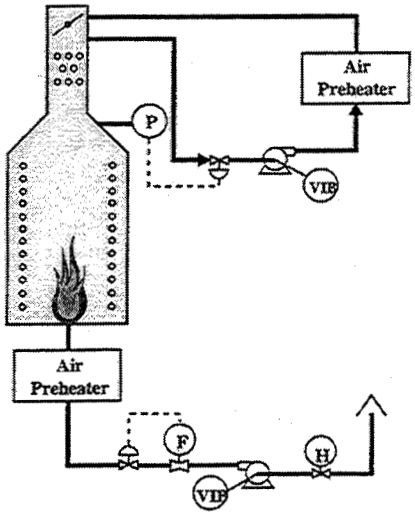
Flue Gas Temperature



- **Bridgewall temperature - velocity thermocouple better accuracy but costly and not normally used**
- **Stack temperature**

FH 2004/9-25

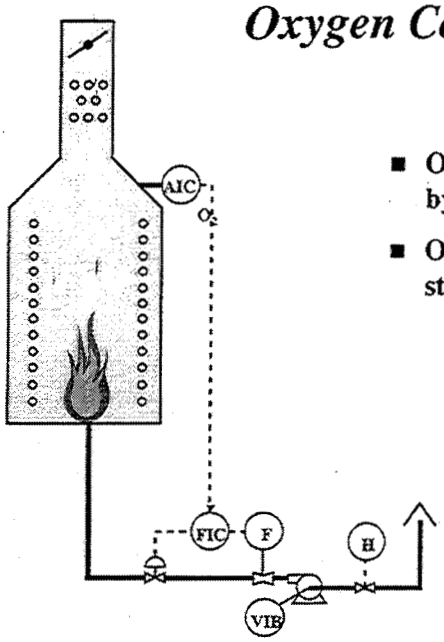
Fan Controls



- **Forced draft fan inlet damper - oxygen control**
- **Induced draft fan inlet damper - draft control**
- **Speed monitoring - detects fan trip**
- **Vibration monitoring - preventative maintenance**
- **Spare fan - auto start on trip**

FH 2004/9-26

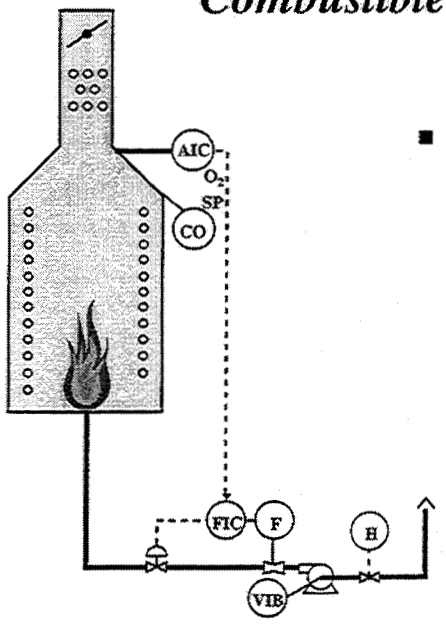
Oxygen Control



- Oxygen analyzer trims fuel input by regulating fuel flow valves
- Oxygen feedback to stack damper

FH 2004/9-27

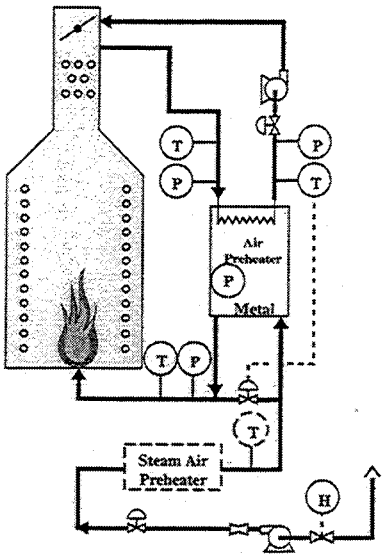
Combustible Control



- Carbon monoxide trims fuel by minimizing oxygen set point. Typically used only on large boilers but few heaters

FH 2004/9-28

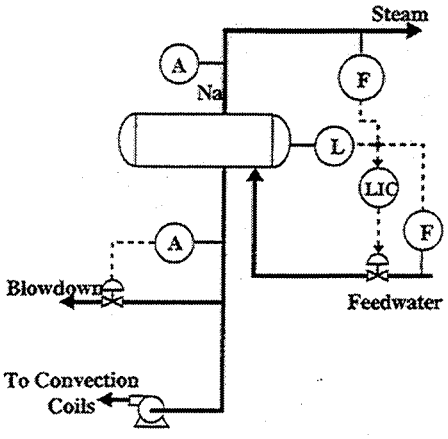
Air Preheater Instruments



- Cold end temperature measurement
- Cold air bypass controls

FH 2004/9-29

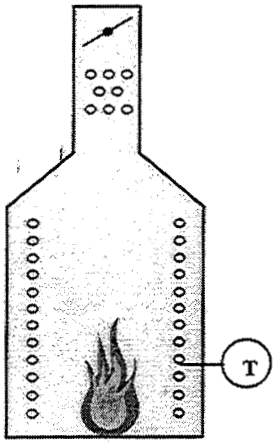
Steam System Instruments



- Steam drum level controls - 3 element control (drum level, steam flow rate, feedwater flow rate)
- Auto start of circulating water pumps
- Blowdown controls - conductivity for dissolved solids
- Steam purity measurements

FH 2004/9-30

Miscellaneous Instruments/Controls



- Sootblowers
  - Automatic sequencing panel
- Tube metal temperature measurement

FH 2004/9-31

Burner Management Controls



- Safety of Operations
- Flame Scanners
- Automatic start up sequencing
- Prefer manual control during startups
- Automatic trip systems possible

FH 2004/9-32

This is a hidden slide
This is to indicate which 11x17 drawings are requested at the back of this section

11 x 17 Fired Heater piping & Instrument Diagram #1018010

11 x 17 Fired Heater piping & Instrument Diagram #1018015

*** FURNISHED BY HEATER VENDOR/CONTRACTOR

TO STACK
TI
TE
ISOLATION BLIND (LOCATE IN HORIZONTAL RUN)
PV
PY
FO
DAMPER
ISOLATION BLIND (LOCATE IN HORIZONTAL RUN)
FROM INDUCED DRAFT FAN
TO COMBUSTION AIR PREHEATER
TE
TI
PY B
HIGH PRESSURE SIGNAL OPENS THIS DAMPER LAST
PIC
PAH
PT
SPLIT RANGE
PSHH
HIGH PRESSURE SIGNAL OPENS THIS DAMPER FIRST
PY A
DRAFT GAUGE (LOCATE GAUGE AND VALVES AT GRADE)
PI
DT
¾"S SAMPLE NUMBER
FOR BURNER PIPING ARRANGEMENT SEE STD DWG 2-101
AT
OXYGEN
AI
AAL
AR
TE
TAH
TI
HEATER SEE DWG
BURNER FUEL GAS PRESSURE (PORTABLE, ONE PER HEATER)
PI
PI
BA
BE
ONE FOR EACH BURNER
COMBUSTION AIR (PORTABLE, ONE PER HEATER)
PROVIDE PURGE STEAM TO HEATER FIREBOX
FROM CONTROLLER AT HEATER OUTLET
HIGH TEMPERATURE SIGNAL DECREASES FUEL
TY A
LOW TEMPERATURE SIGNAL INCREASES AIR
TOTAL FUEL
FIC
TY B
HY B
HIC
HEADER
PROVIDE LIMIT STOP
FV
OPPOSED BLADE DAMPER
FO
FY
FE
FT
VENTURI
FROM COMBUSTION AIR PREHEATER
FROM TEMPERATURE INDICATOR
HA
CRIT
HS
EMERGENCY SHUTDOWN
UV
VENT
INSTRUMENT AIR
R
S
UY
PALL
CRIT
LOW PRESSURE SHUTDOWN
PSLL
PCV
STRAINER
FI
FR
FAL
PAL
PI
PT
¾"
1" HEADER
TOTAL FUEL
FY
HY A
HIC
FIC
FIC
HY
HIC
" HEADER
¾" X 1" PSV
TI
" HEADER
TO SAFE LOCATION
PAHH
CRIT
HIGH PRESSURE SHUTDOWN
FR
PDT
PDI
PDAH
PDAL
PDSLL
PT
PAL
PI
PDY
PIC
PAH
PAL
PT
PI
PV
PY
FC
FI
FE
FT
PI MUST BE READABLE FROM BYPASS
NORMALLY NO FLOW (FOR START-UP)
PI
PDY
PDV
PDIC
PDALL
CRIT
LOW PRESSURE SHUTDOWN
PROVIDE LIMIT STOP
ATOMIZING STEAM
UV
UY
R
S
VENT
INSTRUMENT AIR
I"P
INS
MIN
END METER RUN OF PIPE FROM SUPPLY HEADER TO HEATER
FUEL OIL RETURN HEADER
FUEL OIL SUPPLY HEADER
LOW PRESSURE SHUTDOWN
PSLL
CRIT
PALL
UV
UY
R
S
VENT
INSTRUMENT AIR
LOCATE REMOTE FROM HEATER (OPERATED FROM GRADE)
PI
PAH
PAL
PT
PI
PV
PY
FC
FE
FT
FI
SEE STD DWG 8-138 TYPE II
PI MUST BE READABLE FROM BYPASS
FE
FT
MASS FLOW
FY A
FY B
FI A
FI B
FR A
FR B
SWITCH
STRAINER
AY
AI
DENSITY
AT
FIF
TO RELIEF HEADER
" FUEL GAS
¾"
FROM AIR PREHEAT SHUTDOWN
FROM LOW FLOW SWITCHES
TO DAMPER
A1A1
A2A1
40°F
LO
UNC

THIS P AND I DIAGRAM IS DIAGRAMMATIC ONLY. DESIGN OF PIPE LINES MUST BE INVESTIGATED FOR VENTING OF GAS AND VAPOR POCKETS IN PIPING AND EQUIPMENT, LOW POINTS IN PIPING, PUMPS AND EQUIPMENT FOR FREEZING AND DRAINING, AND ACCESSIBILITY OF ALL VALVES, FLANGES, INSTRUMENTS INCLUDING THERMOCOUPLES, ETC.

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REV	DFTR	CKR	APVD	DATE	PROJECT

UOP
25 East Algonquin Road • PO Box 5017 • Des Plaines, Illinois 60017-5017 • USA

PIPING AND INSTRUMENT DIAGRAM
OF
HEATER FIRING

DFTR _____ CKR _____
APVD _____ DATE _____

DWG NO	REV
1018010	2

COMBUSTION AIR PREHEATER
(BY OTHERS)
CONTRACTOR TO PROVIDE CLEARANCE AND ACCESS FOR WATER WASHING AND DRAINING OF PREHEATER

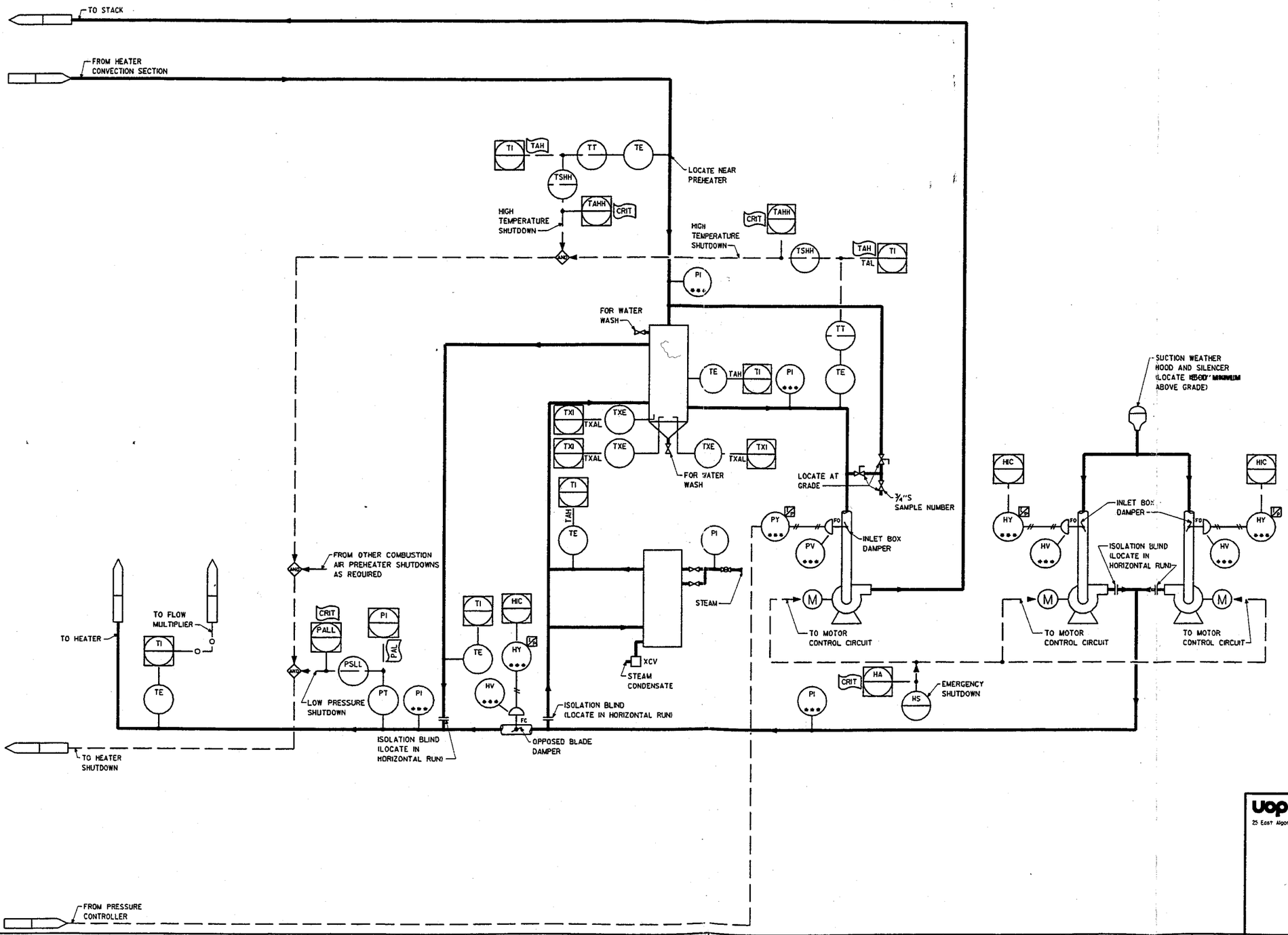
STEAM-AIR PREHEATER (BY OTHERS)

INDUCED DRAFT FAN (BY OTHERS)

COMBUSTION AIR FANS (BY OTHERS)

NOTE:
INSTRUMENTATION SHOWN IS MINIMUM REQUIRED FOR PROCESS CONTROL. CONTRACTOR SHALL VERIFY COMPLIANCE WITH APPLICABLE CODES AND STANDARDS

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PIPING AND INSTRUMENT DIAGRAM
OF
AIR PREHEAT

DFTR _____ CKR _____
APVD _____ DATE _____

DWG NO 1018015

REV 2

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TO STACK
TI
TE
ISOLATION BLIND (LOCATE IN HORIZONTAL RUN)
PV ***
PY ***
FO
DAMPER
ISOLATION BLIND (LOCATE IN HORIZONTAL RUN)
FROM INDUCED DRAFT FAN
TO COMBUSTION AIR PREHEATER
TE
TI
PY B
HIGH PRESSURE SIGNAL OPENS THIS DAMPER LAST
PIC
PAH
PT
SPLIT RANGE
PSHH
HIGH PRESSURE SIGNAL OPENS THIS DAMPER FIRST
PY A
DRAFT GAUGE (LOCATE GAUGE AND VALVES AT GRADE)
PI
DT
¾"S SAMPLE NUMBER
FOR BURNER PIPING ARRANGEMENT SEE STD DWG 2-101
HEATER SEE DWG
AT
OXYGEN
AAL
AI
AR
TE
TAH
TI
BURNER FUEL GAS PRESSURE (PORTABLE, ONE PER HEATER)
PI
PI ***
BA
BE
ONE FOR EACH BURNER
COMBUSTION AIR (PORTABLE, ONE PER HEATER)
PROVIDE PURGE STEAM TO HEATER FIREBOX
FROM CONTROLLER AT HEATER OUTLET
HIGH TEMPERATURE SIGNAL DECREASES FUEL
TY A
LOW TEMPERATURE SIGNAL INCREASES AIR
TOTAL FUEL
FIC
TY B
HY B
HIC
HEADER
PROVIDE LIMIT STOP
FV
FO
OPPOSED BLADE DAMPER
¾"
1" HEADER
PI
PT
PAL
PSLL
DT
PCV
STRAINER
LOW PRESSURE SHUTDOWN
PALL
CRIT
EMERGENCY SHUTDOWN
HS
HA
FROM COMBUSTION AIR PREHEATER
FROM TEMPERATURE INDICATOR
FE
FT
FAL
VENTURI
FI
FR
UY
VENT
INSTRUMENT AIR
PAHH
CRIT
HIGH PRESSURE SHUTDOWN
" HEADER
¾" x 1" PSV
TO SAFE LOCATION
TI
HIC
HY
FIC
FR
" HEADER
PDT
PDI
PDAH
PDAL
PDSLL
PDY
PT
PAL
PI
PIC
PAH
PAL
PT
PI
PV
PY
FC
E
PI MUST BE READABLE FROM BYPASS
FE
FT
FI
FY
NORMALLY NO FLOW (FOR START-UP)
UV A
UV B
UY
R
S
VENT
INSTRUMENT AIR
I"P
MIN
INS
RUN OF PIPE FROM SUPPLY HEADER TO HEATER
FUEL OIL SUPPLY HEADER
FUEL OIL RETURN HEADER
PI
PDY
PDIC
PDV
PDALL
CRIT
LOW PRESSURE SHUTDOWN
PROVIDE LIMIT STOP
ATOMIZING STEAM
FROM LOW FLOW SWITCHES
AND
SEE STD DWG 8-138 TYPE II
PI MUST BE READABLE FROM BYPASS
PSLL
CRIT
PALL
UV
UY
R
S
VENT
INSTRUMENT AIR
LOCATE REMOTE FROM HEATER (OPERATED FROM GRADE)
" FUEL GAS
AY
AI
DENSITY
AT
FIF
TO RELIEF HEADER
FI A
FR A
FI B
FR B
STRAINER
SWITCH
MASS FLOW
FY A
FY B
TOTAL FUEL
FY
Σ
FROM AIR PREHEAT SHUTDOWN
TO DAMPER

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PIPING AND INSTRUMENT DIAGRAM
OF
HEATER FIRING

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REV	DFTR	CKR	APVD	DATE

PROJECT

DFTR ____ CKR ____
APVD ____ DATE ____

DWG NO 1018010
REV 2

COMBUSTION AIR PREHEATER (BY OTHERS) CONTRACTOR TO PROVIDE CLEARANCE AND ACCESS FOR WATER WASHING AND DRAINING OF PREHEATER

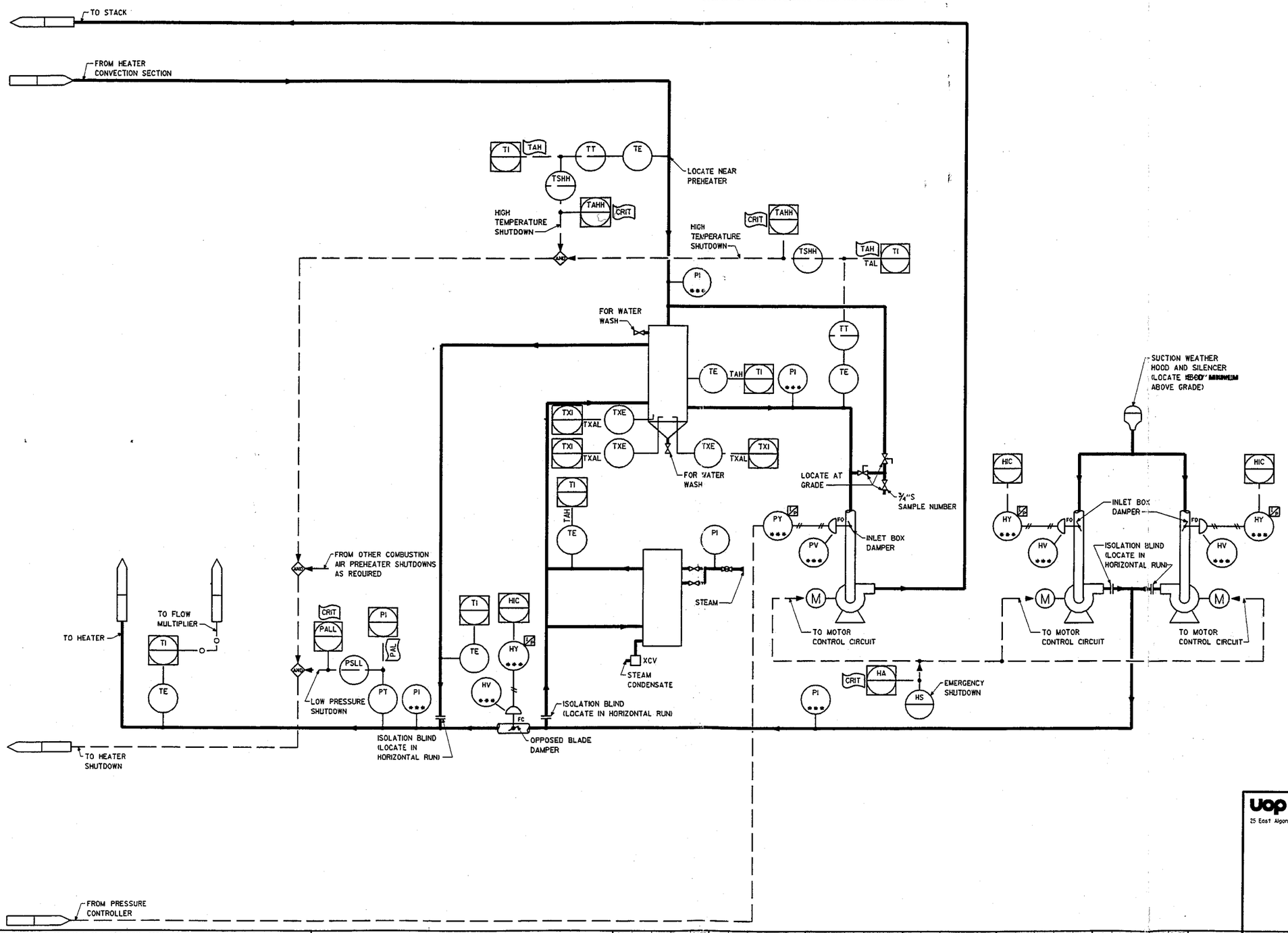
STEAM-AIR PREHEATER (BY OTHERS)

INDUCED DRAFT FAN (BY OTHERS)

COMBUSTION AIR FANS (BY OTHERS)

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REV	DFTR	CKR	APVD	DATE	PROJECT

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PIPING AND INSTRUMENT DIAGRAM
OF
AIR PREHEAT

DFTR _____ CKR _____
APVD _____ DATE _____

DWG NO 1018015

REV 2

Training Services

Troubleshooting

Field Evaluation of Heaters

FH 2004/10-1

Section Outline

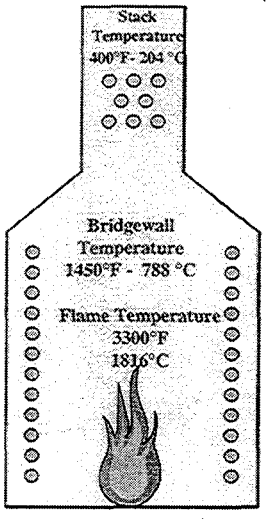
- Process Considerations
- Data Collection
- Visual Observations - Combustion
- Visual Observations - Firebox
- Process Heat Balance
- Flue Gas Heat Balance
- Radiant Flux Rate
- Bridgewall Temperature Evaluation
- Efficiency

FH 2004/10-2

Data Collection- Control Room

- BWT- (Radiant Flux Rate)
- Stack Temperature-(Cold End Approach, Efficiency)
- Flue Gas O2- (Firing Quality, 2% minimum)
- Burner Pressure-(High Pressure=High Firing Rate)
- Process Flow Rate-(Over Design=High Duty)
- Process Delta Temperature-(Process Duty, Flux Rate)

Typical Furnace Temperatures



- Flame Temperature
  - Depends on Fuel Composition
- Radiant exit or Bridgewall Temperature
- Depends on Tube Temperatures and Heat Flux

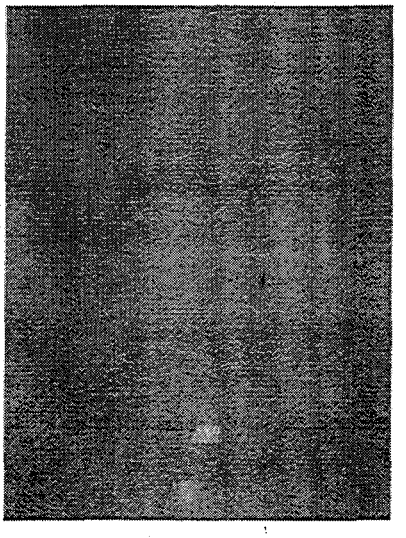
Bridgewall Temperature Evaluation

- How hard is the heater operating?
  - 1400-1500°F (760-815°C) is normal
  - 1600-1650°F (871-900°C) is approaching maximum limits
- Fouling of tubes
  - Every 100° fouling raises the BWT 50°
- It is rare to find heater problems with BWTs below 1300°F (705°C)

FH 2004/10-5

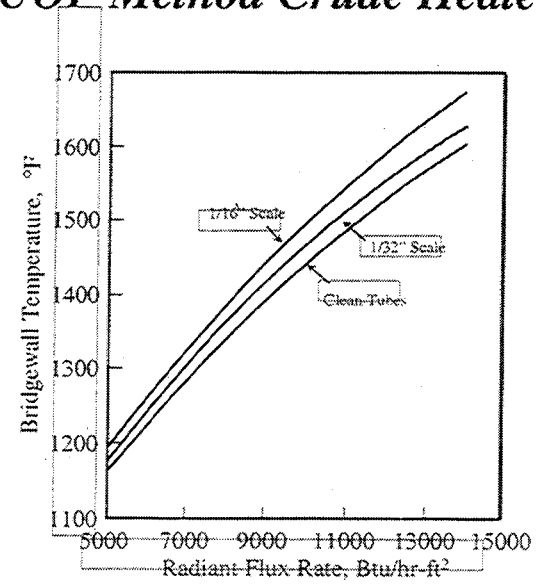
Low BWT

- Dark Box
- BWT below 1300°F (705°C)
- Low Firing Rate
- Rare to find problems



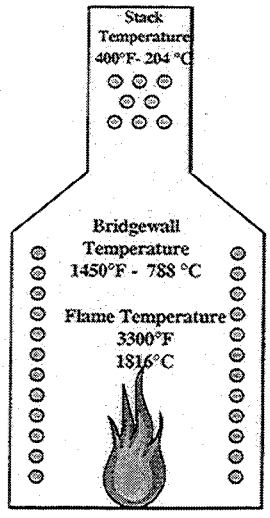
FH 2004/10-6

UOP Method Crude Heater



FH 2004/10-7

Typical Furnace Temperatures



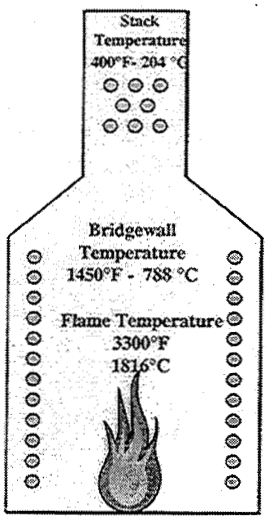
- Use the UOP fuels program to determine radiant-convection split
- Use the fuel meter to determine the heat release and radiant flux rate
- Compare actual BWT to predicted from the flux chart

FH 2004/10-8

Bridgewall Temperature Evaluation

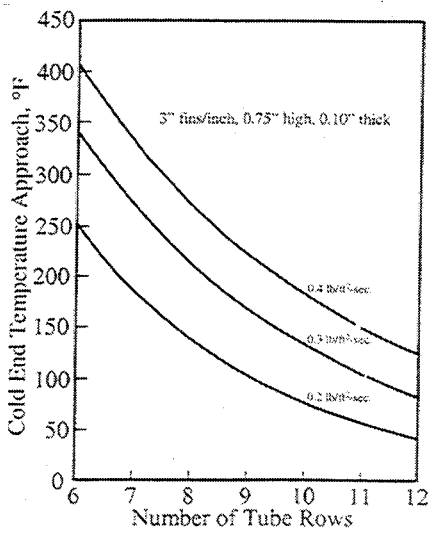
- If actual BWT higher than predicted?
  - Fouling of tubes (insideor outside)?
  - Flame height is too high?
  - Short Firebox Height?
  - Combustion Problems?
- If actual BWT lower than predicted?
  - Heater is performing very well

Typical Furnace Temperatures



- Use Cold End Approach=(Flue Gas Out-Process Inlet)
- Use convection charts to determine predicted cold end approach

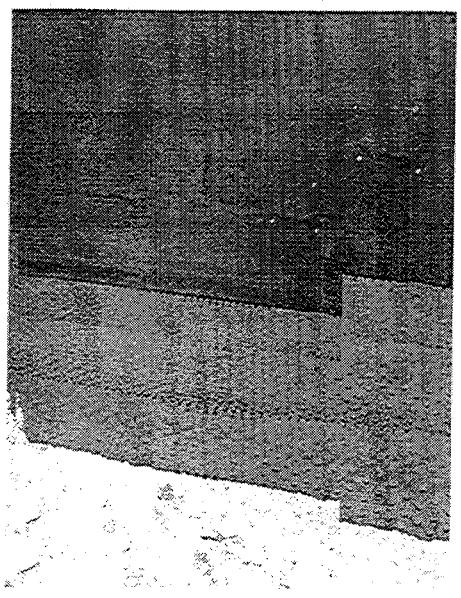
Cold End Approach
450 °F Process Inlet - Fuel Oil - NPS 6



FH 2004/10-11

Convection Tubes

- Sootblowing Lane-Heater is in operation with gas temperature at 800°F (427°C)
- Light brown material is ash
- Ash reduces heat transfer and results in high cold end approach
- Grey tube areas are clean
- Sootblowers should be operated once or twice a day



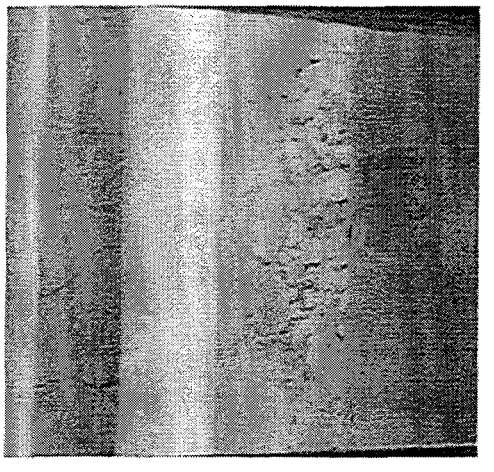
FH 2004/10-12

High BWT

- High Radiant Flux Rates
- High TWTs
- High Corrosion Rates

High BWT

- Refractory Damage
- FCC Catalyst Fines Cover Tubes
- TWT is 200°F Hotter
- Higher Firing Rate
- Bright color indicates hot firebox



Visual Observations - Firebox

- Coloring of Refractory and Tube Supports
- Appearance of Tubes
  - Bowing
  - Scale
  - Hot Spots
- Appearance of Refractory
- Smoke from Stack

FH 2004/10-15

Color in Relation to Temperature of Steel

Temperature		Color
990°F	(532°C)	Black red, dark blood red
1050	(566)	Dark red, blood red, low red
1175	(635)	Dark cherry red
1375	(750)	Cherry red
1550	(850)	Light cherry, light red
1650	(900)	Orange
1725	(950)	Light orange
1825	(1000)	Yellow
1975	(1080)	Light yellow
2200	(1200)	White

FH 2004/10-16

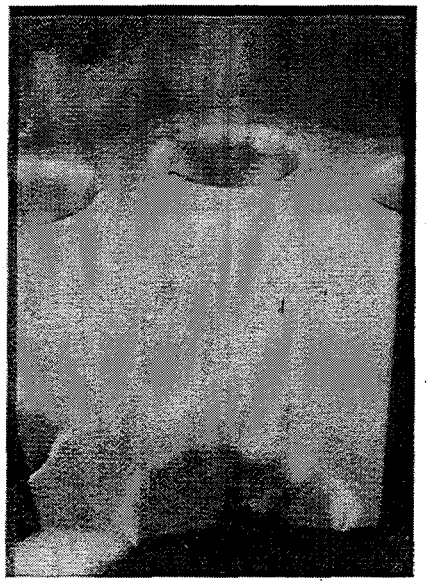
Visual Observations - Combustion

- Color of Flame
- Evidence of Incomplete Combustion
- Flame Impingement
- Flame Shape and Stability
- Burner Air Register Settings
- Fuel Pressure
- Draft

FH 2004/10-17

Color of Refractory

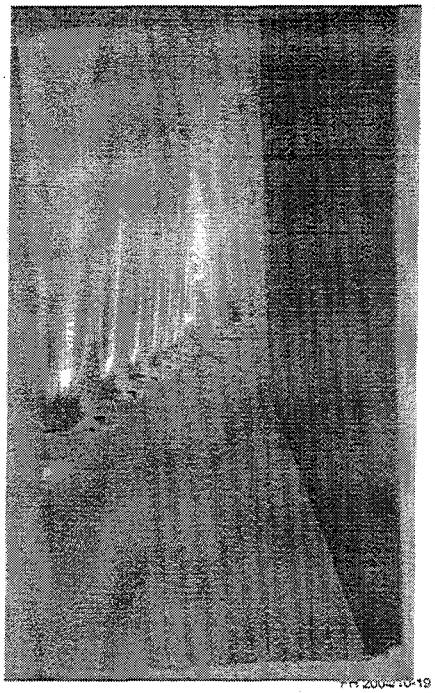
- Vertical Cylindrical Floor
- Bright Refractory Color
- Heater is Overfired
- Floor is normally 1200°-1600°F (650°C-870°C)
- Actual Floor is 2000°F (1100°C)
- High Maintenance



FH 2004/10-18

Color of Refractory

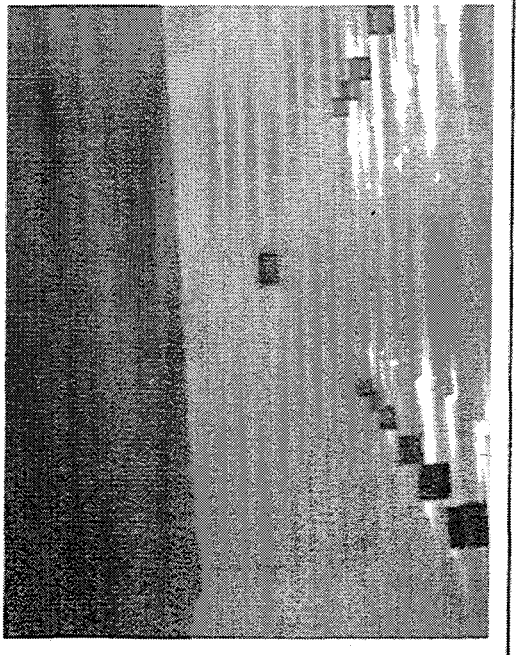
- Hydrogen Reformer (Terrace Wall)
- Bright Refractory Color
- Tube is 1697°F(925°C)
- Floor is 1790°F (977°C)
- Sidewall is 1840°F (1005°C)
- One of the hottest heaters



FH 2004/10-19

Color of Refractory

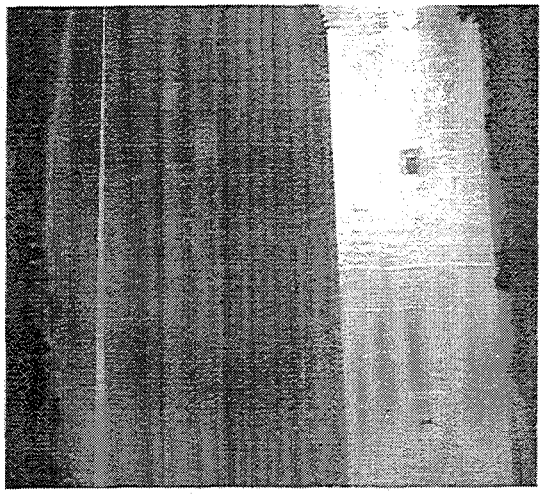
- Hydrogen Reformer
  (Side Fired)
- Bright Refractory Color
- Tube is 1770°F(966°C)
- Floor is 1890°F (1032°C)
- Sidewall is 1909°F
  (1043°C)
- One of the hottest heaters



FH 2004/10-20

Color of Tubes

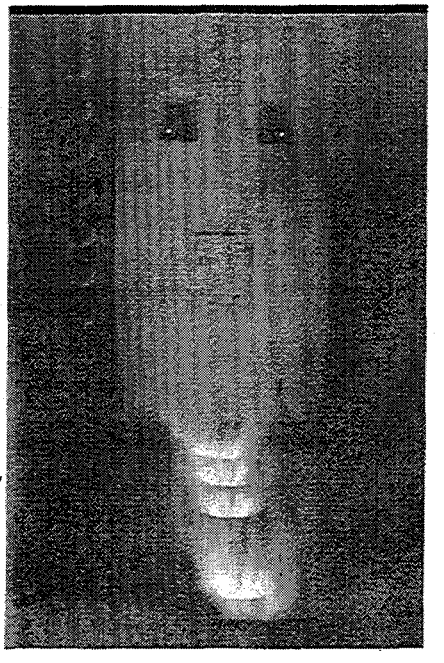
- Hydrogen Reformer (DownFired)
- Hot Spots on Tube 1706-1742°F(930-950°C)
- Coke due to naphtha feed with poor steam/HC control



FH 2004/10-21

Troubleshooting

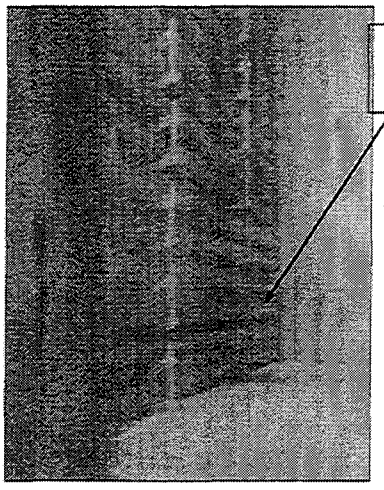
- Tube Support Temperature indicates Gas temperature and Flux rate at that point
- Bright Refractory Color indicates low excess air(2% O2)
- High excess air would have stiff blue flames and no refractory color



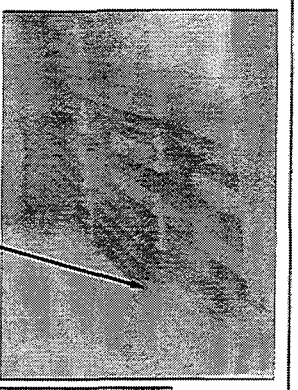
FH 2004/10-22

Color of Tube Supports

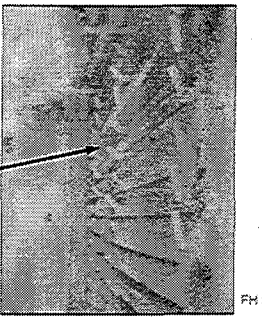
- Look for hot spots and heat maldistribution on tube supports



Colder tubes near the floor



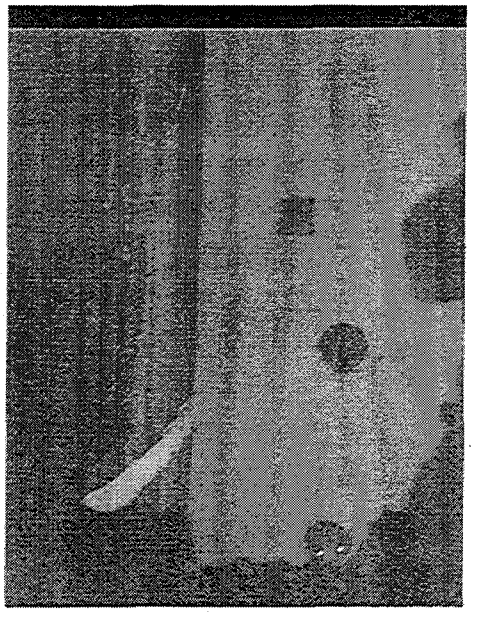
Hot tubes in the middle



FH 2004/10-23

On-Line Inspection

- Tube Guide Temperature indicates Gas temperature and Flux rate at that point
- Colder Color at wall on guide



FH 2004/10-24

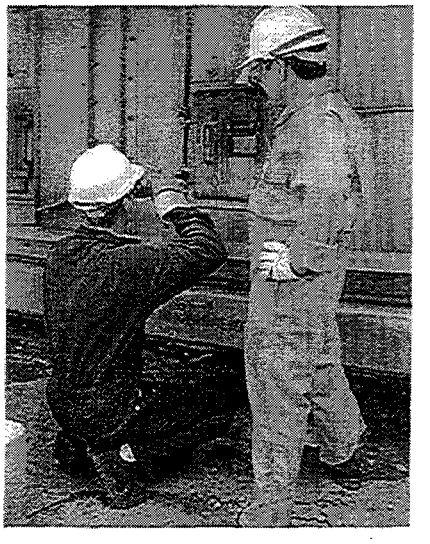
Portable Instruments

- Infrared Pyrometer-(Minolta-Land Model 339)
- Portable O_2 Analyzer(O_2 and CO)
- UOP uses Testo Model 342-3 (O_2 ,NO_x, CO, Temperature)
- Temperature Probe

FH 2004/10-25

Infrared Pyrometer

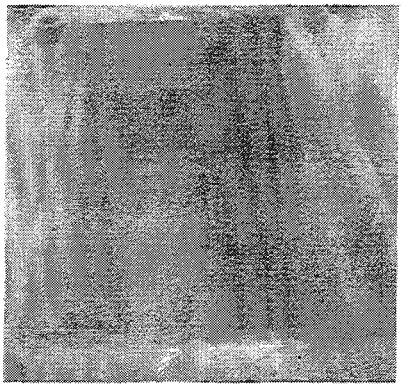
- Minolta-Land Model 339 (Furnace Pro)
- Obtain Tube Surface Temperatures
- Obtain Firebox Temperatures



FH 2004/10-26

Infrared Pyrometer

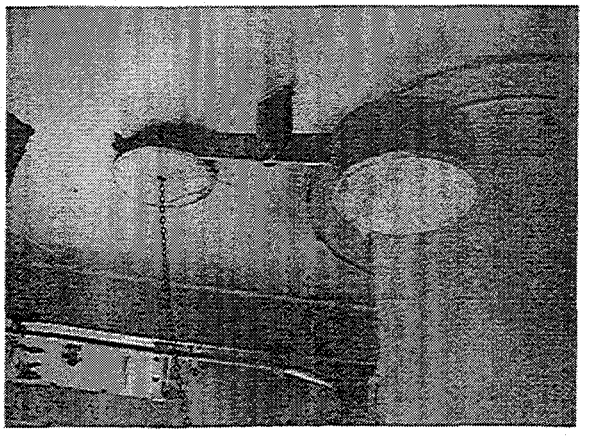
- Shock Tube Support Temperature indicates BWT (flue gas exit)
- Obtain Radiant Flux Rate
- Shot from side view port on Vertical Cylindrical Crude Heater



FH 2004/10-27

Floor Viewports

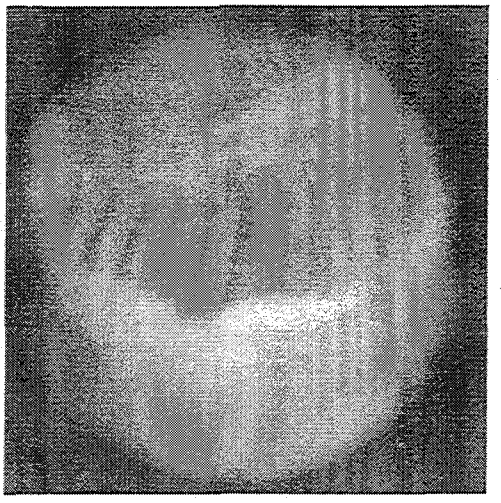
- Floor Observation (6") recommended on vertical tube heaters
- Needed to view all tube surfaces



FH 2004/10-28

Infrared Pyrometer

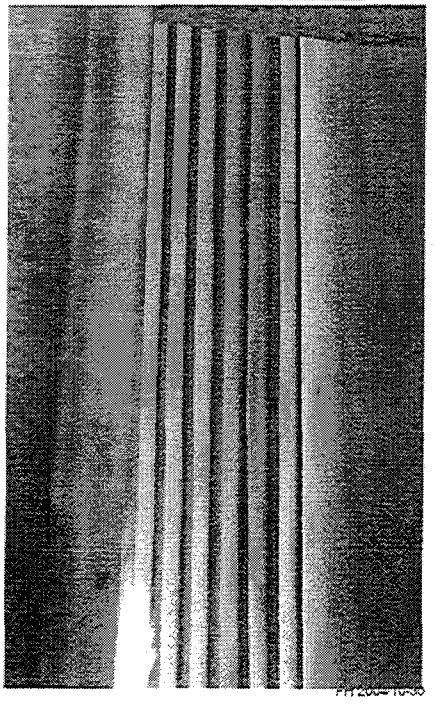
- Floor Observation (3"-75mm) Opening on Vertical Cylindrical Crude Heater
- Shoot roof refractory temperature, if no tube support available
- Obtain BWT and Flux Rate



FH 2004/10-29

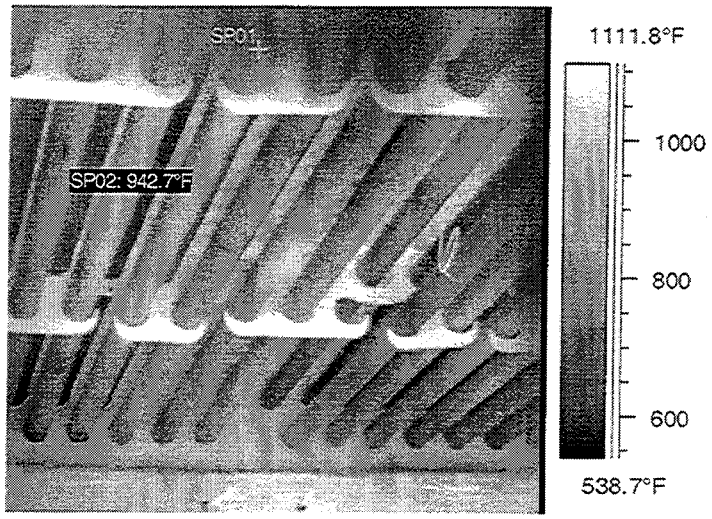
On-Line Inspection

- Oil fired Crude Heater tubes
- White tubes covered with ash were 1000°F
- Dark tubes were 1240°F with 7-10 mm coke
- Very dark front face is where oil ash has melted due to high temperature
- Dark tubes are bowed



Infrared Scan to Detect Hot Spot

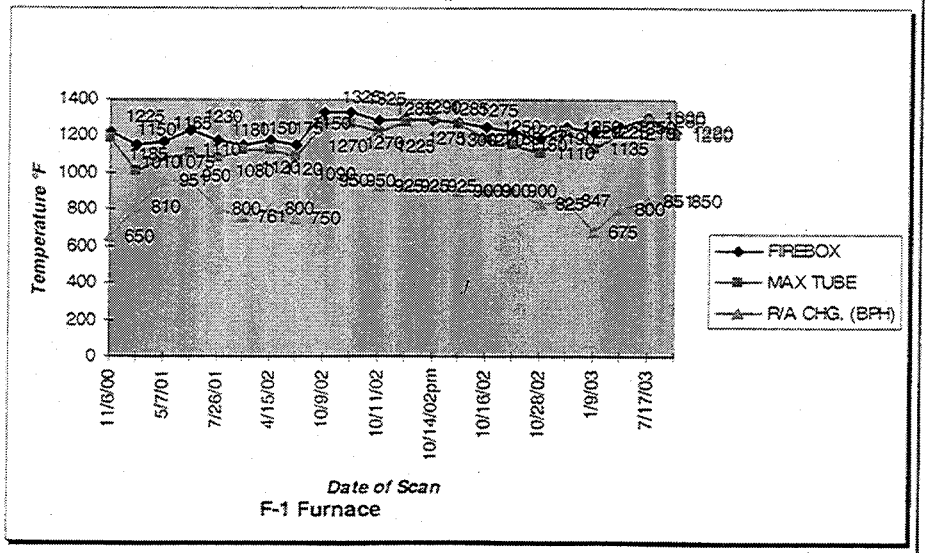
- **Monitor at least every 3 month**



FH 2004/10-31

Infrared Scans

Plot TWT , firebox temperature, feedrate



F-1 Furnace

FH 2004/10-32

On-line Inspection

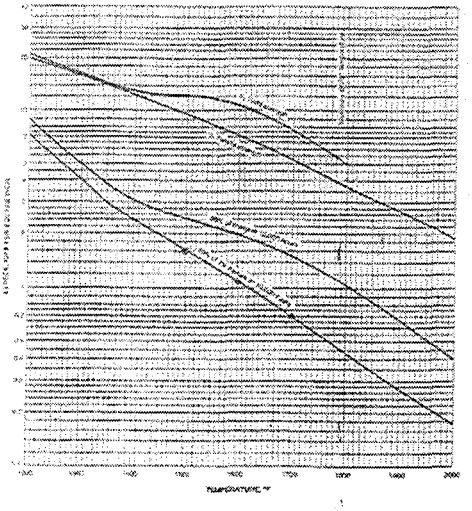
- Flames Are Too Long
- Flame Height Should be 50% of Firebox Height
- High BWT
- Hot Roof Supports
- Poor Revamp



FH 2004/10-33

Tube Supports

- API 560
- Increasing temp. From 1500F to 1600F decreases strength 31%
- Increasing temp. From 1600F to 1700F decreases strength 41%
- Older heaters usually designed for 1600F



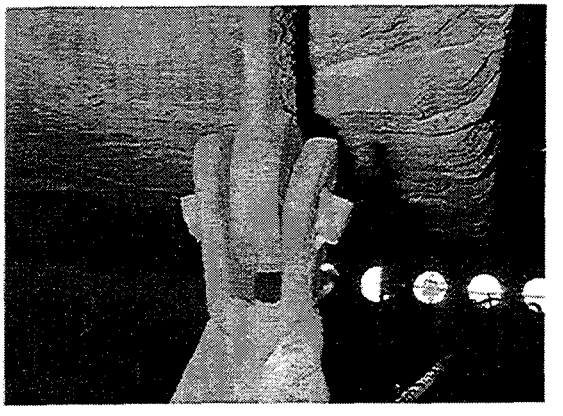
FH 2004/10-34

Damaged Shock Tube Support

Severe creep and yielding of pin and support

Small web thickness 5/8" (13 mm) Need 19-25mm web

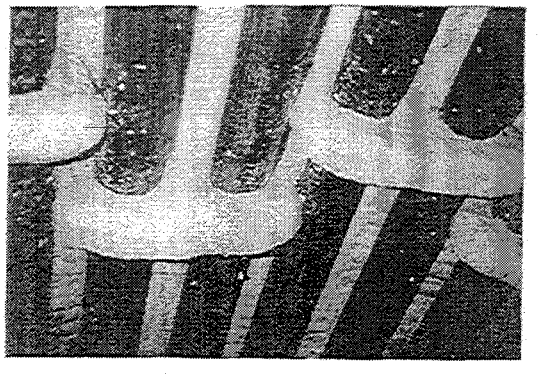
Sigma phase embrittlement of 25Cr-12Ni Material



FH 2004/10-35

Tube Supports

- Unifrax Fiberfrax LDS Moldable Ceramic Fiber
- Air Hardens
- Protects Tube Supports
- Good for Oil Firing Protection
- Sacrificial Coating
- Phone 716-278-3800
- Fax 716-278-3900



FH 2004/10-36

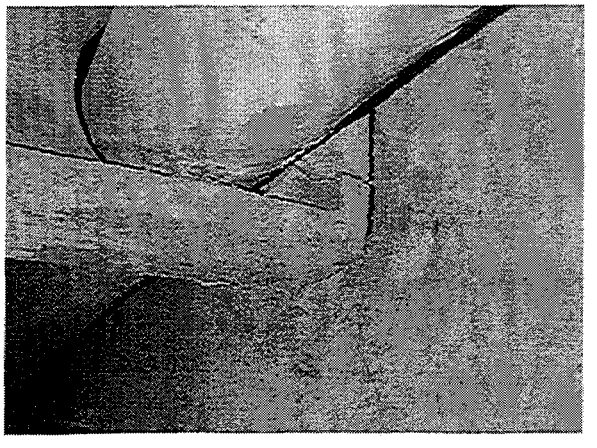
Tube Support Inspection

- Broken Supports
- Bowing or Yielding
- Cracks
- Oxidation
- Corrosion
- Replication Test (normally not performed)

Broken Tubesheet

Crack in tubesheet

Is support binding on refractory ?



Combustion Monitoring

- Flue Gas O_2 >2-4% on Gas Firing
- Operating Excess Air = 12%-26%
- New Unit Design = 15%

- Flue Gas O_2 >4-6% on Oil Firing
- Operating Excess Air = 26%-44%
- New Unit Design = 25%

Combustion Monitoring

- Low O_2 = Bad Combustion
- Afterburning
- Flame Impingment
- Tube Failures
- Refractory Damage
- Tube Support Failures

Burner Problems

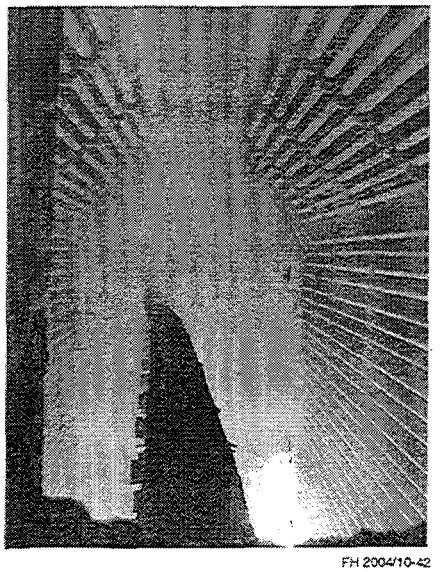
- Poor Oil Flame
- Burners Need Air
- Oil Haze impinging on roof tubes
- Unburned Fuel burning in convection section
- Burned off extended surface



FH 2004/10-41

On-Line Inspection

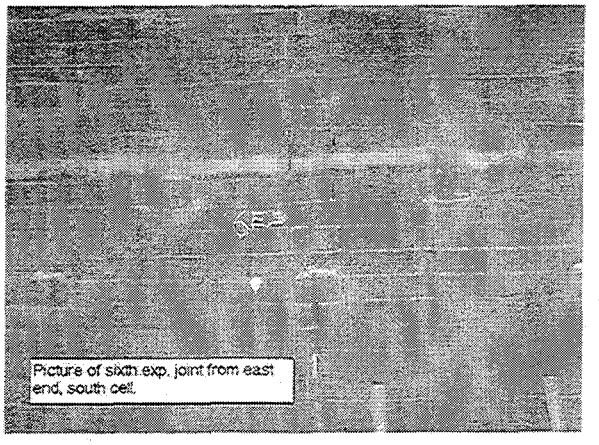
- Brickwall- Improper interlaced expansion joint
- Brickwall- expansion joint gaps are too small
- High Maintenance problems



FH 2004/10-42

On-Line Inspection

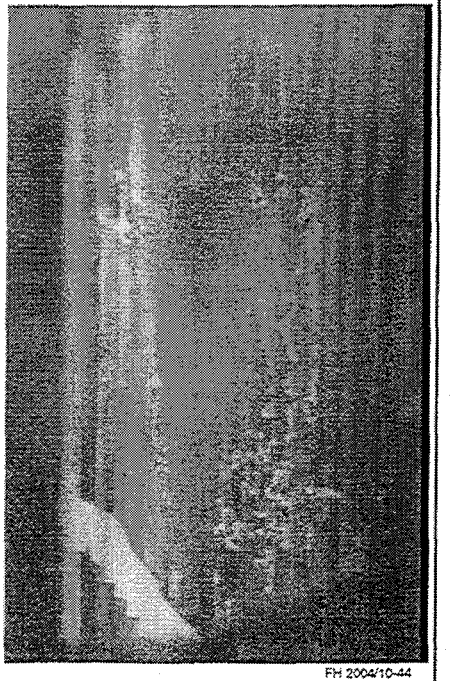
- Brickwall- Improper interlaced expansion joint
- Marked for removal at 10° Lean



FH 2004/10-43

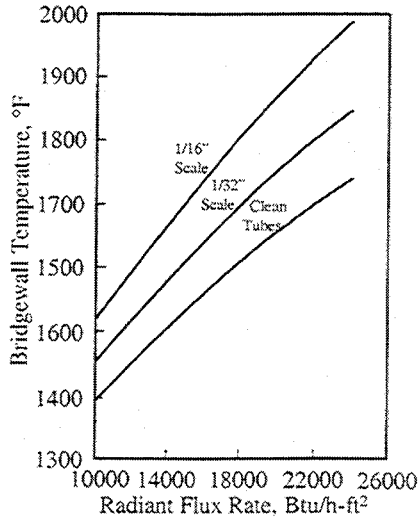
Troubleshooting

- Platforming Heater- 9 Cr tubes
- Heavy Scale
- Increased fuel firing
- Yellow scale 1750°F
- Average surface 1350°F



FH 2004/10-44

UOP Method Platforming Double Fired U Tube Heater



FH 2004/10-45

Troubleshooting

- Use walnut shells to blast loose scale off tubes on line
- Reduced fuel firing
- Only recommended for loose scale
- Cetek does this service



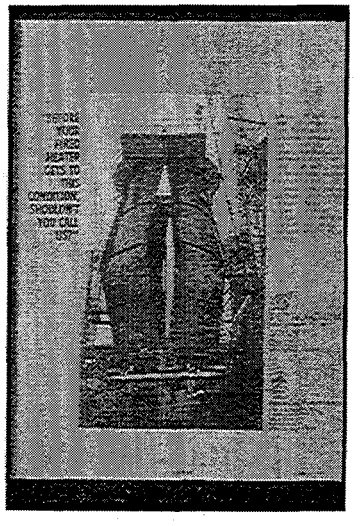
FH 2004/10-46

Establish Work Scope of Turnaround

- On-line Inspection
- Operational Data
- Past Inspection Reports
- Code or Jurisdictional Requirements
- Projected Service

Troubleshooting

- Most Heater Accidents Occur During Lightoff
- 15-20 Minute Purge
- Test for Combustibles?
- Good Operating Instructions
- Burner Management Systems



Training Services

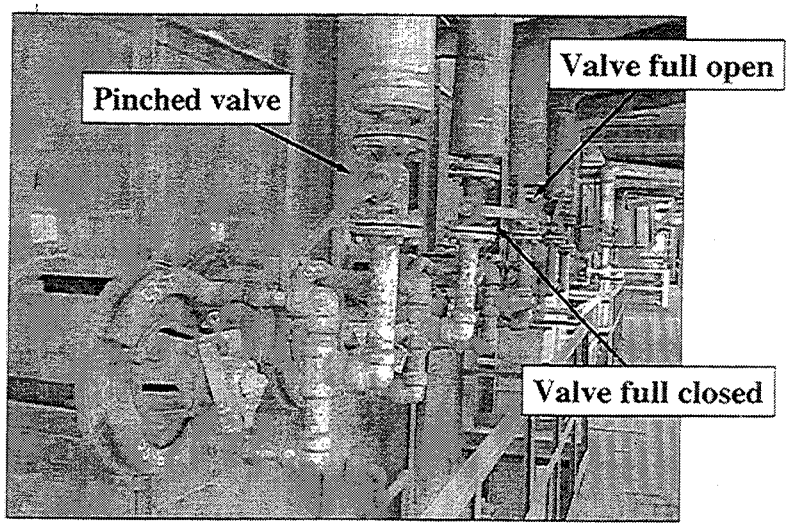
Troubleshooting Combustion Problems Part 1

Visual Observations - Combustion

- Flame Impingement
- Flame Shape and Stability
  - Same height and shape
  - Same fuel and air to each burner
- Color of Flame
- Evidence of Incomplete Combustion
  - Unburned fuel haze
- Fuel Pressure
- Draft
  - Limited turndown on natural draft burners

Burner Problems

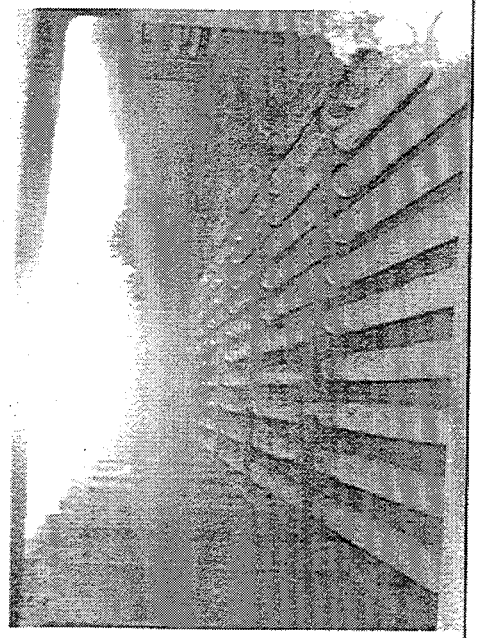
Fuel gas valves should not be pinched



FH 2004/10-51

On-line Inspection

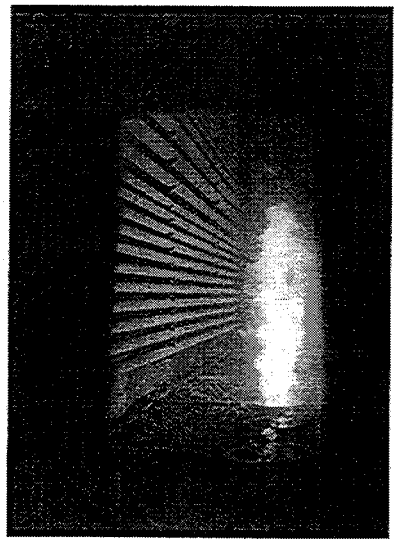
- Flames Are Too Long, impinging on roof tubes
- Flame Height Should be 50% of Firebox Height
- High BWT
- Hot Roof Supports
- Poor Revamp



FH 2004/10-52

On-Line Inspection

- Flames Are Proper Height
- Flame Height Should be 50% of Firebox Height
- Good Heater Design



FH 2004/10-53

Combustion Monitoring

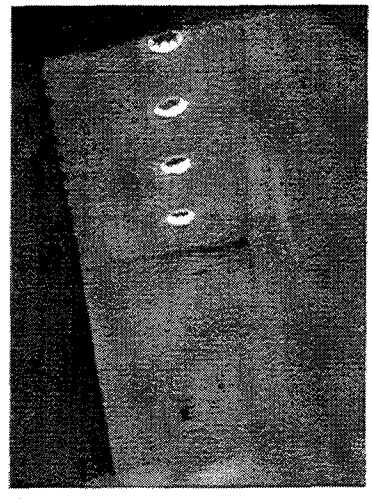
- Flue Gas O_2 >2-4% on Gas Firing
- Operating Excess Air = 12%-26%
- New Unit Design = 15%

- Flue Gas O_2 >4-6% on Oil Firing
- Operating Excess Air = 26%-44%
- New Unit Design = 25%

FH 2004/10-54

Combustion Monitoring

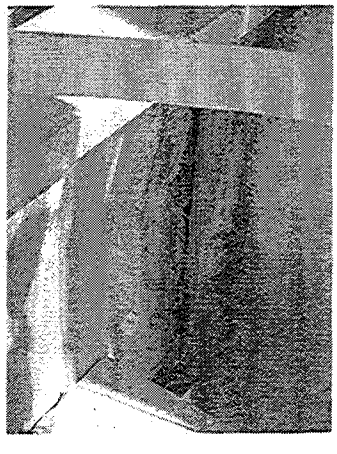
- Low O2 operations require tight fireboxes that do not leak air
- Dark corner indicates large air leak
- Seal firebox with high temperature RTV silicone and aluminum tape
- Good tube seals on tube inlets and outlets

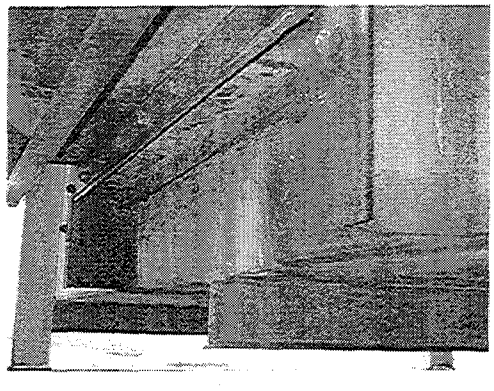


FH 2004/10-55

Firebox Sealing

- Low O2 operations require tight fireboxes that do not leak air
- Glue metal over firebox holes with high temperature silicone (and self tapping metal screws





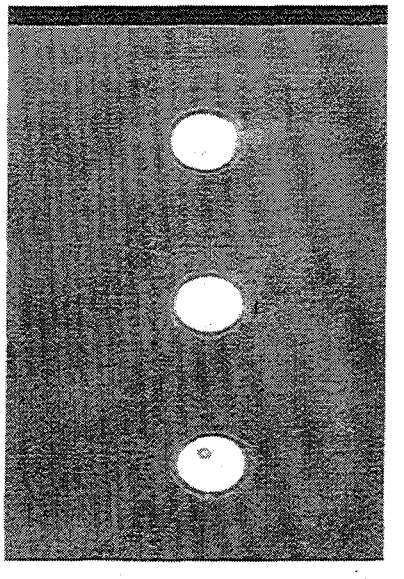
FH 2004/10-56

Combustion Monitoring

- Low O_2 = Bad Combustion
- Afterburning
- Flame Impingment
- Tube Failures
- Refractory Damage
- Tube Support Failures

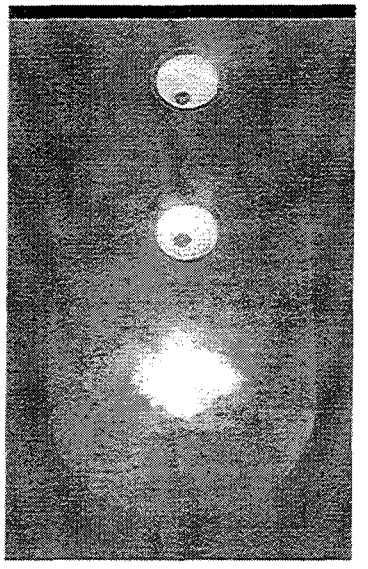
Good Combustion

- Uniform Flame Pattern
- No Haze



Poor Combustion

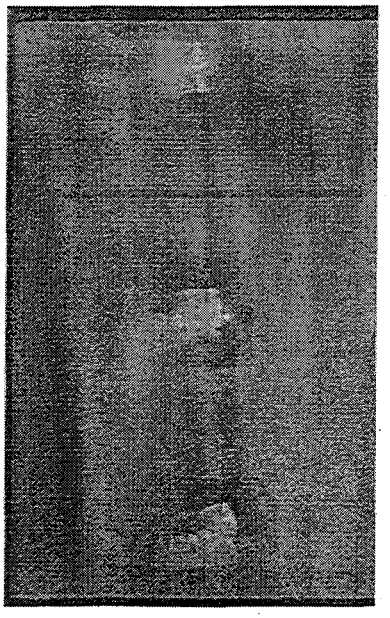
- Non-Uniform Flame Pattern
- Haze-Unburned Fuel
- Bottom Burner Needs Air
- Potential Impinging on Tubes



FH 2004/10-59

Good Combustion

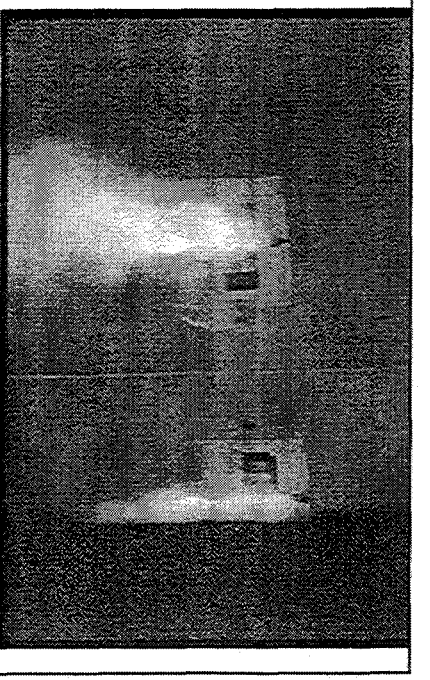
- Uniform Flame Pattern
- Blue Hydrogen Flame



FH 2004/10-60

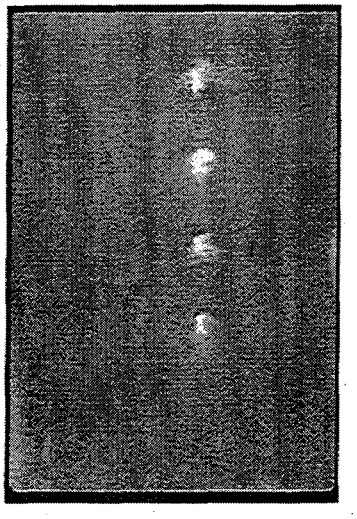
Poor Combustion

- Fuel Composition Change
- Haze-Unburned Fuel
- Burners Need Air



Good Combustion

- Uniform Color
- Same Flame Shape



Burner Problems

- Top Burner- High Air (Secondary Fuel Out)
- 2nd Burner- Good Combustion
- 3rd Burner- Orange Color on Refractory (Secondary Tip Rotated 180°)
- 4th Burner- Haze- Needs Air



FH 2004/10-63

Burner Problems

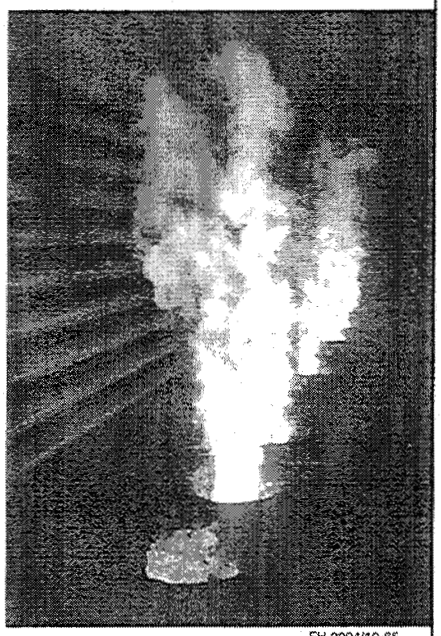
- Coke growth on Secondary Gas Tip from liquid entrainment



FH 2004/10-64

Burner Problems

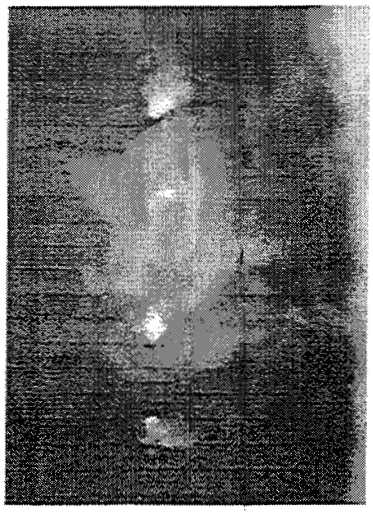
- Plugged gas tips on first burner
- All Flames should be the same height and shape
- Consider fuel gas coalescer and steam heater to reduce tip plugging problems



FH 2004/10-65

Burner Problems

- Top Burner- starting to run out of air
- 2nd Burner- Haze- Needs Air
- 3rd Burner- Haze- Needs Air
- (Secondary Tip Rotated 180°)
- 4th Burner- Secondary Tips Out of Service



FH 2004/10-66

Burner Problems

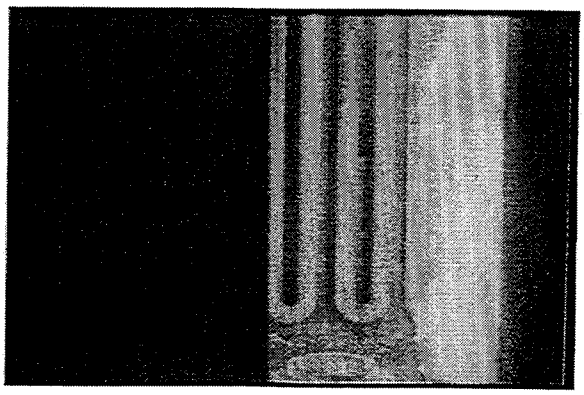
- Unburned Fuel Haze
- Burners Need Air
- Gas Haze Similar to Oil Haze



FH 2004/10-67

Good Oil Combustion

- Bright, well defined flame
- No haze



FH 2004/10-68

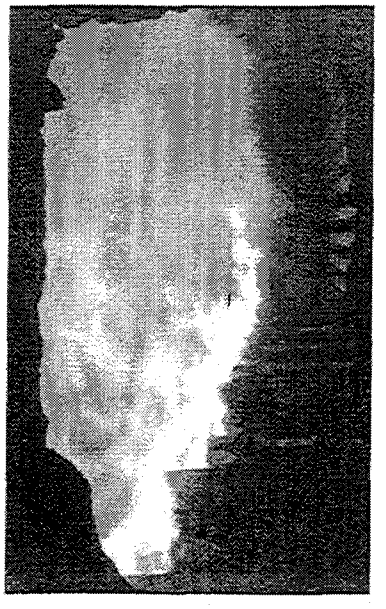
Burner Problems

- Poor Oil Flame
- Burners Need Air
- Oil Haze Similar to Gas Haze
- Unburned Fuel at end of flame



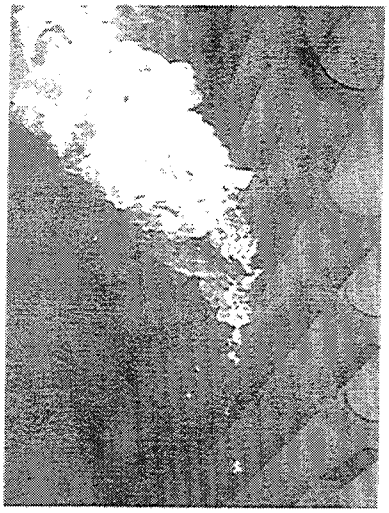
Burner Problems

- Poor Oil Flame
- Burners Need Air
- Oil Haze Similar to Gas Haze
- Unburned Fuel at end of flame



On-Line Inspection

- Leaking Tube
- Most Failures Caused by Flame Impingement
- Control O2 Level high enough to keep flames off the tubes



FH 2004/10-71

Burner Problems

- Poor Oil Flame
- Burners Need Air
- Oil Haze impinging on roof tubes
- Unburned Fuel burning in convection section
- Burned off extended surface



FH 2004/10-72

Troubleshooting

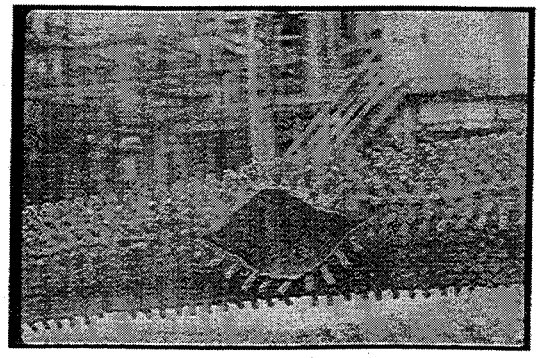
- **Operating Instructions at Heater**



FH 2004/10-73

Burner Problems

- **Flame Impingement caused failure on convection tube**
- **Classic "Fish Mouth" failure**



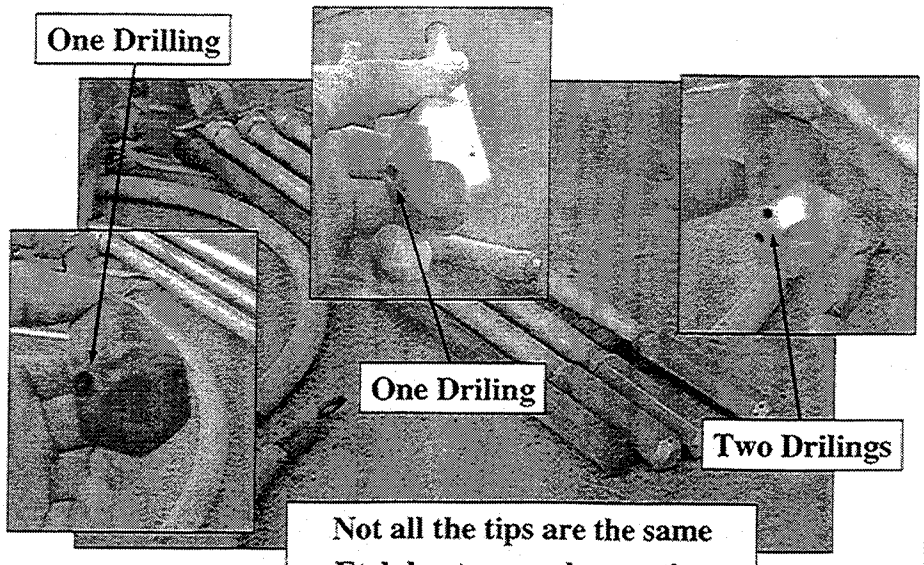
FH 2004/10-74

Burner Inspection

- Burner Tile
  - Properly Centered
  - Damaged?
- Burner Tips
  - Proper Orientation
  - Are tips plugged?
- Air Register Control
  - Does it work?
  - Close for burners out of service

FH 2004/10-75

Burner Tips

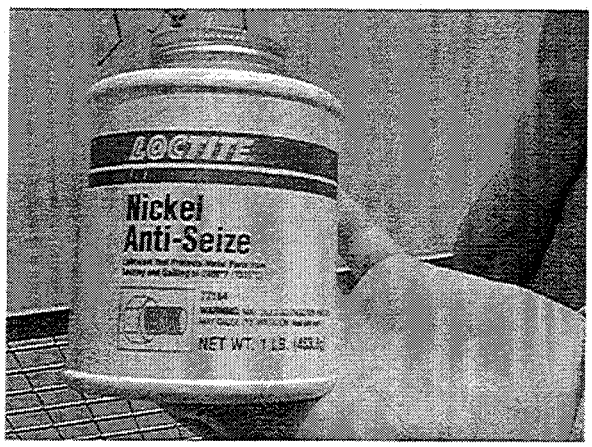


Not all the tips are the same
Etch heater number on tips

FH 2004/10-76

Burner Maintenance

Install tips with high temperature anti-seize



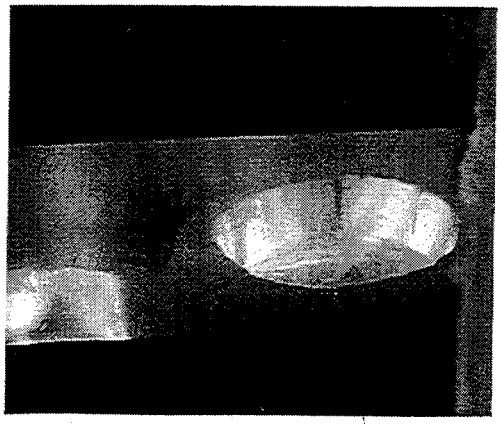
FH 2004/10-77

Burner Tile Problem

Poor Design-Oversized Tiles-40% more air flow-no air control

Low draft design-poor flame shape

Tiles need to be mortered together



FH 2004/10-78

Burner Setup Problems

Incorrect tile dimensions

Tips are wrong height

Tips are different heights

Tip shown is at least 25mm too high

Tip to the right is not aligned



FH 2004/10-79

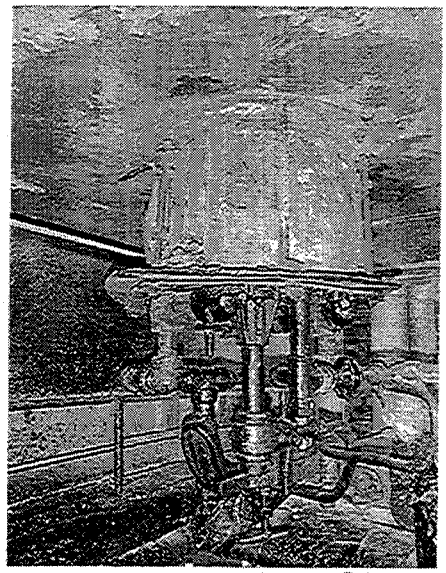
Burner Problems

- Burners are in bad condition
- New Burners are recommended
- Oil and steam should have flexible hose connections
- Hard piping have oil gun at wrong height

FH 2004/10-80

Burner Problems

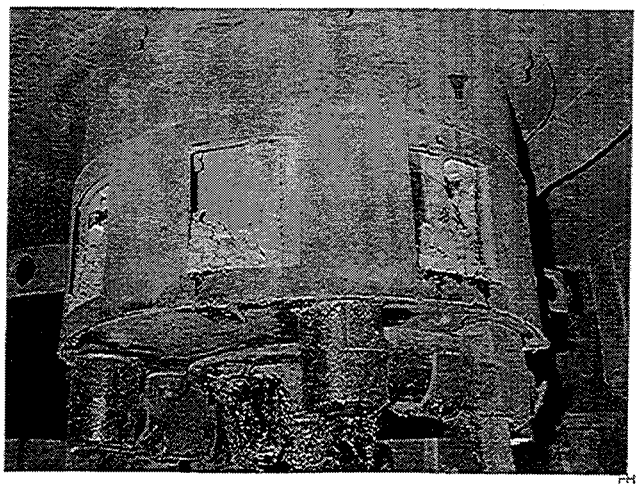
- Burners are in bad condition
- New Burners are recommended
- Oil and steam should have flexible hose connections
- Hard piping have oil gun at wrong height



FH 2004/10-81

Burner Problem

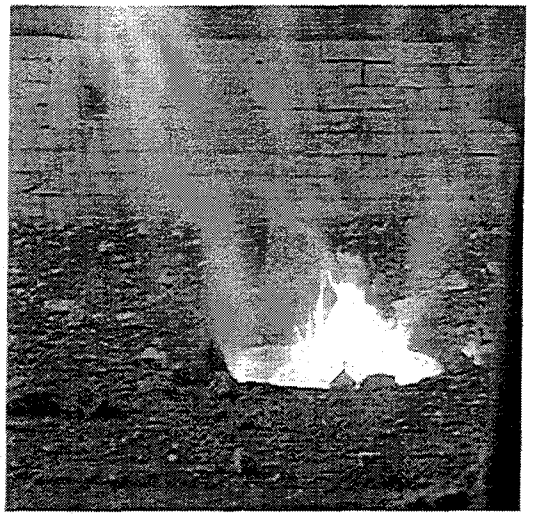
- Air register plugged with refractory
- Burners account for 5-10% of heater cost, but account for most heater problems



FH 2004/10-82

Burner Problems

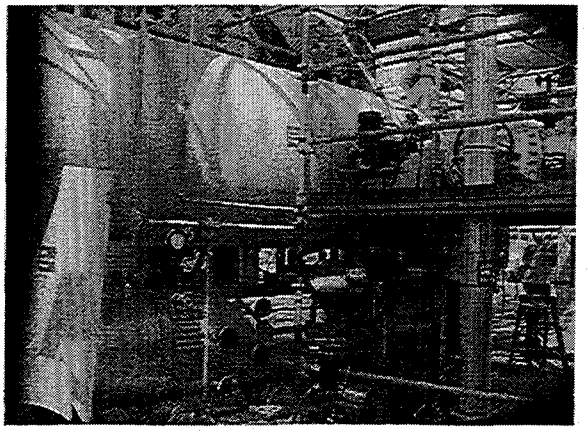
- Coke growth on Secondary Gas Tip from liquid entrainment



FH 2004/10-83

Fuel Gas Coalescer

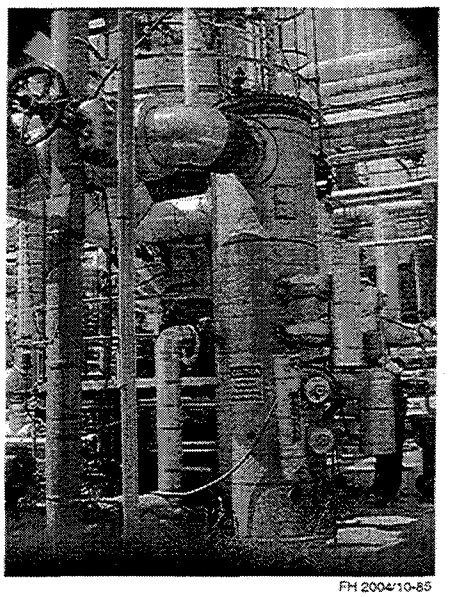
- Required for Low NOx burners
- King Tool –self cleaning-good,but expensive
- PECO-Gemini Peach- good, cost effective



FH 2004/10-84

Vertical In-tube Fuel Gas Coalescer

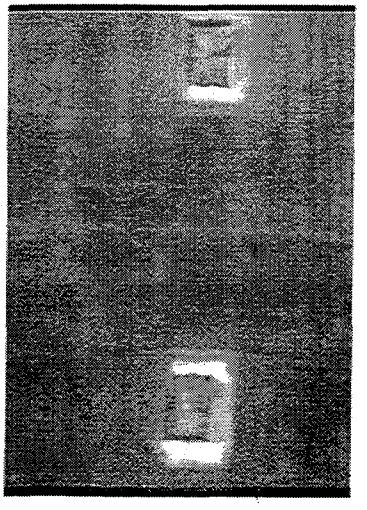
- Low particulate capability
- Require prefilter to remove particulate
- Not recommended
- Supplied by Pall and Peco



FH 2004/10-85

Burner Problems

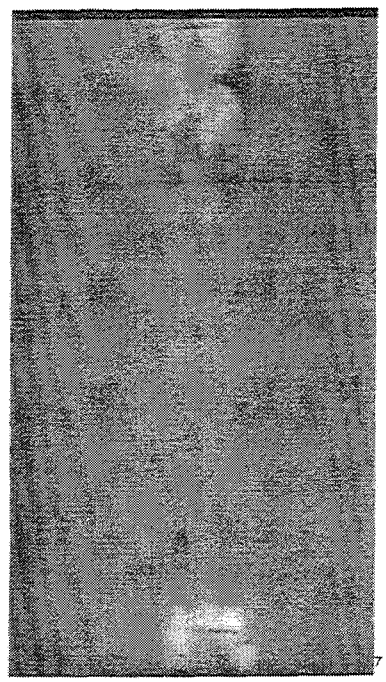
- Flame Impingement on Tile
- Short Tile Life
- Tips too Low in Tile
- Repair at Turnaround



FH 2004/10-86

Burner Problems

- Top Burner Properly Installed
- No Flame Impingement on Tile
- Bottom Burner Improper



Burner Problems



FH 2004/10-88

Troubleshooting

- The Chinese Characters on the wall translate to
- "When at work, always do your very best"



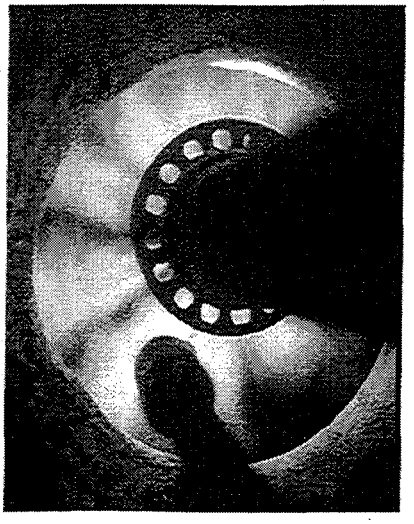
FH 2004/10-89

Training Services

Field Evaluation of Fired Heaters-Part 2

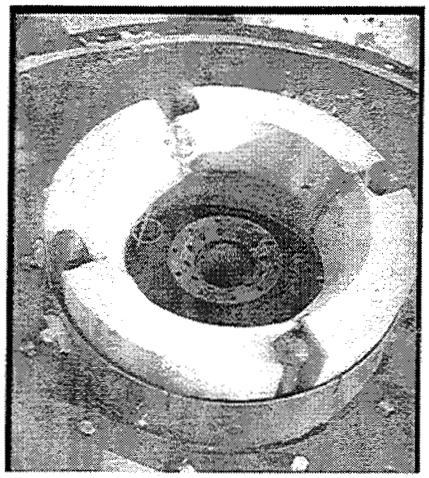
Burner Problems

- Improper Tile Installation
- High Air Close to Pilot



Tile Alignment Ring

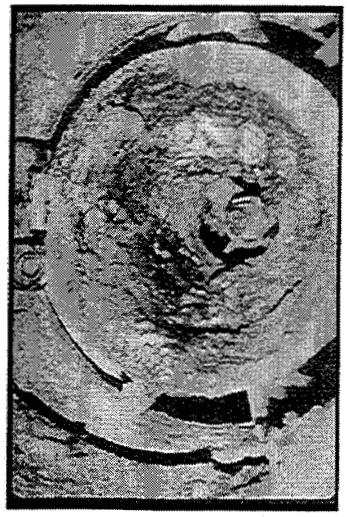
- Tile Alignment Ring Ensures Proper Installation



FH 2004/10-92

Burner Problems

- Burner tile damage caused oil spray on tubes
- High temperature caused tube failure



FH 2004/10-93

Flameout

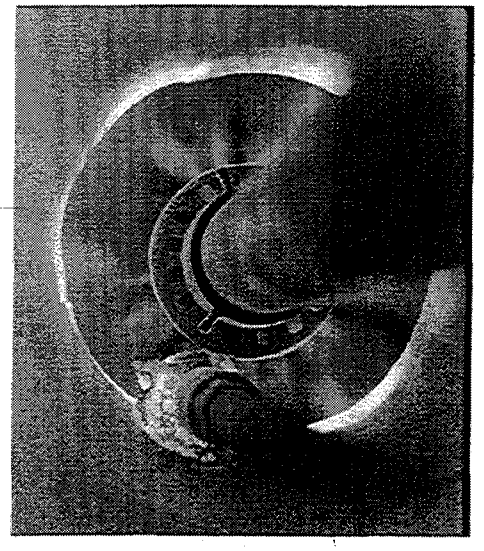
- Oil burner spaying unburnt fuel into heater
- Caused by Low Oil Pressure or High Steam Pressure
- Particulate Fouling on tubes



FH 2004/10-94

Burner Problems

- Flame Cone Located too Low
- Plugged gas tips



FH 2004/10-95

Burner Problems

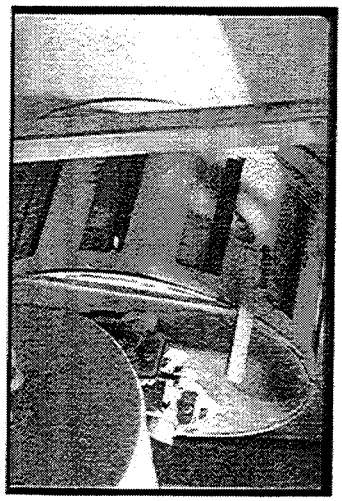
- Improper Secondary Tip Location in center of burner



FH 2004/10-96

Burner Air Controls

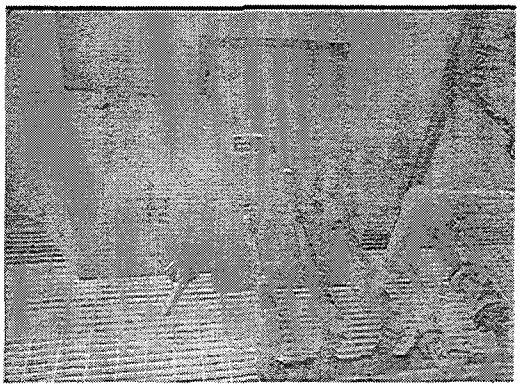
- Concentric Register
- Prone to Sticking
- Difficult to Adjust
- Leaks 50% Air when Closed



FH 2004/10-97

Burner Air Controls

- Damper Control
- Tight Shutoff
- Positive Position
- Good Excess Air Control



FH 2004/10-98

Burner Air Controls

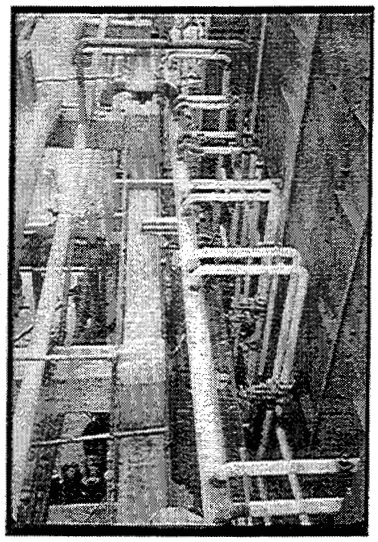
- Leaking Oil
- Burners Need Cleaning
- Improper Oil Temperature at Burner
- Poor Damper Control with Wing Nut
- Cannot Set Same Air Rate to Each Burner



FH 2004/10-99

Troubleshooting

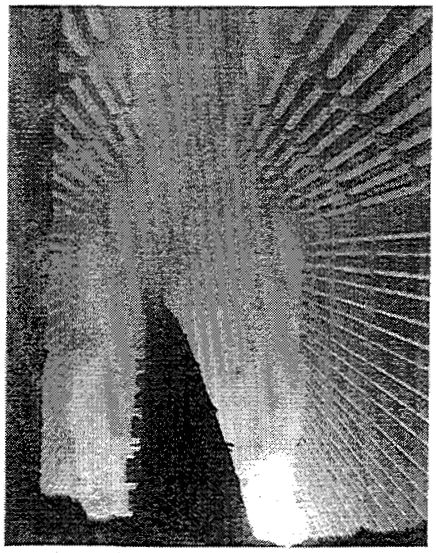
- Gas Piping to Burner
- Connection off top of pipe to prevent scale plugging
- Insulation Required
- Packaged Coalescer to prevent tip plugging
- Low pressure drop pipe to get same fuel rate to each burner



FH 2004/10-100

On-Line Inspection

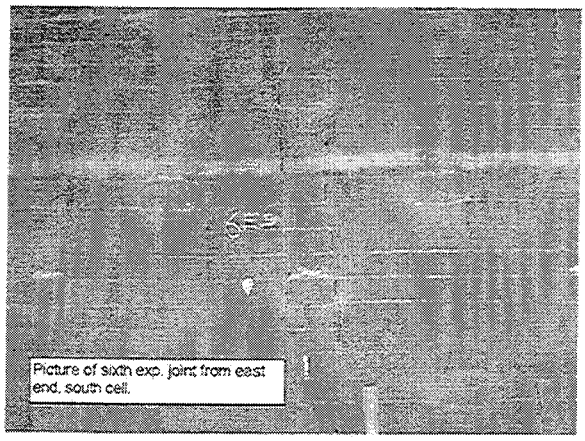
- Brickwall- Improper interlaced expansion joint
- Brickwall- expansion joint gaps are too small
- High Maintenance problems



FH 2004/10-101

On-Line Inspection

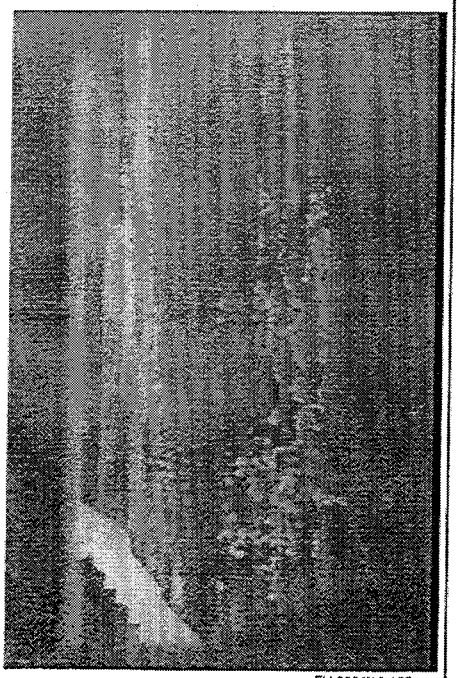
- Brickwall- Improper interlaced expansion joint
- Marked for removal at 10° Lean



FH 2004/10-102

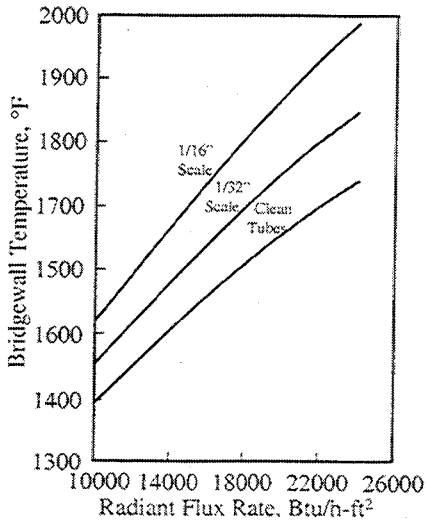
Troubleshooting

- Platforming Heater- 9 Cr tubes
- Heavy Scale
- Increased fuel firing
- Yellow scale 1750 F
- Average surface 1350 F



FH 2004/10-103

UOP Method Platforming Double Fired U Tube Heater



FH 2004/10-104

Troubleshooting

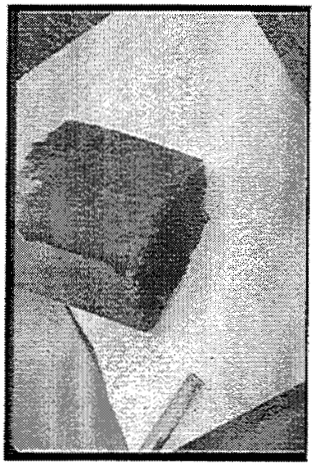
- **Use walnut shells to blast loose scale off tubes on line**
- **Reduced fuel firing**



FH 2004/10-105

Troubleshooting

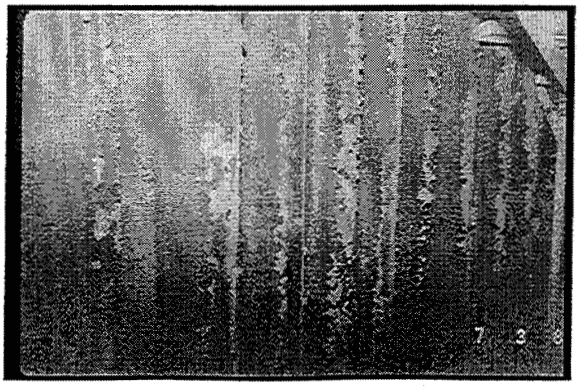
- Platforming tube-9Cr
- Sandblasted to white metal on bottom
- 15-30 mils hard scale on top



FH 2004/10-106

Troubleshooting

- Oil Ash acts as insulation
- Increased fuel firing
- Sandblast clean



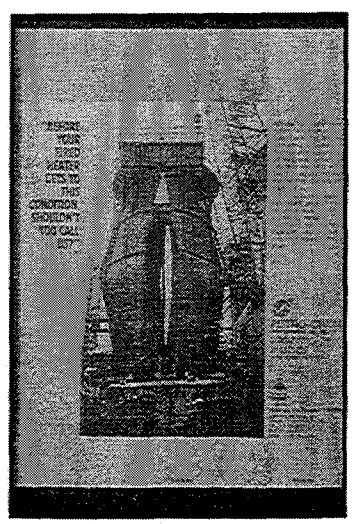
FH 2004/10-107

Establish Work Scope of Turnaround

- On-line Inspection
- Operational Data
- Past Inspection Reports
- Code or Jurisdictional Requirements
- Projected Service

Troubleshooting

- Most Heater Accidents Occur During Lightoff
- 15-20 Minute Purge
- Test for Combustibles?
- Good Operating Instructions
- Burner Management Systems



Troubleshooting

- Operating Instructions at Heater



FH 2004/10-110

Troubleshooting

- Call UOP for New Heater Design



FH 2004/10-111

Training Services

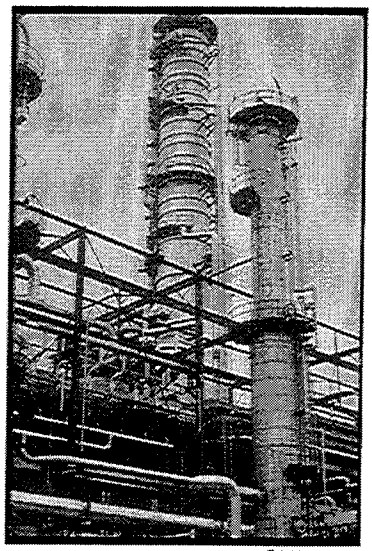
Troubleshooting Crude and Vacuum Heaters

Crude Heaters

- Most Important Heater in refinery
  - Affects entire refinery
  - Large energy consumption
- Very forgiving
  - Two phase flow-high heat transfer
  - Low pressure operation
  - Tube leak not dangerous

Crude Column

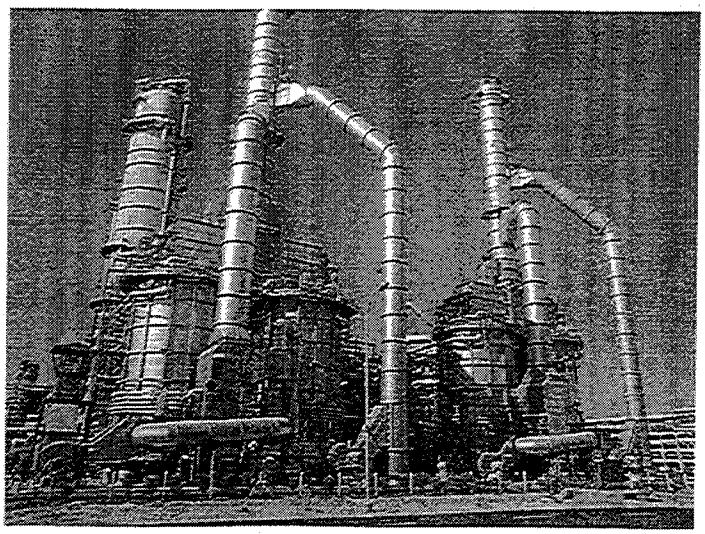
- Arab Light Crude
- 61% volume lift
- Heat input is 423 Btu/lb or 128,000 Btu/BBL.
- Crude Exchange Train 55% Heat Input ($0.21/BBL)
- Crude Heater 45% Heat Input($0.17/BBL)



FH 2004/10-114

World Class Crude Heaters

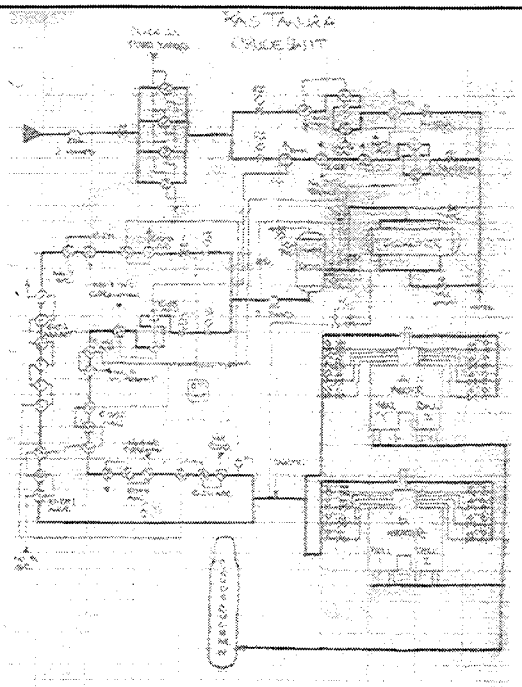
- Air Preheat with 92.5% fuel efficiency



FH 2004/10-115

Ras Tanura Crude Exchange Train

- World Class 550°F
- Vertical crude-ovhd exchange
- Flash Drum to control vapor pressure at heater control valve
- Exchanger Bypass valves for cleaning



FH 2004/10-116
EXC-Reo-09

Crude Heater Coking Problems

- Benchmark- 6" tubes-200MV @ 10,000 Flux rate
  - High Flux Rate with low pass flow causes coking
- Outlet Temperature Range 640°F-735°F
  - Cracking initiates around 700°F
  - Low coking potential @ 640-680°F
- Low pass Flow
  - Limit pass balancing to 10%
  - Clear heater passes during power failure
  - Monitor pass ΔP and % control valve opening
  - Flash Drum Vaporization- 5-10% maximum

FH 2004/10-117

Crude Heater Coking Problems

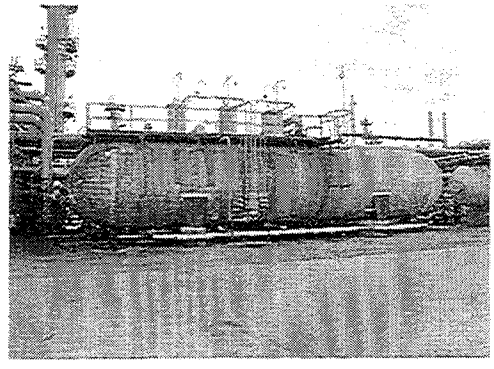
- Asphaltene Precipitation
- Salt Precipitation
- High Oil Film Temperature
- Local Overheating
- Improper Pass Purging
  - Emergency shutdown

Crude Heater Coking Problems

- Asphaltene Precipitation
  - Opportunity Crudes
  - Crude Blending Incompatibility
  - Exchanger Fouling in last two services
  - Loss of Inlet Temp > (7°C/year)
    - Benchmark-Heater Inlet 260°C (500°F)
- Solutions
  - Crude Compatibility Testing
    - Soluble Solutions(www.SolubleSolutions.com)
    - Phone:908-470-0847
  - Spriref Spings
  - Exchanger Bypassses and On-line Cleaning

Crude Desalter

- Operates 121°C (250°F)
- Removes salts in crude
- Removes some Filterable Solids



FH 2004/10-120

Crude Heater Coking Problems

- Salt Precipitation
  - Poor Desalting
  - Caustic Injection
  - Loss of Inlet Temp > (7°C/year)
- Solutions (Optimize Desalter)
  - (90% efficiency per stage)
  - 4-6% water wash rate(water quality)
  - 1/3 water to upsteam exchanger train
  - Demulsifier 4-8 ppm
  - Mix Valve ΔP Optimization(typical 0.5 bar)
  - Acidic pH
  - Limit caustic injection(3-5ppm) with 30 ppm Cl in ovhd water

FH 2004/10-121

Crude Heater Coking Problems

- Asphaltene Precipitation
  - Opportunity Crudes
  - Crude Blending Incompatibility
- Salt Precipitation
  - Poor Desalting
  - Caustic Injection
- High Oil Film Temperature
  - High Flux-low mass velocity design
  - Outlet temperatures around 388°C (730°F)

FH 2004/10-122

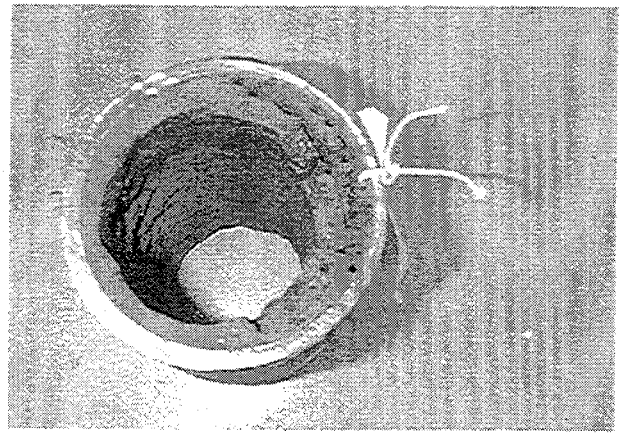
Crude Heater Coking Problems

- High Oil Film Temperature
  - High Flux-low mass velocity design
    - (150 lb/sec-ft2 @ 12,000 Btu/h-ft2)
  - Outlet temperatures around 388 °C (730°F)
  - Skin Temp Rise > (12°C/year)
- Solutions
  - decrease number of passes(expensive)
  - 1-2% condensate injection to heater inlet
    - potential column flooding
    - water disposal
  - frequent pig cleaning

FH 2004/10-123

Crude Tube Deposit

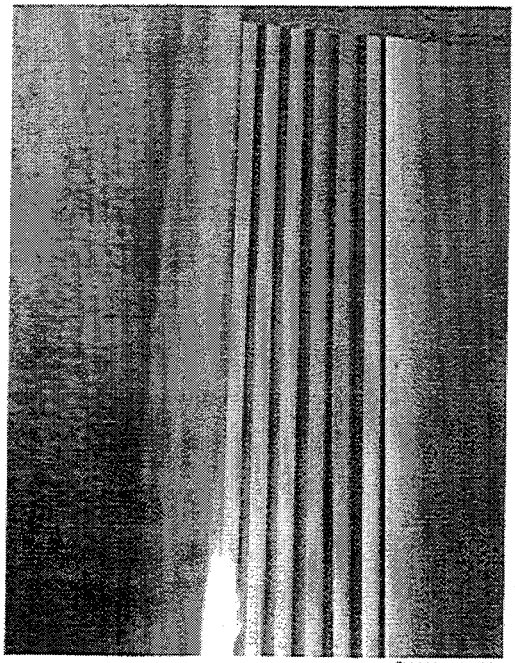
- Typically 85% organic -15% inorganic deposit
- LOI Test for % organic
- 4 mm thickness gives 50-60°C TWT rise



FH 2004/10-124

Crude Tube Deposit

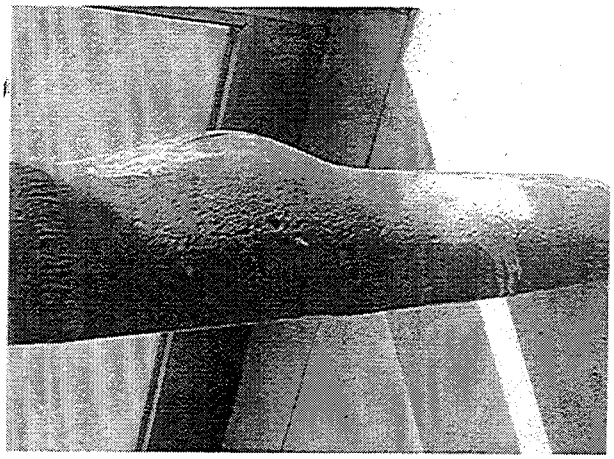
- Oil Fired Heater
- Light color is ash deposit
- Dark color is 7-10 mm coke in tube
- Ash Deposit melt around 670°C(1240°F)
- Coke caused by not clearing oil from pass during power failure



FH 2004/10-125

Crude Heater Tube Failures

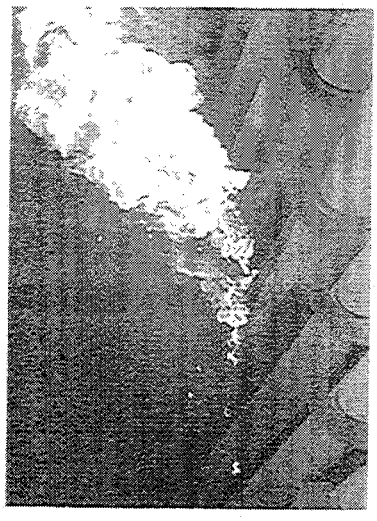
- Local Hot Spot- Internal Coke Deposit-External Pitting
- Replace at 3-5% diameter growth



FH 2004/10-126

Crude Heater Tube Failure

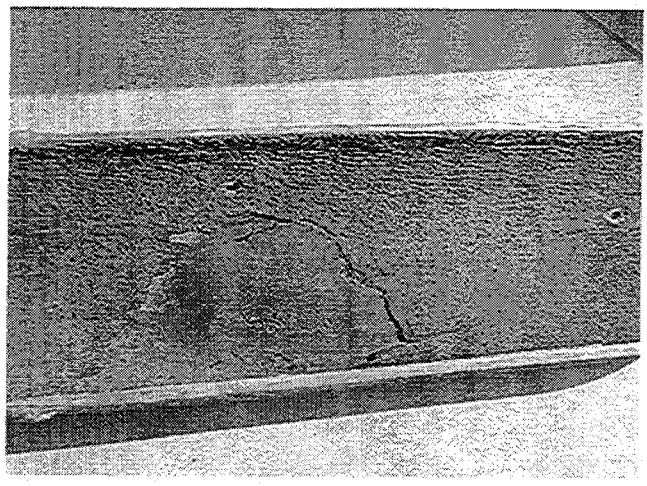
- Leaking Tube
- Most Failures Caused by Flame Impingement
- Poor Burner Maintenance
- Control O2 Level high enough to keep flames off the tubes



FH 2004/10-127

Crude Heater Shock Tube Failure

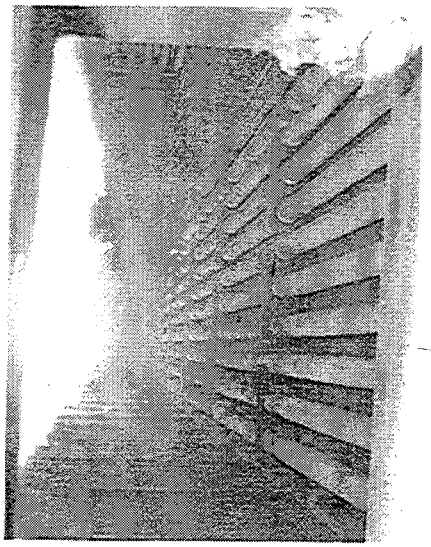
- Often the hottest tube in the heater
- Flame Impingement Problems



FH 2004/10-128

Poor Low NOx Burner Retrofit

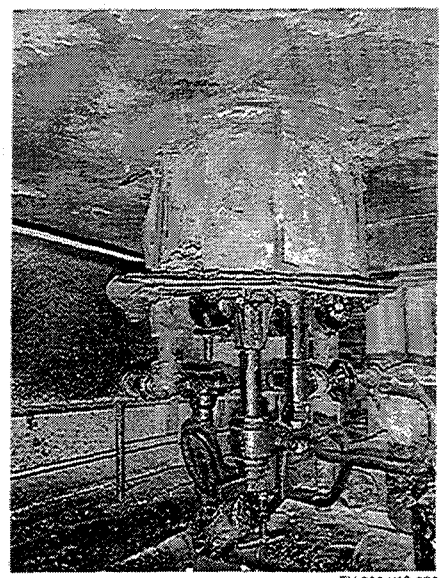
- Flames Are Too Long for 20 foot(6 m) Firebox Height
- Flame Height Should be 50% of Firebox Height
- High BWT
- Hot Roof Supports
- Poor Revamp



FH 2004/10-129

Burner Problems

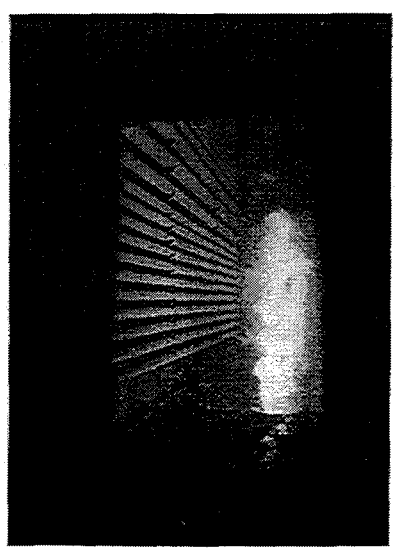
- Burner Problems cause most tube failures
- Burners are in bad condition
- New Burners are recommended
- Oil and steam should have flexible hose connections
- Hard piping have oil gun at wrong height



FH 2004/10-130

Crude Heaters

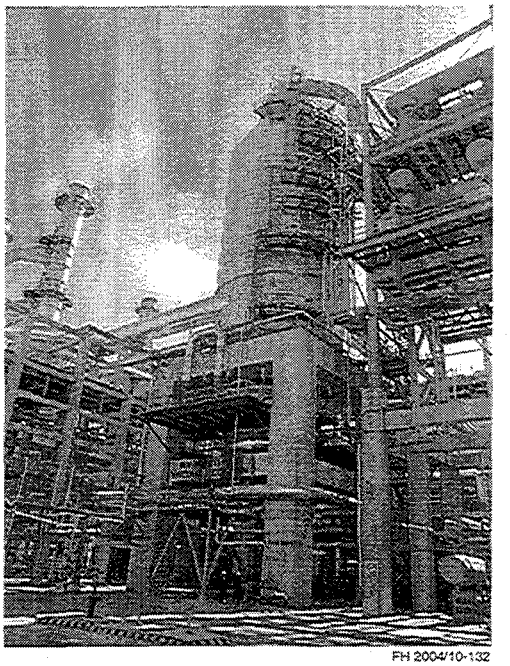
- Flames Are Proper Height
- Flame Height Should be 50% of Firebox Height
- Good Heater Design



FH 2004/10-131

Vacuum Unit

- 700- 760°F Flash Zone
- 20- 100 mm Hg Flash Zone
- Higher Temperatures, more coking potential
- Wet, Dry, Damp Units
- Minimize transfer line ΔP
- Heater located minimum distance from column



FH 2004/10-132

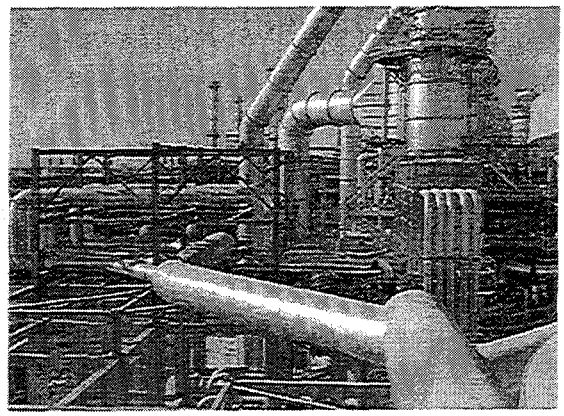
Deep Cut Operations

- 1100°F (593°C) Cut Point
- Low Flash Zone-20-30 mmHg
- Heater Outlet-800°F(426°C)
- Steam Injection
  - 0.5 wt% to heater
  - 1.0 wt% to column boot
- Low Transfer Line Pressure Drop
- Low Radiant Flux Rate

FH 2004/10-133

Vacuum Column Transfer Line

- Design for 2-4 psi ΔP to give 15-35°F rise at heater
- Minimize flow disturbances(erosion-corrosion problems)
- Thermal Expansion Concerns



FH 2004/10-134

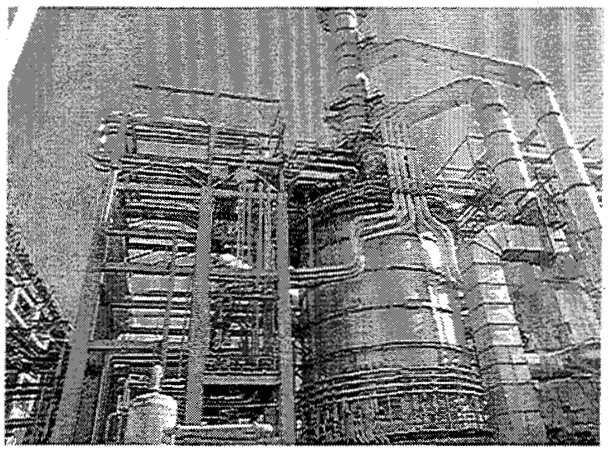
Vacuum Heaters

- Benchmark- 4-6" tubes-250MV @ 8,000 Flux rate
  - Outlet Tube 8'-10"
- Outlet Temperature Range 735°F-800°F
  - Cracking problems around 11,000 flux
  - High coking potential @ 800°F
- Pass Balancing
  - Limit pass balancing to 10%
  - Clear heater passes during power failure
  - Monitor pass ΔP and % control valve opening
  - Steam Injection-0.5 wt%

FH 2004/10-135

Vacuum Heater

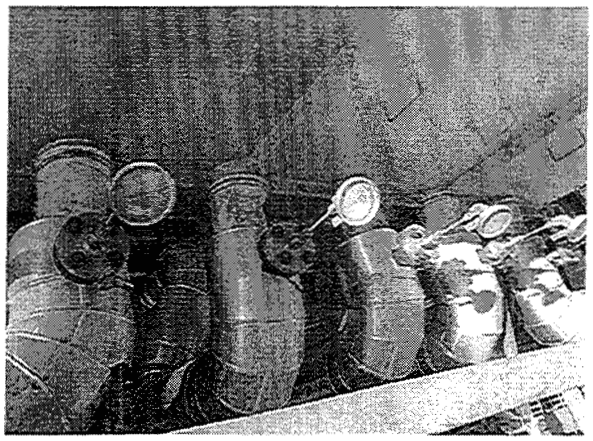
- Modern Vertical Tube design minimizes hot residence time
- Accommodates new low NOx burners better



FH 2004/10-136

Vacuum Heater Crossover Lines

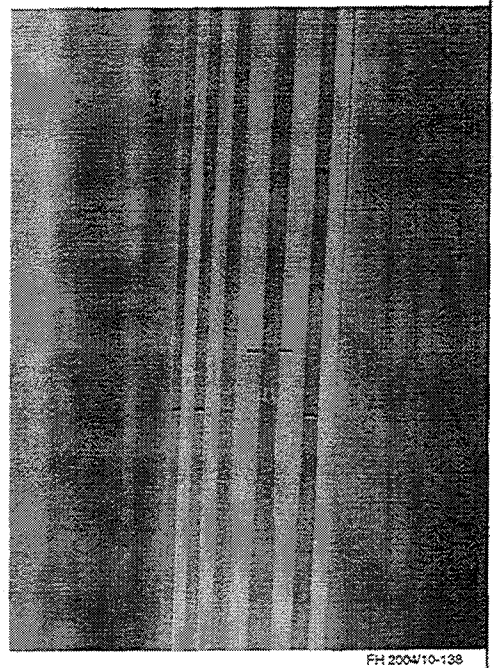
- Liquid Phase until the last few tubes
- Local TI gives convection duty



FH 2004/10-137

Vacuum Heater

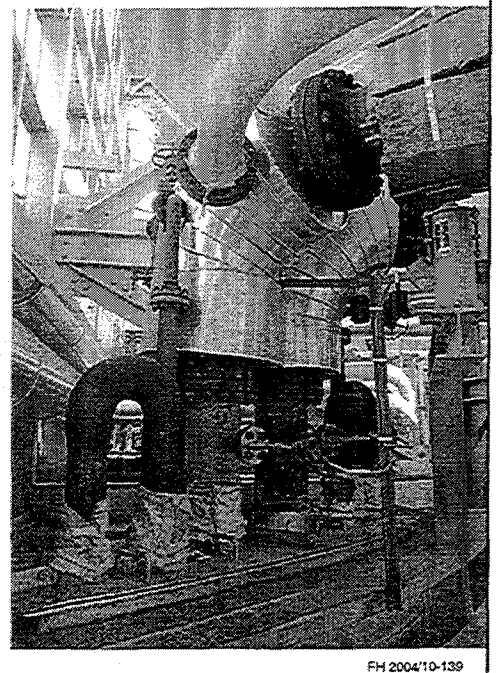
- Increase tube diameter at point of vaporization to minimize temperature rise
- Steam injection at vaporization point
- Sonic velocity (380-420 fps)
- Locate Large outlet tube in cold location



FH 2004/10-138

Vacuum Heater

- 0.5 wt% steam injection at point of vaporization to reduce temperature peaking
- Removeable pipe spool on outlet for tube pigging
- Swing elbow for decoking
- Outlet TI on each outlet



FH 2004/10-139

Cracking Monitoring

(Ejector Off-Gas&Ovhd Drum Liquid)

- Normal 0.15 Wt% Dry Gas
- Normal 0.4 Wt% Naphtha
- Normal 1.0 Wt% Distillate

- Maximum 0.3 Wt% Dry Gas
- Maximum 0.75 Wt% Naphtha
- Maximum 2.0 Wt% Distillate

FH 2004/10-140

Process Operating Considerations

Crude

This service is very forgiving. Large cabin heaters can operate at 14,000-15,000 Btu/h-ft^2 radiant flux . Tube support mechanical design usually limits operations. Monitor coil ΔP, BWT, and flux rates.

Vacuum

This service is not as forgiving as crude heaters. Radiant flux rates are usually limited to 10,000-11,000 Btu/h-ft^2 due to thermal cracking considerations. Monitor coil ΔP, BWT, and flux rates. Monitor ejector off-gas rates to determine the rate of cracking. The off- gas rates should be less than 0.3 wt.% of charge rate. Highly paraffinic crudes with UOP 'K' factors of 11.8-12.0 may be prone to cracking.

Visbreaking

Monitor fuel oil stability. Overcracking results in an unstable fuel oil. Skin thermocouple readings and coil ΔP are very important. Measure the product BS&W. It should be 0.1 higher than the feed. If it is the same as the feed , raise the outlet temperature 5∞F. If it is 0.2 higher than the feed , lower the outlet temperature 5∞F.

Naphtha Hydrotreating

Monitor radiant flux rates, skin thermocouple readings and BWT. Fouled combined feed exchangers are a problem for this process, resulting in much higher flux rates. H2S corrosion is significant for non austenitic coils. Shot cleaning removes this scale. Tube thickness readings to ensure proper thickness are very important for this process.

Platforming

Monitor radiant flux rates and BWT. Sandblasting tubes will lower BWT and improve fuel efficiency.

Distillate Hydrotreating and Hydrocracking

Monitor radiant flux rates, skin thermocouple readings and BWT. Neutralize 347H coil during shutdown to prevent stress-corrosion cracking.

Visual Observation

Fire Box

1. Colors of Flame

The flame from a gas should be blue or blue with small flecks of yellow. With oil the flame should be "clean" yellow color. These represent good combustion.

2. Evidence of Incomplete Combustion

Any smoke in the fire box is to be avoided. This is evidence of incomplete combustion. Long "hazy" yellow flames which may be impinging on the tubes is an excellent indication of poor combustion.

3. Coloring of Refractory and Tube Supports

Uniform coloration of brick work is very desirable. Many times the first indication of poor flame distribution can be gained by observation of extremely hot brick work or tube hangers in certain portions of the furnace.

4. Appearance of Tubes

Hot spots can be observed on tubes many times by their cherry red appearance. Care and experience are required in such observations. Many times what appears to be a hot tube may be only incandescent scale on the tube surface.

A quick estimation of firebox temperatures can be made visually using the following temperature color scale taken from Mark's "Standard Handbook for Mechanical Engineers".

Relation of Color to Temperature of Steel

Temperature °F	Color
990	Barely discernable red, black red, dark blood red
1050	Dark red, blood red, low red
1175	Dark cherry red
1375	Cherry, full red
1550	Light cherry, light red, bright cherry red
1650	Orange
1725	Light Orange
1825	Yellow
1975	Light Yellow
2200	White

This scale can also be used to estimate tube wall temperatures above 1000°F. It can be used to estimate the flue gas temperature leaving the radiant section. Look at the color of the roof tube supports. The tube supports will be approximately at the same temperature as the flue gas.

5. Appearance of Refractory, Tube Supports, Bridge Wall and Floor

All refractory should be in first class conditions. Any loose rubble or broken brick laying on the furnace floor may become a flame target and give poor heat distribution. One point on floors, during turnarounds, care should be taken that floor bricks are not installed tight because resulting expansion after the start-up will cause the floor to buckle.

Broken tube supports will result in eventual sagging of furnace tubes and perhaps leaks at the header.

Items in this category are to be watched after they occur, and should be reported to the area supervisor.

6. Appearance of Convection Section

The convection section should be clear with maybe a wisp of smoke coming in from time to time. This will occur when very efficient combustion is occurring.

Furnaces and Tubes

1. Stack Damper Setting

In general the stack damper on most of the Refinery furnaces should be set to give a .05 to .15 inches of water(vacuum) draft inside the fire box. By draft we mean the difference between the atmospheric pressure outside the lower pressure inside the furnace.

2. Appearance of Flue Gas from Stack

Many times the first indication of incomplete combustion or leaks in convection section tubes can be observed by smoke coming from the stacks. When firing fuel gas, no visible smoke should come from the stacks. On oil, a slight smoke haze may come from the stack with good combustion.

3. Stack Appearance

On lined stacks, evidence of loss of lining can be observed by spot discoloration of the stack.

4. Coil Inlet and Outlet Pressure

Most furnaces are equipped with pressure gauges on the coil inlet and outlet. Some indication of furnace coking can be obtained by periodic observation of coil pressure drop. Other factors that must be taken into consideration in analysis of this information are feedrate, temperature, and feedstock.

Cause of Hot Spots

What causes local "hot spots" in furnaces?

An even distribution of heat in a furnace is required for high efficiency and avoidance of local overheating. Most of the heat in a furnace is secured by direct radiation from the flame and combustion gases. The remaining heat is secured by radiation from the hot refractory walls and by convection, the transfer of heat by the moving gases.

It is important that the velocity and distribution of gas flow be controlled. The combustion-gas layer nearest to the cold surface transmits heat faster than does the gas in the middle of the chamber, leaving a hot center. When no control combustion-gas flow exists, the hot portion may sweep out of the furnace without transmitting its heat efficiently. This causes high stack temperature and may result in local hot spots.

Furnace tubes are constantly being heated on the outside and cooled on the inside. If an insulator such as a large coke deposit is developed on the inside surface, then the rate of heat transfer is decreased and hot spots develop on the tubes. Similarly, an insulating liquid film at low charge rates (nonturbulent flow) will result in a general tube-temperature increase.

The materials of which furnaces and tubes are built are not designed to continuously withstand the temperatures which develop in hot spots. Therefore, unless these temperatures can be lowered, the result is a rapid material failure.

25 Pointers on Better Burner Operation

1. A thin, fluttery oil fire means too much steam for atomization.

2. A long, smokey flame may be due to improper burner-tip design, insufficient combustion air, or insufficient atomizing steam.

3. A dazzling white oil fire means either too much excess to much atomizing steam.

4. An oil burner operating at proper adjustment makes a bright yellow flame which verges on whiteness.

5. A reddish, dusty looking flame with flocks of smoke over the bright part of the flame indicates lack of enough air to burn the fuel.

6. Sparks in the flame may come from a dirty tips wet steam, solids in the fuel, water in the fuel, or cold oil.

7. An uneven flame may mean a, dirty tip.

8. A properly adjusted oil burner makes a clean bright yellow flame with no trace of smoke.

9. Failure of ignition may be due to a low fuel pressure, too much atomizing. steam, fuel valve plugged. improper design of tip, or wet steam with slugs of water from steam line.

10. Puffing in the furnace may be caused by poor tip design, in-sufficient excess air, or partial stoppage of the burner tip.

11. Operation may be erratic and the flame may sputter at fuel-oil temperatures above the flash point.

12. The higher the temperature of fuel oil, the better the performance unless premature vaporization occurs or unless water content of the fuel is high.

13. Excessive coking in burner tips may be caused by highly superheated atomizing steam along with a high fuel-oil temperature.

14. Rapid coke formation and refractory deterioration may be caused by excessive flame impingement against the muffle block. This is usually caused by the oil burner being too short, the oil too cold, the flame angle too wide, the oil tip misaligned, too much fuel being burned, too much excess air, or faulty installation of the burner.

15. When firing 200 SSU (43 cs) viscosity oil, set the steam differential pressure controller at 30 psig (2.1 Kg/Cm2). For every increase of 100 SSU in the oil viscosity, raise the atomizing steam pressure by 5 psig (0.35 Kg/Cm2).

16. The oil flame may be shortened by increasing the excess air or increasing atomizing steam slightly.

17. Uneven furnace temperatures may be caused by nonuniform burner flames.

18. When lighting an oil burner, always turn on the atomizing steam first, then open the oil valve slowly until the burner lights.

19. Always use a large torch in place before the fuel is turned on to light burners.

20. Always blow down steam lines before lighting burners.

21. A cold furnace should only be lighted after all air registers and dampers have been opened several minutes, allowing ample time for steam purging to remove through the stack any accumulated gas in the fire box.

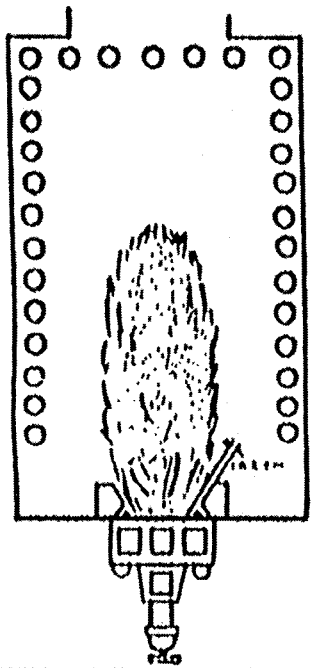
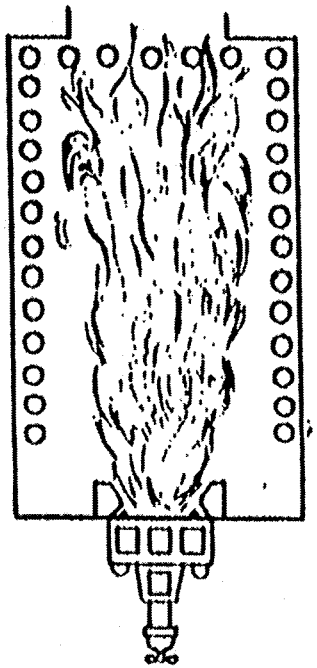
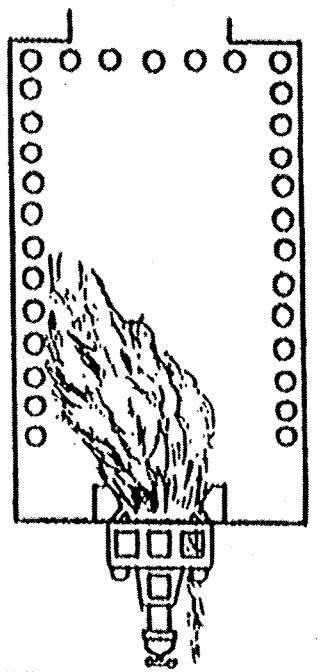
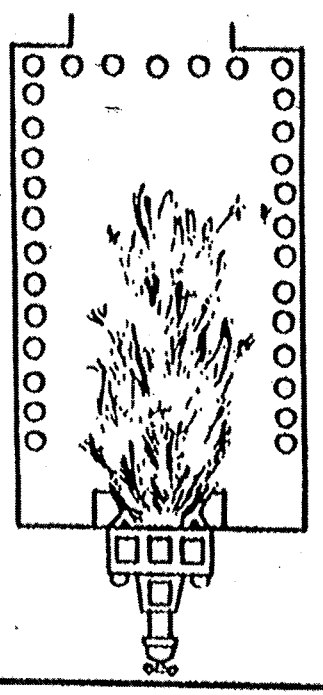
22. Be sure that the stack damper is open when lighting a cold furnace; adjust draft with burner air registers.

23. Do not fire a burner simultaneously on gas and oil because it will starve the oil fire for air, and a puffing of the heater typical of air starvation may result. The only exception is lighting the oil burner with the gas burner. Once the oil flame is stable, turn the gas flame off.

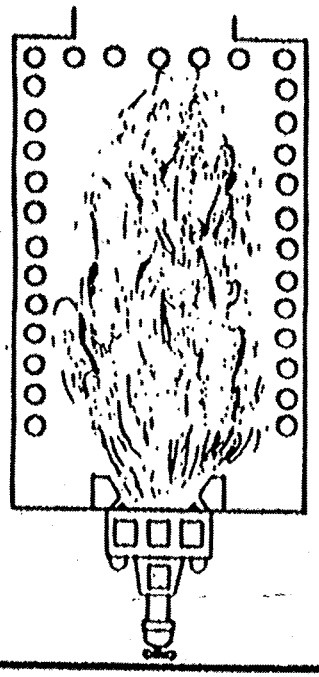
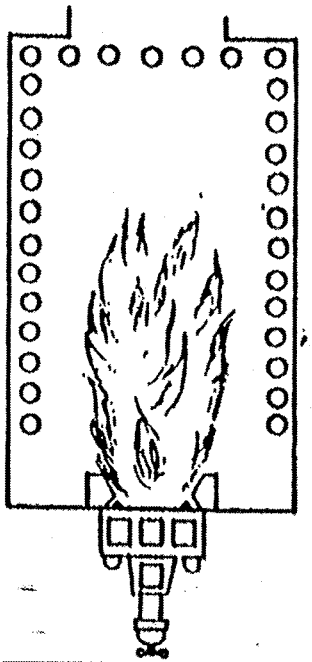
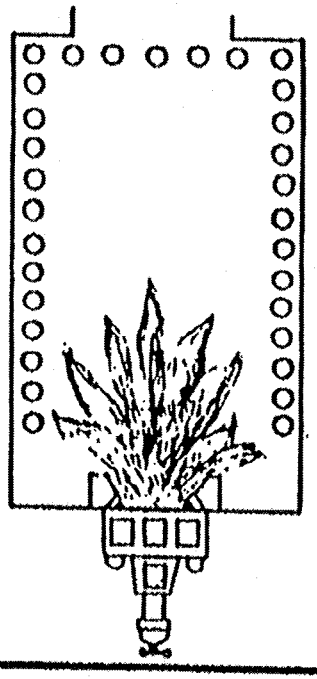
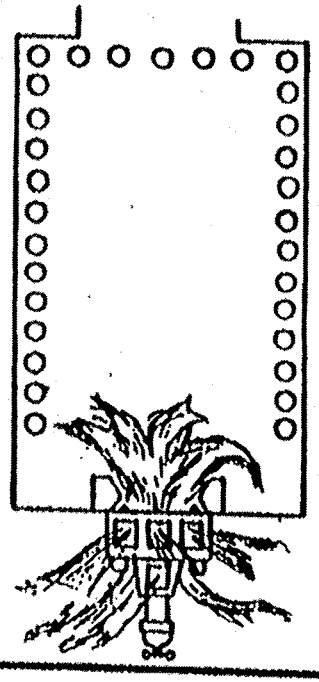
24. The flame may be shortened by increasing the excess air. The PVYD burners in heater B-801 will typically have a flame length of 1 1/2 Ft/MM BTU at 20% E.A. and 3 Ft/MM BTU at 10 E.A. The PMA burners in heaters B-51 and B-52 will have a flame length of 1 Ft/MM BTU at 20% E.A. and 2 Ft/MM BTU at 10% E.A.

25. At 20% E.A. a gas flame will not have any haze. At 15% E.A. the gas flame will appear feathery with fuzzy edges. At 10% E.A. there will be 12 to 24 inches (0.3 to 0.6 meters) of haze on the end of the flame. A properly adjusted gas flame may be colorless (PMA type burner) or be blue with traces of yellow (PVYD burner).

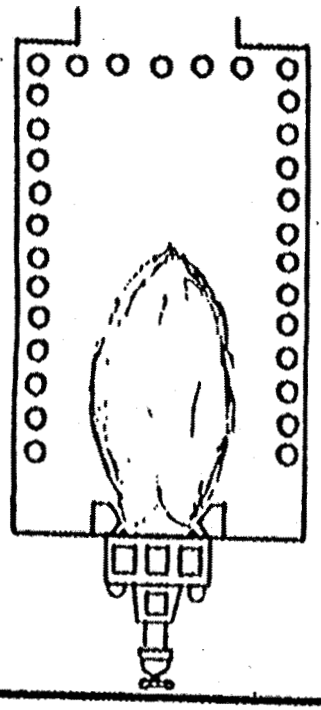
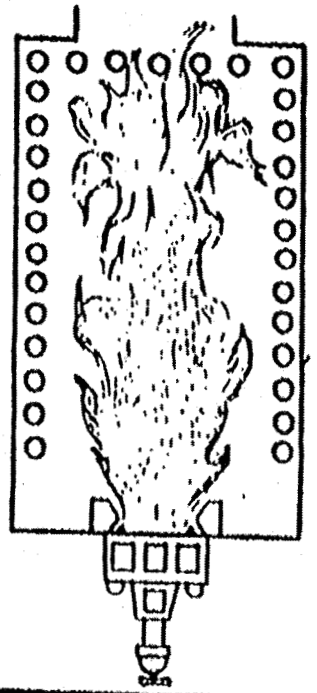
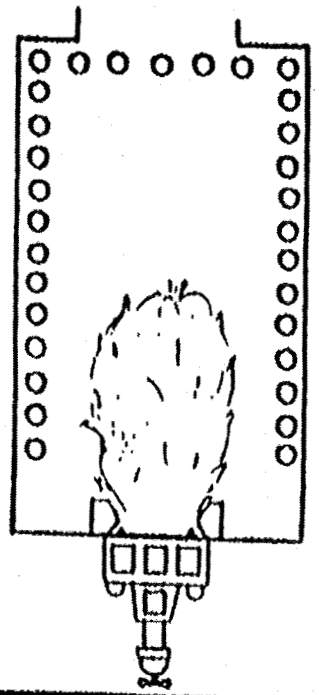
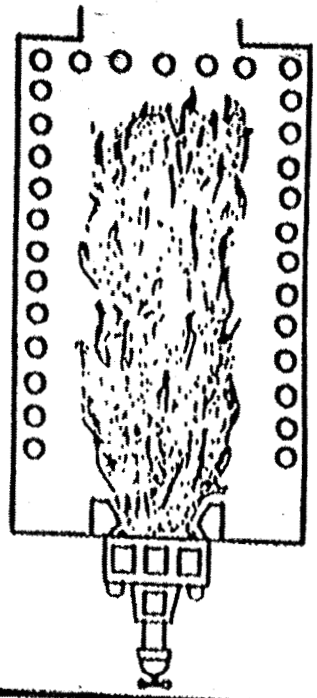
A good understanding of fuel, burners, and combustion principles leads to intelligent, efficient furnace operation.

1. Good oil flame	2. Bad oil flame	3. Bad oil flame	4. Bad oil flame
			
Quiet, clear, smokeless, steady flame. Color: bright yellow to orange	Hellfire Flame impinging against tubes. Incomplete combustion. Too much oil, too little air, or too little atomizing steam, or worn out burner head.	Slanting flame. Flame impinges against tubes on one side. Burner lance slanting or too low. Faulty atomizer. Oil leaks - fouling. Oil temperature too low. Too little atomizing steam.	Sparks Water in oil. Dirt in oil. Oil too cold. Too little atomizing steam.

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5. Bad oil flame	6. Bad oil flame	7. Bad oil flame	7a. Flame flash-back
			
Smoky flame. Too little combustion air. Worn out nozzle. Oil too cold. Burner head fitted too low.	Dazzling flame. Too much secondary air. Too much atomizing steam.	Poor shape. Too little primary air. Spalled brickwork.	Too little draft. Too little primary air (oil). Gas pressure too low.

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8. Good gas flame	9. Bad gas flame	10. Bad gas flame	11. Bad gas flame
			
Quiet, bright, smokeless steady flame. Color: blue with yellow flakes (dependent on carbon content of gas)	Hellfire, long yellow flame. Flame impinges against tubes. Poor combustion. Too much fuel. Hardly sufficient air. Worn our burner orifices.	Dazzling flame. Too much secondary air.	Smoky flame. Too little combustion air.

"Reprinted courtesy of John Zink, Co."

Process Heat Balance

The best method to obtain the process side heat balance is to use a process simulation package such as HYSYS. If this is not available, accurate results can be obtained from specific heat charts and latent heat charts.

Liquid phase calculations are easy. All that is required is a specific heat from PD Chart-854 or PD Chart-855.

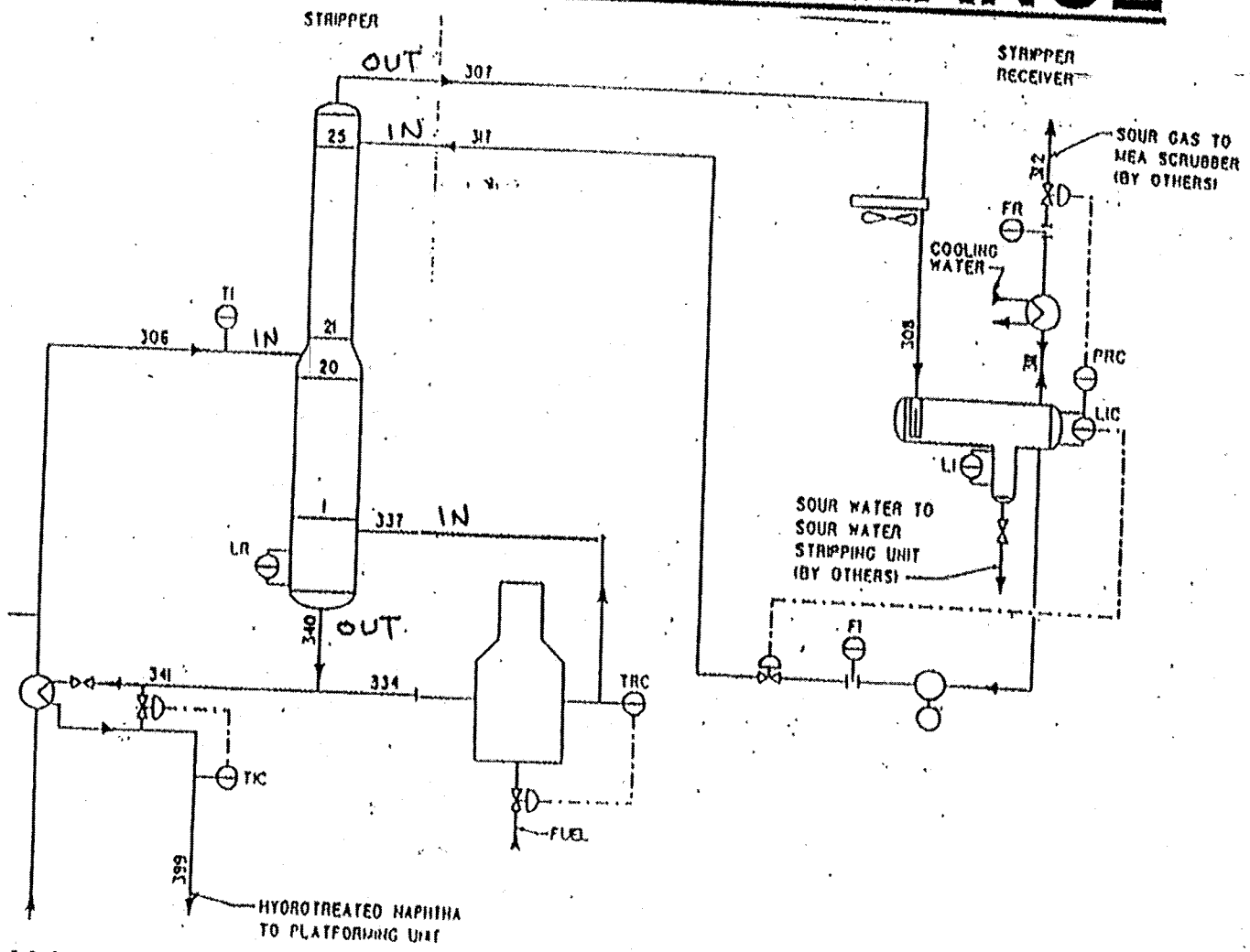
Vapor phase calculations require a latent heat calculation and a sensible heat calculation.

Latent heat calculations are performed as follows:

1. Obtain the temperature and pressure of the vapor steam.
2. Use the Cox charts (PD Chart-854 or PD Chart-855) to adjust the boiling point at elevated pressure to boiling point at atmospheric conditions.
3. Use the Latent heat chart. Read up from the atmospheric boiling point to MW or API. Read across to obtain latent heat of vaporization.

For the Reboiler Heat Balance example, the overhead steam leaves the column at 294°F at 165 psia. Go to PD Chart-850. Read up from 294°F to 165 psia. Draw a parallel line. Extrapolate down to 15 psia. Obtain an atmospheric boiling point of 125°F at 15 psia.

REBOILER HEAT BALANCE



UOP

Reboiler Heat Balance

Overhead-307

Stream Conditions = 72.4 MW @ 294°F

Latent Heat = 152 Btu/lb (Chart PD-866)

Vapor Specific Heat = 0.46 (Chart PD-855)

Sensible Heat = 0.46 (294°F-60°F) = 107.6 Btu/lb

Enthalpy = Sensible Heat + Latent Heat = 107.6 + 152 = 259.6 Btu/lb

Heat Out = 62292 lb/h (259.6 Btu/h) = **16.17 MM Btu/h**

Reflux-317

Stream Conditions = 82.6 API @ 115°F

Liquid Specific Heat = 0.53 (Chart PD-854)

Sensible Heat = 0.53 (115°F-60°F) = 29 Btu/lb

Enthalpy = Sensible Heat + Latent Heat = 29 +0 = 29 Btu/lb

Heat in = 62097 lb/h (29 Btu/h) = **1.80 MM Btu/h**

Bottoms Out-340

Stream Conditions = 59.7 API @ 422°F

Liquid Specific Heat = 0.60 (Chart PD-854)

Sensible Heat = 0.60 (422°F-60°F)= 217 Btu/lb

Enthalpy= Sensible Heat + Latent Heat = 217 + 0= 217 Btu/lb

Heat In = 1553612 lb/h (217 Btu/h) = **337.13 MM Btu/h**

Feed In -306

Enthalpy = Sensible Heat + Feed Bottoms Exchanger Duty

Heat In = 11.09 MM Btu/h + 29.17 MM Btu/h = **40.26 MM Btu/h**

Sensible Heat

Stream Conditions = 60.2 API @ 130°F

Liquid Specific Heat = .51 (Chart PD-854)

Sensible Heat = 0.51 (130°F-60°F) = 36 Btu/lb

Enthalpy = Sensible Heat + Latent Heat = 36 + 0= 36 Btu/lb

Heat In = 308213 lb/h (36 Btu/h) = **11.09 MM Btu/h**

Feed-Bottoms Exchanger Duty

Stream Conditions = 59.7 API @ 422°F (inlet) & 281°F (outlet)

Liquid Specific Heat = 0.60 (inlet) (Chart PD-854)

Sensible Heat = 0.60 (422°F-60°F)= 217 Btu/lb (inlet)

Liquid Specific Heat = .55 (outlet) (Chart PD-854)

Sensible Heat = 0.55 (281°F-60°F) = 122 Btu/lb (inlet)

Enthalpy = Sensible Heat + Latent Heat = 36 + 0 = 36 Btu/lb

Duty = 307019 lb/h (217 Btu/h-122 Btu/h) = **29.17MM Btu/h**

Reboiler Heat Balance

Stream	Hand Calc. MM Btu/h	Computer Calc.MM Btu/h
Overhead Out	-16.7	-16.02
Reflux In	+1.80	+1.80
Bottoms Out	-337.13	-336.73
Feed In	+40.26	+40.36
Reboiler In	+311.77	+310.57

To Reboiler, Enthalpy = 1246595 lb/h (217 Btu/h) = 270.51 MM Btu/h

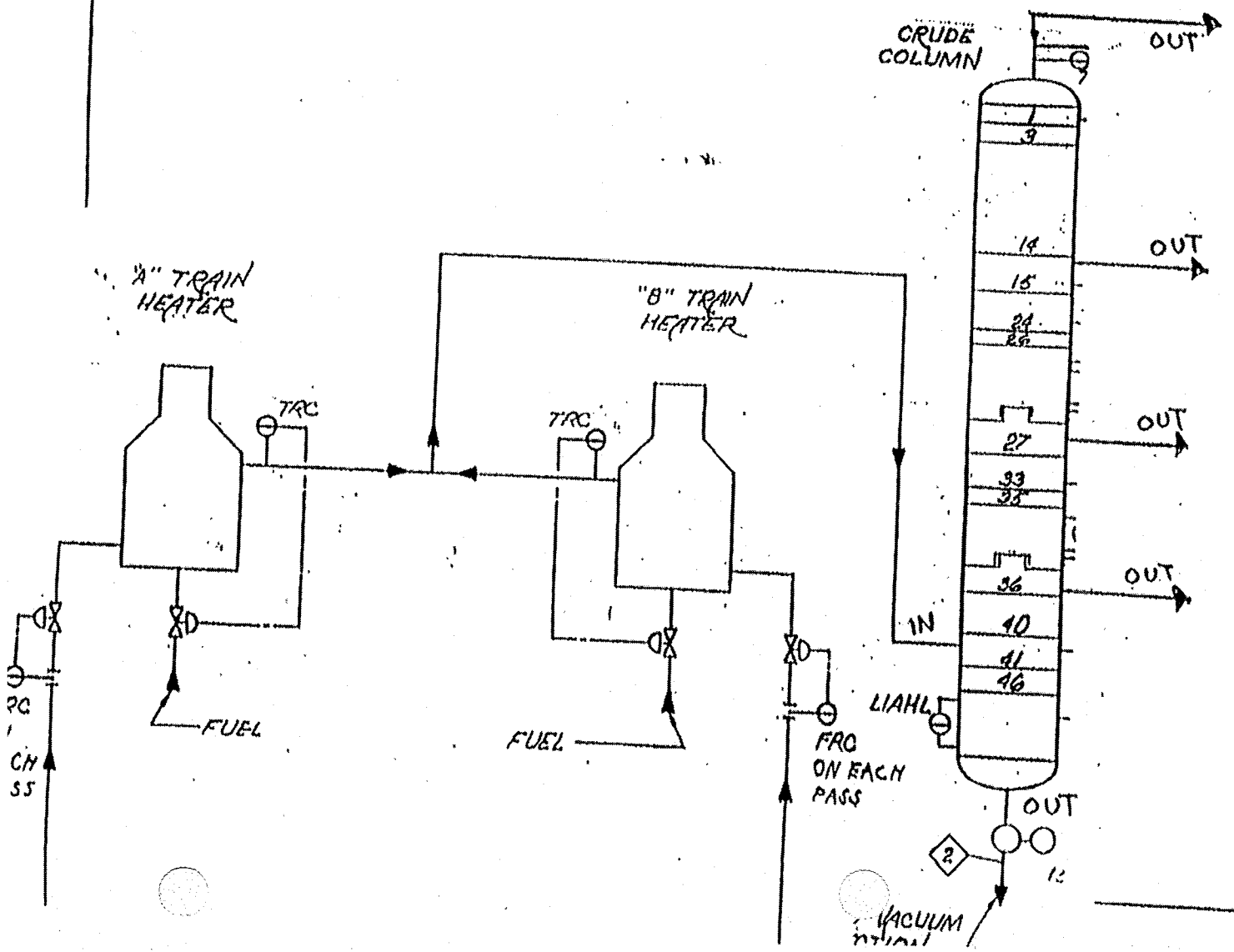
Reboiler Duty = Enthalpy Out - Enthalpy In

Reboiler Duty(Hand Calc.) = 311.77 - 270.51 = **41.26 MM Btu/h**

Reboiler Duty(Computer) = 310.57 - 270.18 = **40.4 MM Btu/h**

Error =2.1%

CRUDE UNIT HEAT BALANCE



Crude Column Heat Balance

Crude to Heater @ 398°F

	BPSD	Lb/h	API	MW	Latent Heat	Sensible Heat	Total Btu/lb	MM Btu/h
Crude	86170	1071353	34.55	205.08	0	184	184	197.13

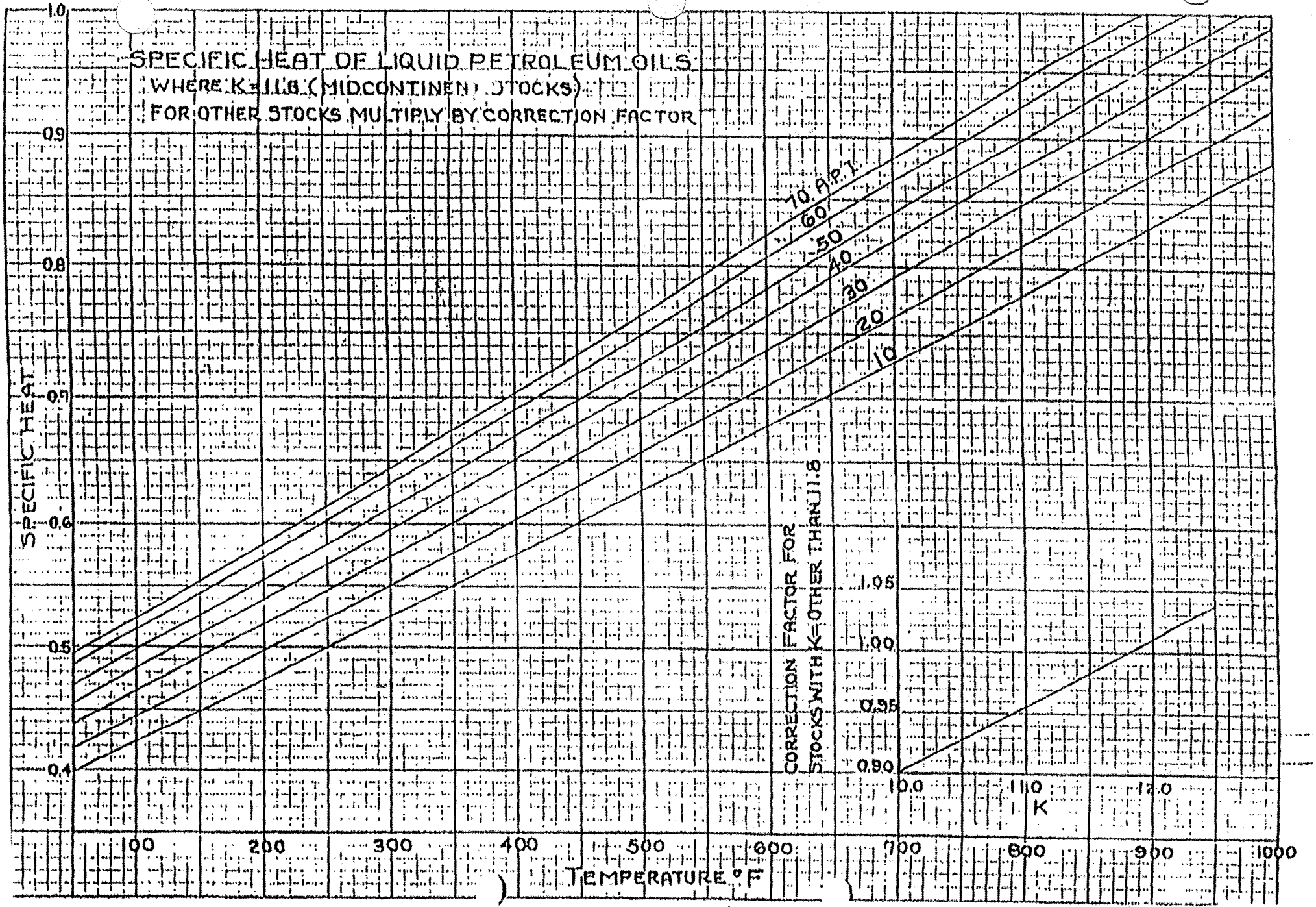
Crude at Flash Zone @ 706°F

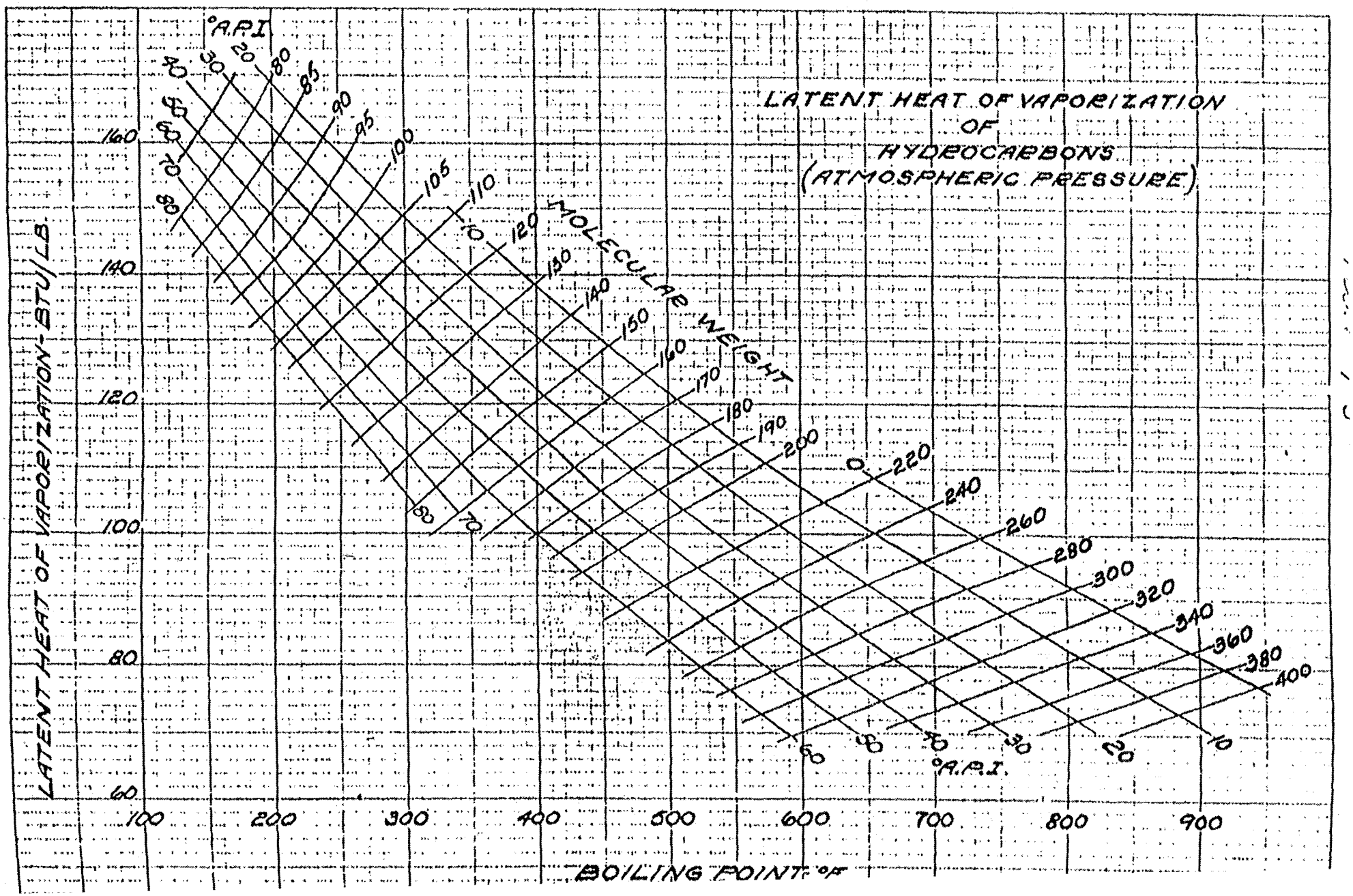
	BPSD	Lb/h	API	MW	Latent Heat	Sensible Heat	Total Btu/lb	MM Btu/h
Ovhd.	2267	18283	-----	52.5	160	458.9	618.9	11.32
LSR	6179	58982	84.8	78.5	144	458.7	602.7	35.55
Naphtha	11739	122878	59.68	106.53	135	439.3	574.3	70.66
Kerosine	12583	145781	46.70	149.2	116	419.9	535.9	78.12
Diesel	15853	195995	35.49	230.82	94	407	501.0	98.19
AGO	3420	44246	28.08	309.45	81	400.5	481.5	21.31
Resid.	34489	485188	15.26	484.3	0	374.7	374.7	181.79
Total	86170	1071353						496.94

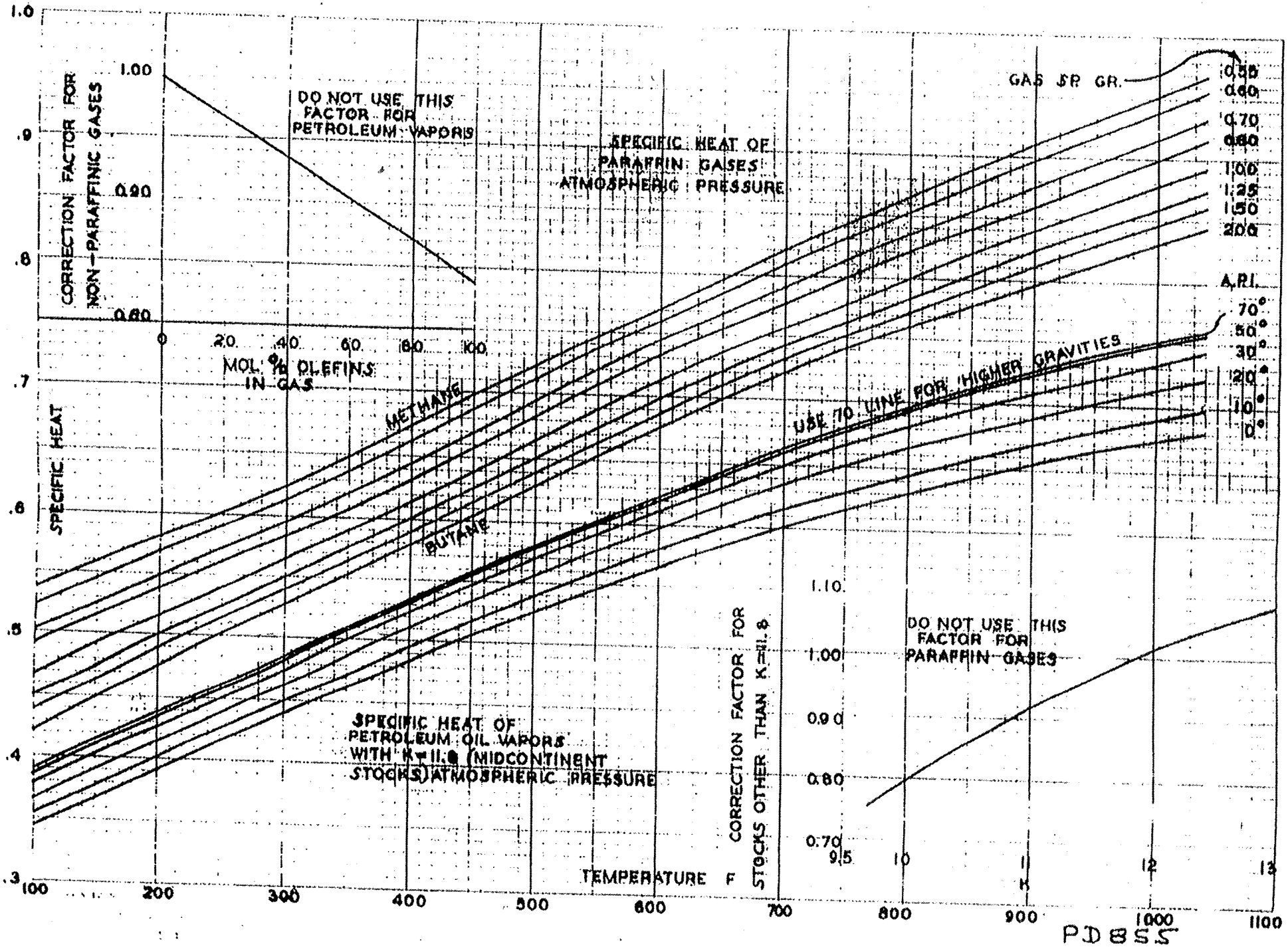
Crude Heater Duty (Hand Calculation) = 496.94- 197.1 = **299.84 MM Btu/h**

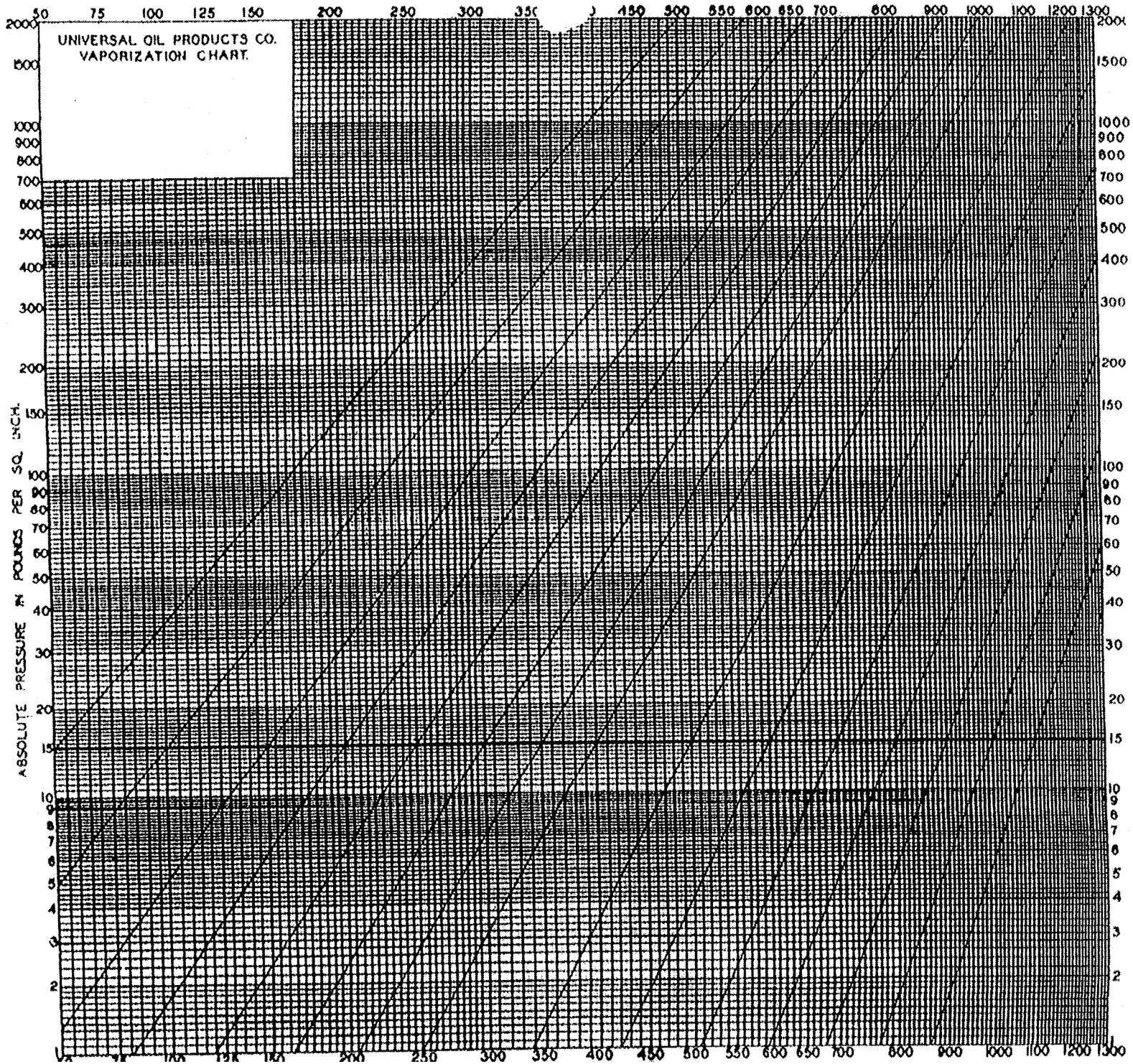
Crude Heater Duty (Computer Calculation) = 477.62- 194.43 = **283.19 MM Btu/h**

Error = 5.9%

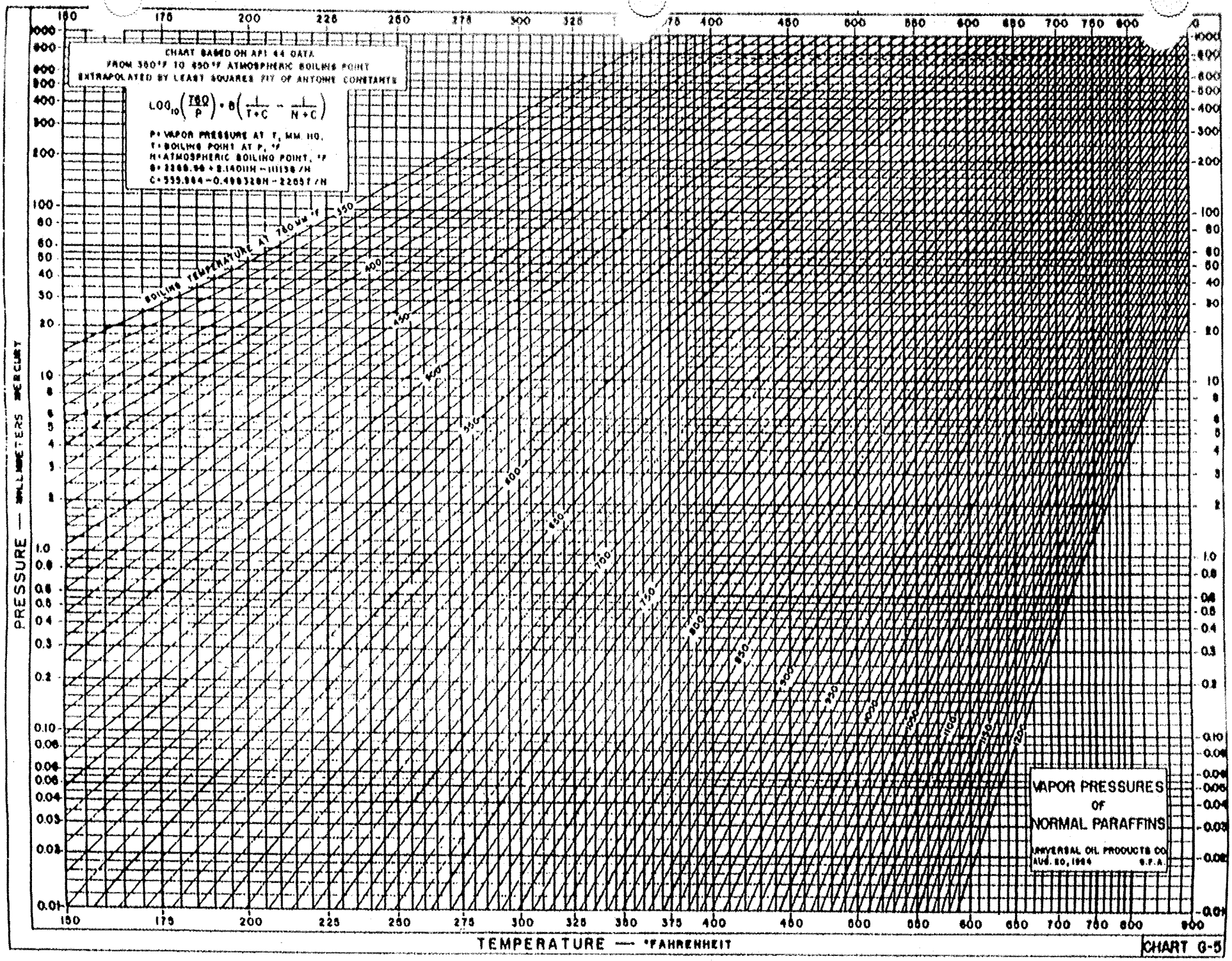

SPECIFIC HEAT OF LIQUID PETROLEUM OILS
WHERE K=11.8 (MIDCONTINENT STOCKS)
FOR OTHER STOCKS MULTIPLY BY CORRECTION FACTOR
SPECIFIC HEAT
1.0
0.9
0.8
0.7
0.6
0.5
0.4
70 A.P.I.
60
50
40
30
20
10
CORRECTION FACTOR FOR STOCKS WITH K≠OTHER THAN 11.8
1.05
1.00
0.95
0.90
10.0
11.0
12.0
K
100
200
300
400
500
600
700
800
900
1000
TEMPERATURE °F
PD 854


LATENT HEAT OF VAPORIZATION
OF
HYDROCARBONS
(ATMOSPHERIC PRESSURE)
°A.P.I.
MOLECULAR WEIGHT
LATENT HEAT OF VAPORIZATION-BTU/LB.
BOILING POINT °F
°A.P.I.


SPECIFIC HEAT
CORRECTION FACTOR FOR NON-PARAFFINIC GASES
1.00
0.90
0.80
DO NOT USE THIS FACTOR FOR PETROLEUM VAPORS
0 20 40 60 80 100
MOL % OLEFINS IN GAS
SPECIFIC HEAT OF PARAFFIN GASES ATMOSPHERIC PRESSURE
GAS SP. GR.
0.55
0.60
0.70
0.80
1.00
1.25
1.50
2.00
A.P.I.
70°
50°
30°
20°
10°
0°
METHANE
BUTANE
USE 70 LINE FOR HIGHER GRAVITIES
SPECIFIC HEAT OF PETROLEUM OIL VAPORS WITH K=11.8 (MIDCONTINENT STOCKS) ATMOSPHERIC PRESSURE
CORRECTION FACTOR FOR STOCKS OTHER THAN K=11.8
1.10
1.00
0.90
0.80
0.70
DO NOT USE THIS FACTOR FOR PARAFFIN GASES
9.5 10 11 12 13
K
TEMPERATURE F
100 200 300 400 500 600 700 800 900 1000 1100
1.0 .9 .8 .7 .6 .5 .4 .3
PD855


UNIVERSAL OIL PRODUCTS CO.
VAPORIZATION CHART.
ABSOLUTE PRESSURE IN POUNDS PER SQ. INCH.

PD-250


CHART BASED ON API 44 DATA
FROM 380°F TO 850°F ATMOSPHERIC BOILING POINT
EXTRAPOLATED BY LEAST SQUARES FIT OF ANTOINE CONSTANTS
$\log_{10}\left(\frac{760}{P}\right) = B\left(\frac{1}{T+C} - \frac{1}{N+C}\right)$
P = VAPOR PRESSURE AT T, MM. HG.
T = BOILING POINT AT P, °F
N = ATMOSPHERIC BOILING POINT, °F
B = 3868.98 + 8.140IIN − 111138/N
C = 559.864 − 0.498328N − 22057/N
BOILING TEMPERATURE AT 760 MM °F
PRESSURE — MILLIMETERS MERCURY
TEMPERATURE — °FAHRENHEIT
VAPOR PRESSURES
OF
NORMAL PARAFFINS
UNIVERSAL OIL PRODUCTS CO.
AUG. 20, 1944 U.S.A.
CHART G-5

Training Services

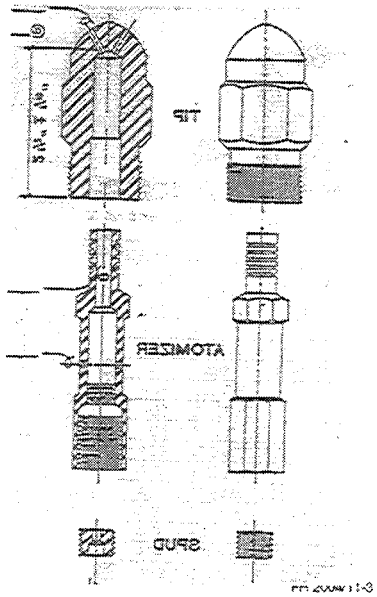
Troubleshooting Oil Firing

Section Outline

- Viscosity
- Fuel Oil Properties
- Oil Gun Location
- Fuel Oil Pressure
- Steam Pressure Differential

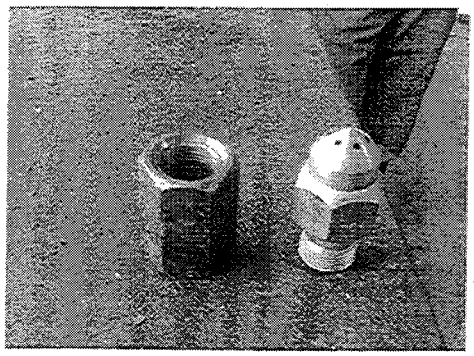
Oil Gun

- Mechanical atomization in oil spud
- 0.3 lb steam/lb oil enters side ports of atomizer and shears oil
- 100 micron liquid oil particles exit through oil tip



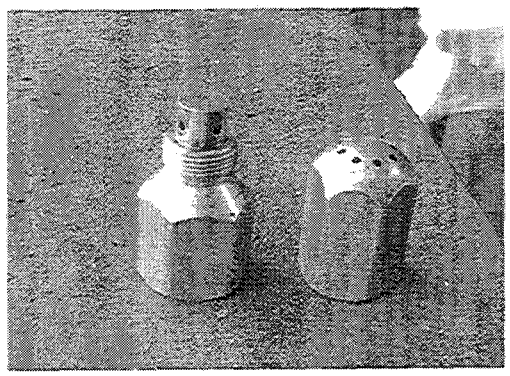
Oil Gun

- Zink No.9 EA Gun with 8 firing ports
- Standard Design



Oil Gun

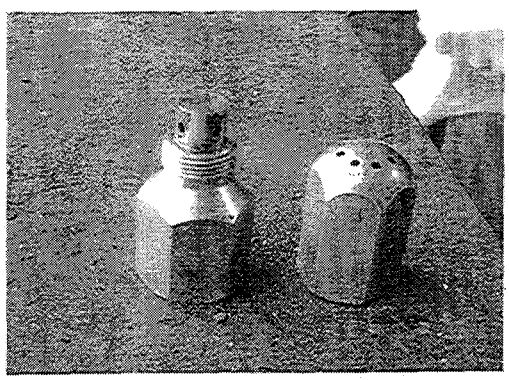
- Zink No.9 MEA Gun with 8 firing ports
- Improved Design
- Better Atomization and Combustion



FH 2004/11-5

Oil Gun

- Zink No.9 MEA Gun with 8 firing ports
- Improved Design
- Better Atomization and Combustion



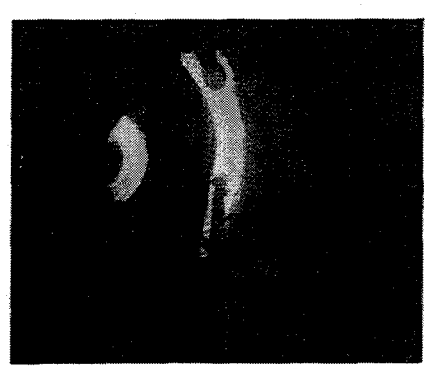
FH 2004/11-6

Viscosity

- Viscosity of the fuel oil is the most important factor in its successful combustion
- Too cold an oil will have too high a viscosity, resulting in poor atomization
- Large oil droplets do not burn completely, leaving "fireflies" in the flame

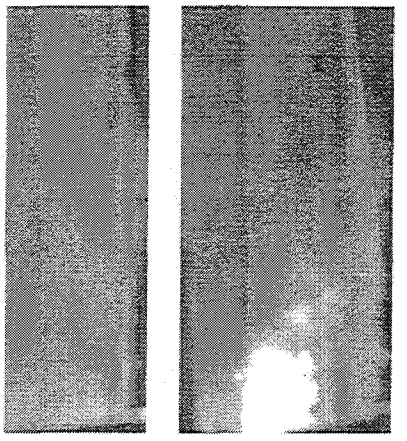
Oil Fireflies

- Poor Combustion
- Damage to tubes
- Particulate
- Cause- Cold oil or Wet Steam



Oil Fireflies

- Poor Combustion
- Damage to tubes
- Particulate
- Cause- Cold oil or Wet Steam



Viscosity
(continued)

- Heavy oils require higher temperatures to get proper viscosity
  - Required viscosity - 100 SSU (20 cS) typical

Viscosity
(continued)

- Procedure to determine proper fuel oil temperature
  - Obtain one point viscosity at 100-130°F (38-54°C) for No. 2 oils or at 210°F (100°C) for No. 6 oils
  - Plot point on viscosity-temperature chart; draw line parallel to other lines
  - Determine proper operating temperature for 20 cS (100 SSU)

Fuel Oil Properties

- Flash Point
- Pour Point
- Heat of Combustion
- Ash Content
- Viscosity vs. Temperature
- Sulfur and Nitrogen Content
- Heavy Metals Content (Sodium, Vanadium, Arsenic, Mercury, etc.)

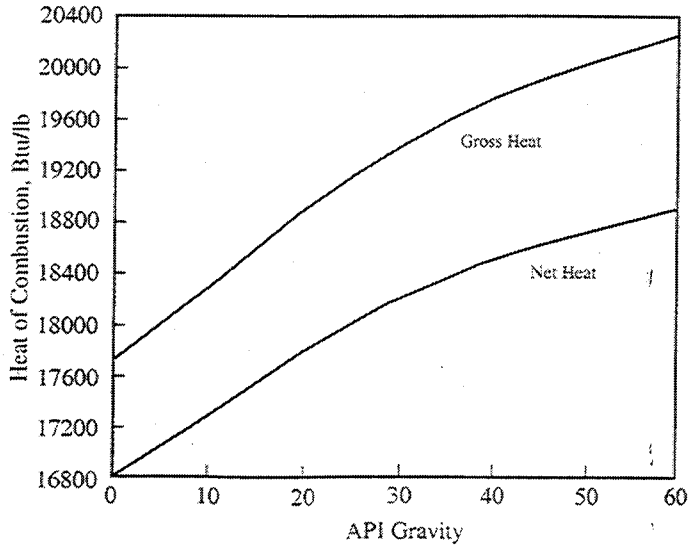
Fuel Oil Properties

Property	ASTM Test Method	No.1	No.2	Grade No. 4 (Light)	No. 5 (Light)	No. 4	No. 5 (Heavy)	No. 6
Flash Point °C, min	D93	38	38	38	55	55	55	60
Water and sediment, vol%, max	D2709	0.05	0.05	-----	-----	-----	-----	-----
	D95+D473	-----	-----	(0.50)	(1.00)	(0.50)	(1.50)	(1.00)
Distillation temperature °C	D86							
10% vol. recovered, max		215	-----	-----	-----	-----	-----	-----
90% vol. recovered min		-----	282	-----	-----	-----	-----	-----
90% vol recovered, max		288	388	-----	-----	-----	-----	-----
Kenematic viscosity at 40°C, mm%	D445							
min		1.3	1.9	1.9	-----	≥5.5	-----	-----
max		2.1	3.4	3.5	-----	24.0	-----	-----
Kenematic viscosity at 100°C, mm%	D445							
min		-----	-----	-----	5.0	-----	9.0	15.0
max		-----	-----	-----	8.9	-----	14.9	50.0
Ramsbottom carbon residue on 10% distillation residue, % mass, max	D524	0.15	0.35	-----	-----	-----		
Ash, % mass, max	D482	-----	-----	0.05	0.15	0.10	0.15	-----
Sulfur, % mass, max	D129	0.50	0.50	-----	-----	-----	-----	-----
Copper strip corrosion rating, max 3 h at 50°C	D130	No.3	No.3	No.3	-----	-----	-----	-----
Density at 15°C, kg/m³	D1298	-----	-----	-----	-----	-----	-----	-----
min		-----	-----	≥876	-----	-----	-----	-----
max		850	876	-----	-----	-----	-----	-----
Pour Point, °C, max	D97	-18	-6	-6	-----	-6	-----	-----

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FH 2004/11-13

Fuel Oil Heat of Combustion



FH 2004/11-14

Fuel Oil Viscosity

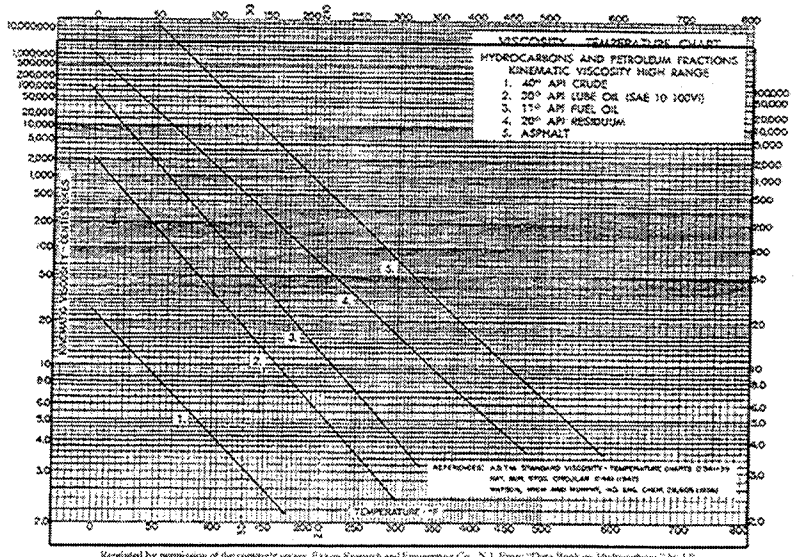
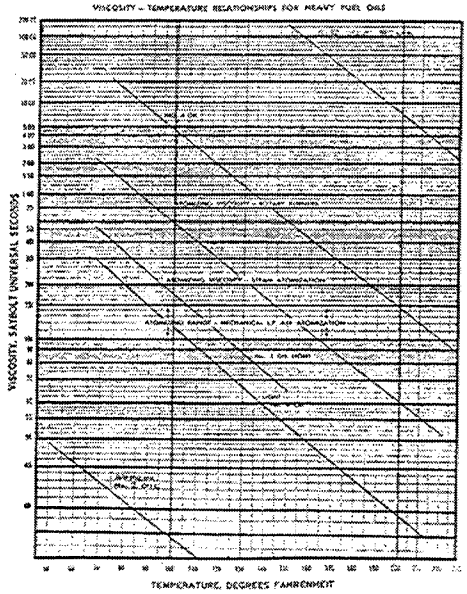


FIGURE P-6.3A

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FH 2004/11-15

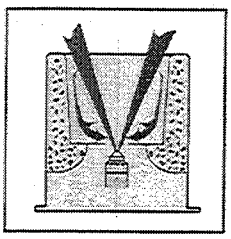
Fuel Oil Viscosity



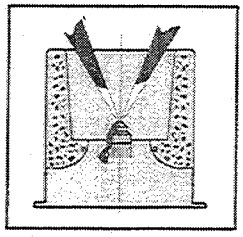
FH 2004/11-16

Oil Gun Location

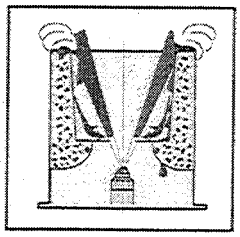
- Proper position is critical to proper spray pattern and air mixing



Proper location



Too far in

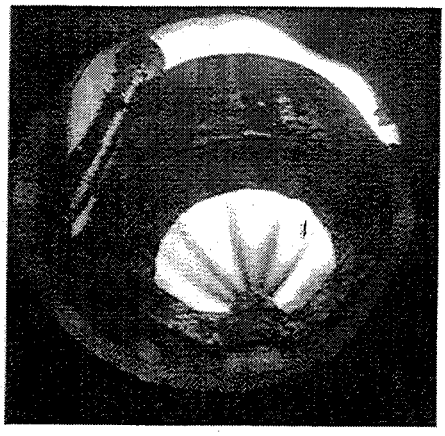


Too far back

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Proper Oil Gun Position

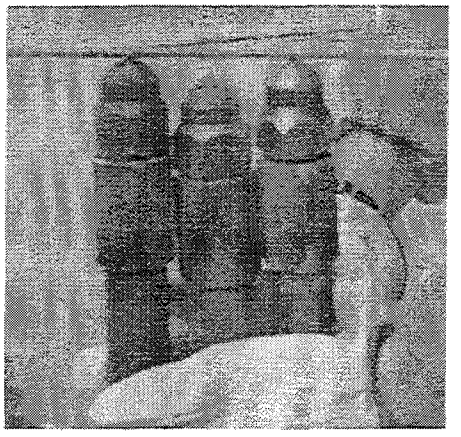
- Zink No.9 EA Gun with 8 firing ports
- Approximately 1" (25mm) below tile ledge
- Combustion occurs 2" (50mm) above tip
- Flame recirculation



FH 2004/11-18

Proper Oil Gun Length

- Zink No.9 EA Gun with 8 firing ports
- Oil Gun Inserts are different Lengths
- Impossible for operator to position properly
- Flame impingement, coking and oil drips result



FH 2004/11-19

Troubleshooting

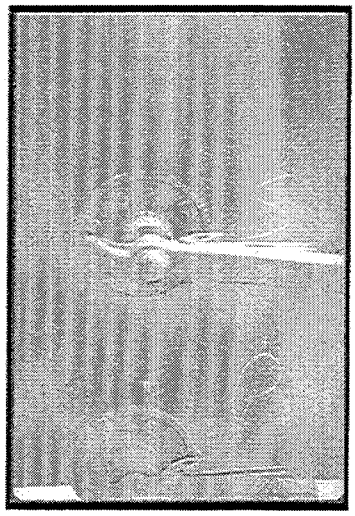
- Leaking Oil
- Burners Need Cleaning
- Improper Oil Temperature at Burner
- Poor Damper Control with Wing Nut
- Cannot Set Same Air Rate to Each Burner



FH 2004/11-20

Troubleshooting

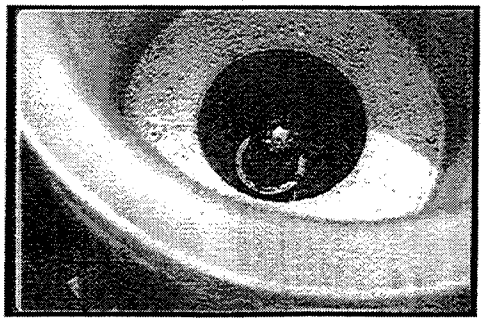
- Good Damper Control
- Positive Position Click Style
- Individual Primary and Secondary Controls on Combination Burners



FH 2004/11-21

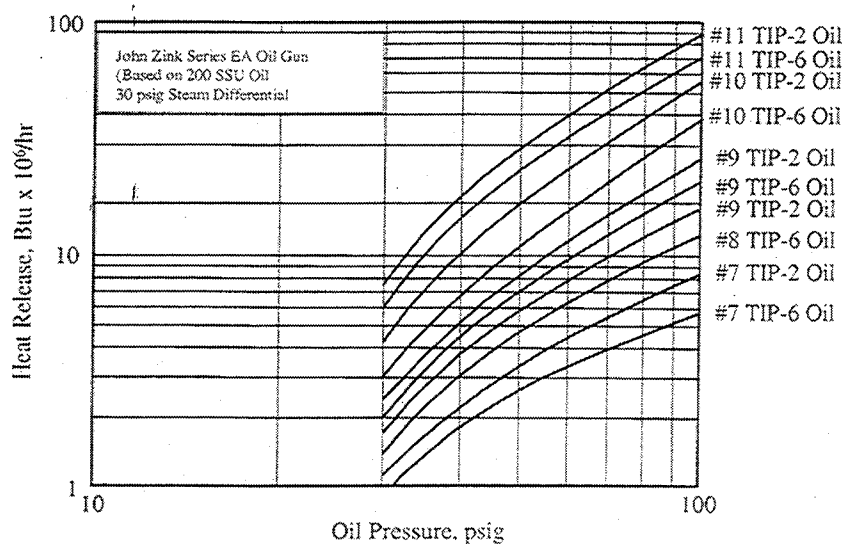
Troubleshooting

- Light Oil Gun Off Gas Flame
- Reduces Heater Fouling
- Oil requires Heat and Temperature to Vaporize
- Vacuum Bottoms in No.6 Oil has boiling range 1000-1500°F (540-815°C)



FH 2004/11-22

Oil Gun Heat Release



FH 2004/11-23

Fuel Oil Pressure

- **Design should be 70-90 psig (5-6.3 kg/cm^{2}g)**
- **Too low pressure can result in poor turndown**
- **Poor atomization**
- **Plugging (coke formation) in oil guns**

FH 2004/11-24

Flameout

- Low Oil Pressure
- High Steam Pressure
- Particulate
- Fouling on tubes

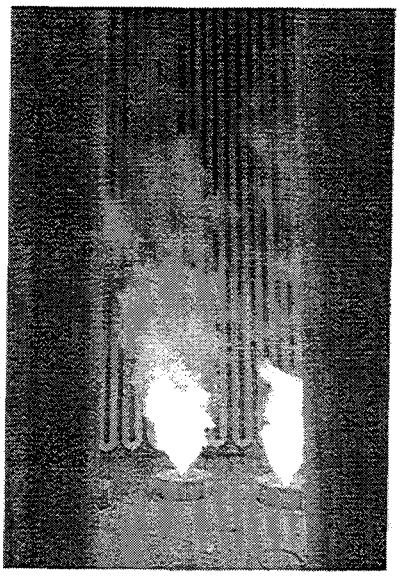


Steam Pressure Differential

- Poor atomization if too low or high
- Typical 30 psi (2.1kg/cm^2) differential above oil pressure
- Steam must be dry
- Wet steam causes fireflys

Steam Pressure Differential

- Low Steam Pressure Differential due to high pressure drop in steam check valve
- Poor Atomization
- Particulate
- Fouling on tubes



FH 2004/11-27

Fuel Oil Strainers

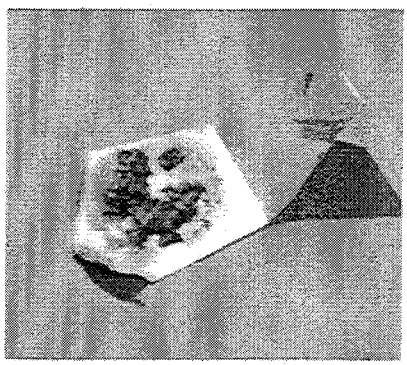
- Locate dual strainers at each heater
- Use large capacity basket type, not Y-type as shown



FH 2004/11-28

Fuel Oil Strainers

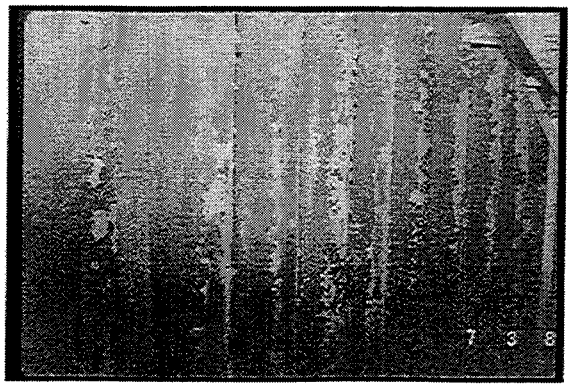
- Crud caught in strainer after tank switch
- Use stainers at each heater in addition to main fuel strainers located at fuel oil pumps



FH 2004/11-29

Troubleshooting

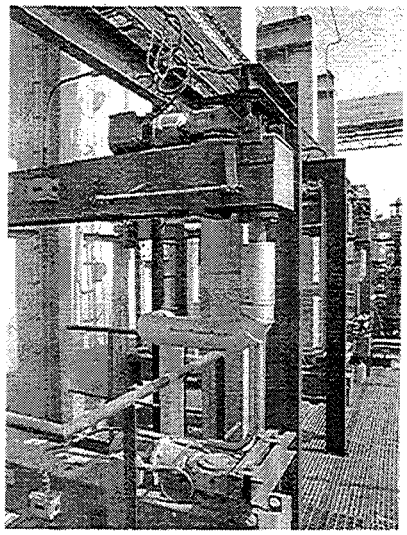
- Oil Ash acts as insulation
- Increased fuel firing
- Sandblast clean



FH 2004/11-30

Retractable Sootblowers

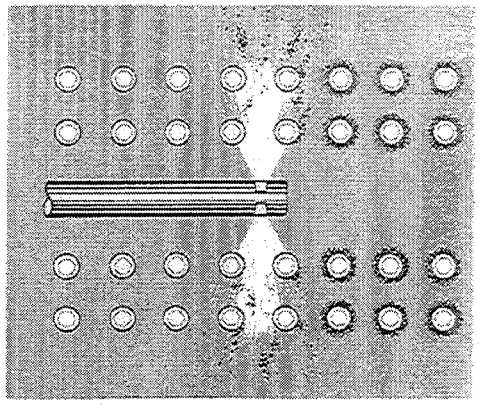
- Retractable sootblowers
- Operated once or twice per day
- Need dry steam to prevent refractory damage
- Steam blowdown before blowing
- Insulate steam lines to prevent condensation



FH 2004/11-31

Retractable Sootblowers

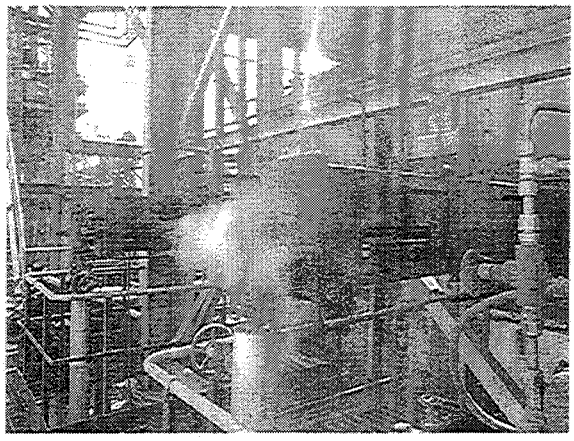
- Lance rotates in a helical motion
- Blowing Rate ~ 10,000 lb/hr
- Blowing Time 20-30 seconds



FH 2004/11-32

Retractable Sootblowers

- **Poppet steam valve leaks are common**
- **Diamond Power 1K sootblowers are recommended**



FH 2004/11-33

True Cost of Oil Firing

- The LP model often shows a very big differential between gas fuel values and oil fuel values. Therefore, there is a large economic incentive to increase the amount of oil firing. However, there are many hidden costs associated with the firing of fuel oil. These costs include;
- 0.3 kg steam consumed in the oil gun per 1 kg oil burnt
- steam consumed in the daily sootblowing
- steam consumed in oil line tracing
- 5-7% fuel efficiency loss due to convection fouling in the crude heaters resulting in high EII energy index
- online radiant section ammonium nitrate cleaning cost
- tube support replacement cost due to high firebox temperatures from oil fouling and oil ash corrosion
- convection tubesheet replacement due to failures caused by sootblowing with wet steam
- high sootblower maintenance cost

FH 2004/11-34

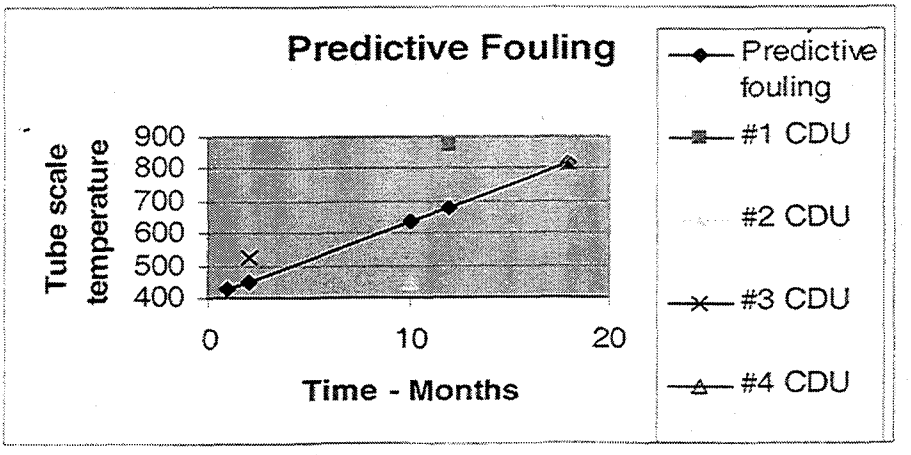
True Cost of Oil Firing

- yearly primary oil tile replacement due to oil ash corrosion
- cost associated with replacing oil gun internals every 1-2 years
- high labor cost associated with weekly oil gun cleaning
- decreased refractory life and heater casing damage due to sulfur corrosion from oil firing
- inventory cost of maintaining oil gun replacement parts and burner tile replacement parts in warehouse
- increased SOx and NOx emissions

- **Use part of fuel savings to finance new burners and parts**

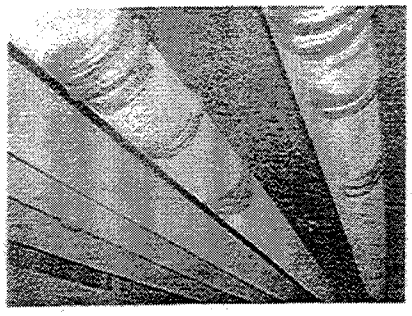
Plot Rate of Fouling

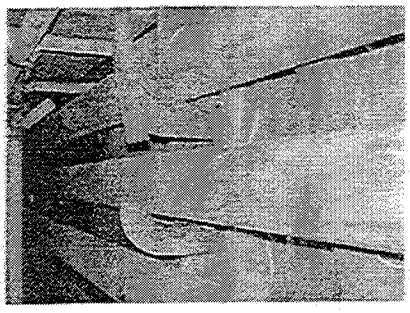
- **On-line clean every 9-18 months**



Chemical Cleaning

- CTP offline phosphoric acid tube cleaning
- Very effective in removing ash deposits
- CTP(phbarre@ctp-environnement.com) Phone 33-139-19-15-50





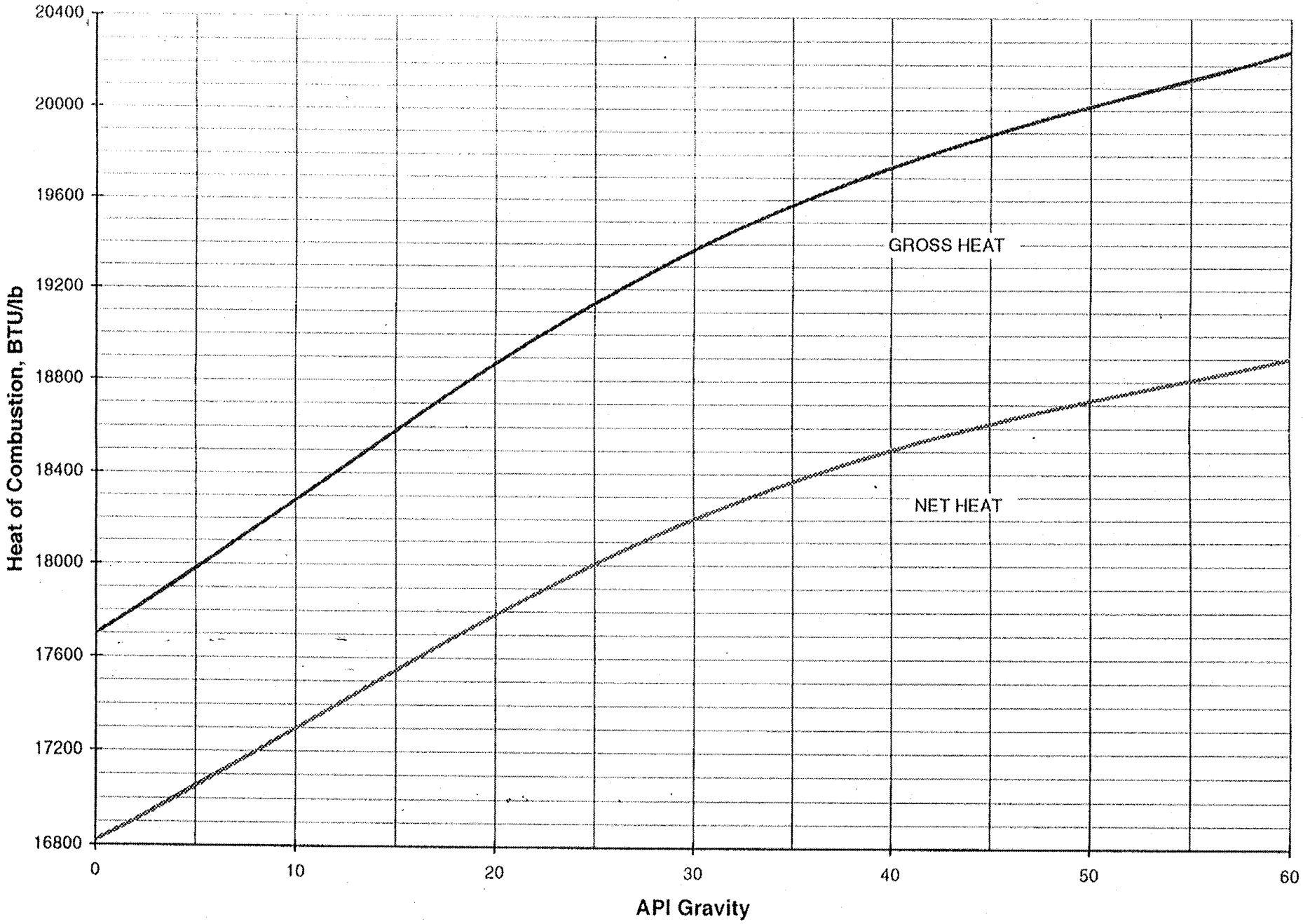
FH 2004/11-37

Detailed Requirements for Fuel Oils (ASTM D 396)

Property	ASTM Test Method	No. 1	No. 2	Grade No. 4 (Light)	No. 4	No. 5 (Light)	No. 5 (Heavy)	No. 6
Flash Point °C, min	D 93	38	38	38	55	55	55	60
Water and sediment, vol%, max	D 2709	0.05	0.05	---	---	---	---	---
	D95 + D 473	---	---	(0.50)	(0.50)	(1.00)	(1.00)	(1.00)
Distillation temperature °C	D86							
10% vol. recovered, max		215	---	---	---	---	---	---
90% vol. recovered, min		---	282	---	---	---	---	---
90% vol. recovered, max		288	338	---	---	---	---	---
Kinematic viscosity at 40°C, mm^2/s	D 445							
min		1.3	1.9	1.9	>5.5	---	---	---
max		2.1	3.4	5.5	24.0	---	---	---
Kinematic viscosity at 100°C, mm^2/s	D 445							
min		---	---	---	---	5.0	9.0	15.0
max		---	---	---	---	8.9	14.9	50.0
Ramsbottom carbon residue on 10% distillation residue, % mass, max	D 524	0.15	0.35	---	---	---	---	---
Ash, % mass, max	D 482	---	---	0.05	0.10	0.15	0.15	---
Sulfur, % mass, max	D 129	0.50	0.50	---	---	---	---	---
Copper strip corrosion rating, max 3 h at 50°C	D 130	No.3	No. 3	---	---	---	---	---
Density at 15°C, kg/m^3	D 1298	---	---	---	---	---	---	---
min		---	---	>876	---	---	---	---
max		850	876	---	---	---	---	---
Pour Point, °C, max	D 97	-18	-6	-6	-6	---	---	---

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Fuel Oil Heat of Combustion



The viscosity of some Non-Newtonian liquids is dependent upon time as well as shear rate. In other words, the viscosity at any particular time depends upon the amount of previous agitation or shearing of the liquid. A liquid whose viscosity decreases with time at a given shear rate is called a thixotropic liquid. Examples are asphalts, glues, molasses, paint, soap, starch, and grease. Liquids whose viscosity increases with time are called rheopectic liquids, but they are seldom encountered in pumping applications.

There are two basic viscosity parameters: dynamic (or absolute) viscosity and kinematic viscosity. Dynamic viscosities are given in terms of force required to move a unit area a unit distance. This is usually expressed in pound-seconds per square foot in the English system which is equal to slugs per foot-second. The Metric system is more commonly used, however, in which the unit is the dyne-second per square centimeter called the Poise. This is numerically equal to the gram per centimeter-second. For convenience, numerical values are normally expressed in centipoise, which are equal to one-hundredth of a poise.

Most pipe friction charts and pump correction charts list kinematic viscosity. The basic unit of kinematic viscosity is the stoke which is equal to a square centimeter per second Metric system. The corresponding English unit is square fo second. The centistoke which is one-hundredth of a st normally used in the charts. The following formula is us obtain the kinematic viscosity when the dynamic or abs viscosity is known:

$$\text{centistokes} = \frac{\text{centipoise}}{\text{sp. gr.}}$$

There are numerous types of viscometers available for d mining liquid viscosities, most of which are designed for sp liquids or viscosity ranges. The Saybolt viscometers are pro the most widely used in the United States. The Saybolt Univ Viscometer measures low to medium viscosity, and the Sa Furol Viscometer measures high viscosities. The correspo units are the SSU (Seconds Saybolt Universal) and the (Seconds Saybolt Furol.) These units are found on most friction and pump correction charts in addition to centistok conversion chart for these and other units is shown in Fi

17D-2A Viscosity Conversion Table

The following table will give an approximate comparison of various viscosity ratings so that if the viscosity is given in terms other t Saybolt Universal, it can be translated quickly by following horizontally to the Saybolt Universal column.

Seconds Saybolt Universal SSU	Kinematic Viscosity Centistokes	Seconds Saybolt Furol SSF	Seconds Redwood 1 (Standard)	Seconds Redwood 2 (Admiralty)	Degrees Engler	Degrees Barbey	Seconds Parlin Cup #7	Seconds Parlin Cup #10	Seconds Parlin Cup #15	Seconds Parlin Cup #20	Seconds Ford Cup #3	Secon Ford Cup
31	1.00	–	29	–	1.00	6200	–	–	–	–	–	–
35	2.56	–	32.1	–	1.16	2420	–	–	–	–	–	–
40	4.30	–	35.2	5.10	1.31	1440	–	–	–	–	–	–
50	7.40	–	44.3	5.83	1.58	838	–	–	–	–	–	–
60	10.3	–	52.3	6.77	1.88	518	–	–	–	–	–	–
70	13.1	12.95	60.9	7.60	2.17	483	–	–	–	–	–	–
80	15.7	13.70	69.2	8.44	2.45	404	–	–	–	–	–	–
90	18.2	14.44	77.6	9.30	2.73	348	–	–	–	–	–	–
100	20.6	15.24	85.6	10.12	3.02	307	–	–	–	–	–	–
150	32.1	19.30	128	14.48	4.48	195	–	–	–	–	–	–
200	43.2	23.5	170	18.90	5.92	144	40	–	–	–	–	–
250	54.0	28.0	212	23.45	7.35	114	46	–	–	–	–	–
300	65.0	32.5	254	28.0	8.79	95	52.5	15	6.0	3.0	30	20
400	87.60	41.9	338	37.1	11.70	70.8	66	21	7.2	3.2	42	28
500	110.0	51.6	423	46.2	14.60	56.4	79	25	7.8	3.4	50	34
600	132	61.4	506	55.4	17.50	47.0	92	30	8.5	3.6	58	40
700	154	71.1	592	64.6	20.45	40.3	105	35	9.0	3.9	67	45
800	176	81.0	677	73.8	23.35	35.2	120	39	9.8	4.1	74	50
900	198	91.0	762	83.0	25.30	31.3	135	41	10.7	4.3	82	57
1000	220	100.7	896	92.1	29.20	28.2	149	43	11.5	4.5	90	62
1500	330	150	1270	138.2	43.80	18.7	–	65	15.2	6.3	132	90
2000	440	200	1690	184.2	58.40	14.1	–	86	19.5	7.5	172	118
2500	550	250	2120	230	73.0	11.3	–	108	24	9	218	147
3000	660	300	2540	276	87.60	9.4	–	129	28.5	11	258	172
4000	880	400	3380	368	117.0	7.05	–	172	37	14	337	230
5000	1100	500	4230	461	146	5.64	–	215	47	18	425	290
6000	1320	600	5080	553	175	4.70	–	258	57	22	520	350
7000	1540	700	5920	645	204.5	4.03	–	300	67	25	600	410
8000	1760	800	6770	737	233.5	3.52	–	344	76	29	680	465
9000	1980	900	7620	829	263	3.13	–	387	86	32	780	520
10000	2200	1000	8460	921	292	2.82	–	430	96	35	850	575
15000	3300	1500	13700	–	438	2.50	–	650	147	53	1280	860
20000	4400	2000	18400	–	584	1.40	–	860	203	70	1715	1150

Fig. 4A

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*Kinematic Viscosity (in centistokes) = Absolute Viscosity (in centipoises) / Density

When the Metric System terms centistokes and centipoises are used, the density is numerically equal to the specific gravity. Therefore, the following expression can be used which will be sufficiently accurate for most calculations:

*Kinematic Viscosity (in centistokes) = Absolute Viscosity (in centipoises) / Specific Gravity

When the English System units are used, the density must be used rather than the specific gravity.

For values of 70 centistokes and above, use the following conversion:

SSU = centistokes × 4.635

Above the range of this table and within the range of the viscosimeter, multiply the particular value by the following approximate factors to convert to SSU:

Viscosimeter	Factor	Viscosimeter	Factor
Saybolt Furol	10.	Parlin cup #15	98.2
Redwood Standard	1.095	Parlin cup #20	187.0
Redwood Admiralty	10.87	Ford cup #4	17.4
Engler—Degrees	34.5		

17D-2B Viscosity Conversion Table

The following table will give an approximate comparison of various viscosity ratings so that if the viscosity is given in terms other than Saybolt Universal, it can be translated quickly by following horizontally to the Saybolt Universal column.

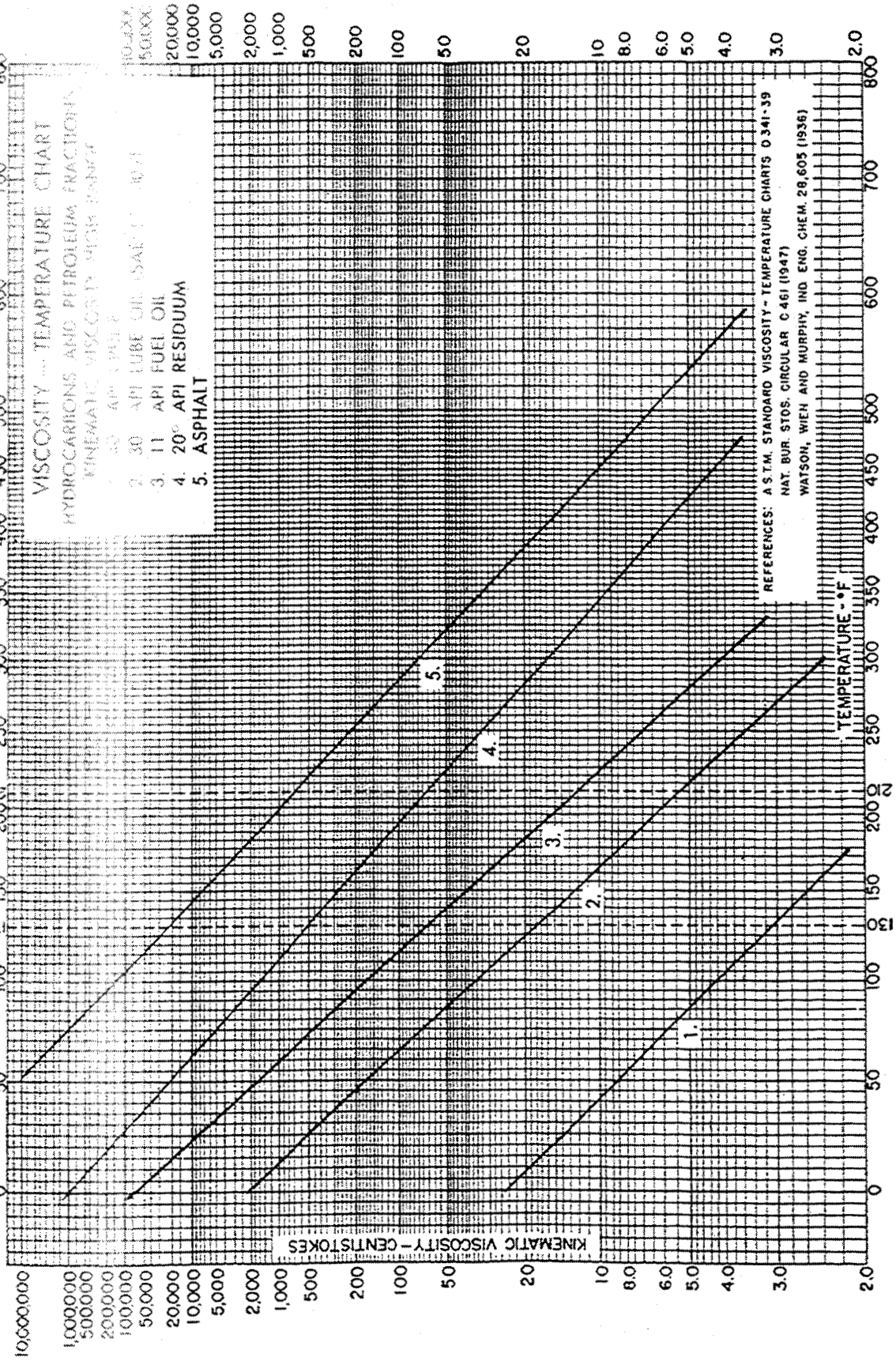
Seconds Saybolt Universal SSU	Kinematic Viscosity Centistokes*	Approx. Seconds Mac Michael	Approx. Gardner Holt Bubble	Seconds Zahn Cup #1	Seconds Zahn Cup #2	Seconds Zahn Cup #3	Seconds Zahn Cup #4	Seconds Zahn Cup #5	Seconds Demmler Cup #1	Seconds Demmler Cup #10	Approx. Seconds Stormer 100 gm Load	Seconds Pratt and Lambert "F"
31	1.00	—	—	—	—	—	—	—	—	—	—	—
35	2.56	—	—	—	—	—	—	—	—	—	—	—
40	4.30	—	—	—	—	—	—	—	1.3	—	—	—
50	7.40	—	—	—	—	—	—	—	2.3	—	2.6	—
60	10.3	—	—	—	—	—	—	—	3.2	—	3.6	—
70	13.1	—	—	—	—	—	—	—	4.1	—	4.6	—
80	15.7	—	—	—	—	—	—	—	4.9	—	5.5	—
90	18.2	—	—	—	—	—	—	—	5.7	—	6.4	—
100	20.6	125	—	38	18	—	—	—	6.5	—	7.3	—
150	32.1	145	—	47	20	—	—	—	10.0	1.0	11.3	—
200	43.2	165	A	54	23	—	—	—	13.5	1.4	15.2	—
250	54.0	196	A	62	26	—	—	—	16.9	1.7	19	—
300	65.0	225	B	73	29	—	—	—	20.4	2.0	23	—
400	87.0	270	C	90	37	—	—	—	27.4	2.7	31	7
500	110.0	320	D	—	46	—	—	—	34.5	3.5	39	8
600	132	370	F	—	55	—	—	—	41	4.1	46	9
700	154	420	G	—	63	22.5	—	—	48	4.8	54	9.5
800	176	470	—	—	72	24.5	—	—	55	5.5	62	10.8
900	198	515	H	—	80	27	18	—	62	6.2	70	11.9
1000	220	570	I	—	88	29	20	13	69	6.9	77	12.4
1500	330	805	M	—	—	40	28	18	103	10.3	116	16.8
2000	440	1070	Q	—	—	51	34	24	137	13.7	154	22
2500	550	1325	T	—	—	63	41	29	172	17.2	193	27.6
3000	660	1690	U	—	—	75	48	33	206	20.6	232	33.7
4000	880	2110	V	—	—	—	63	43	275	27.5	308	45
5000	1100	2635	W	—	—	—	77	50	344	34.4	385	55.8
6000	1320	3145	X	—	—	—	—	65	413	41.3	462	65.5
7000	1540	3670	—	—	—	—	—	75	481	48	540	77
8000	1760	4170	Y	—	—	—	—	86	550	55	618	89
9000	1980	4700	—	—	—	—	—	96	620	62	695	102
10000	2200	5220	Z	—	—	—	—	—	690	69	770	113
15000	3300	7720	Z2	—	—	—	—	—	1030	103	1160	172
20000	4400	10500	Z3	—	—	—	—	—	1370	137	1540	234

Fig. 4B

Above the range of this table and within the range of the viscosimeter, multiply the particular value by the following approximate factors to convert to SSU:

Viscosimeter	Factor
Mac Michael	1.92 (approx.)
Demmler #1	14.6
Demmler #10	146.
Stormer	13. (approx.)

FIGURE P-5.3A



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Fuel Oil Viscosity

Many fuel oil firing problems occur because the fuel oil is too cold at the burner: The oil must be burned as a vapor to get a proper flame pattern. Cold oil results in larger oil particles which take longer to burn. This produces bad flames and particulate emissions.

In the past, the fuel oil temperature was set to hold 43 CS (200ssu) at the burner. Today refiners operate with 20 CS (100ssu) at the burner to obtain better atomization.

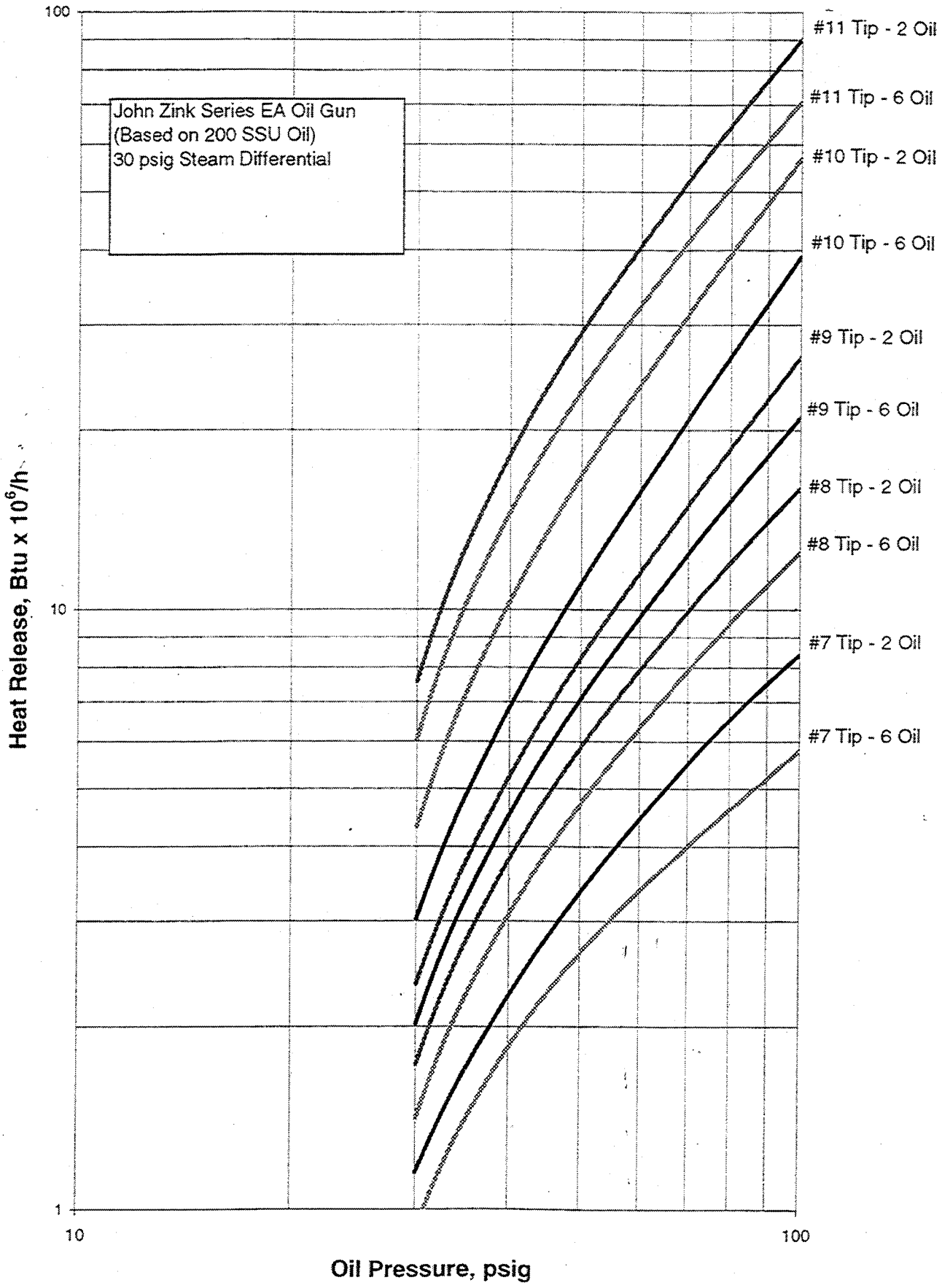
Obtain a sample of the fuel oil to be fired.

For a No.2 oil, obtain one point viscosity at 100-130°F (38-54°C). For a No. 6 oil, obtain one point viscosity at 210°F (100°C).

Plot the point on the viscosity- temperature chart. Draw a line parallel to the five lines. Determine the proper operating temperature at the heater to obtain 20 CS (100ssu).

Many errors occur when engineer use two points to plot the operating line, especially with heavy oils. The cold point tends to be too high of a viscosity. This results in an operating line with too steep of a slope.

Oil Gun Heat Release

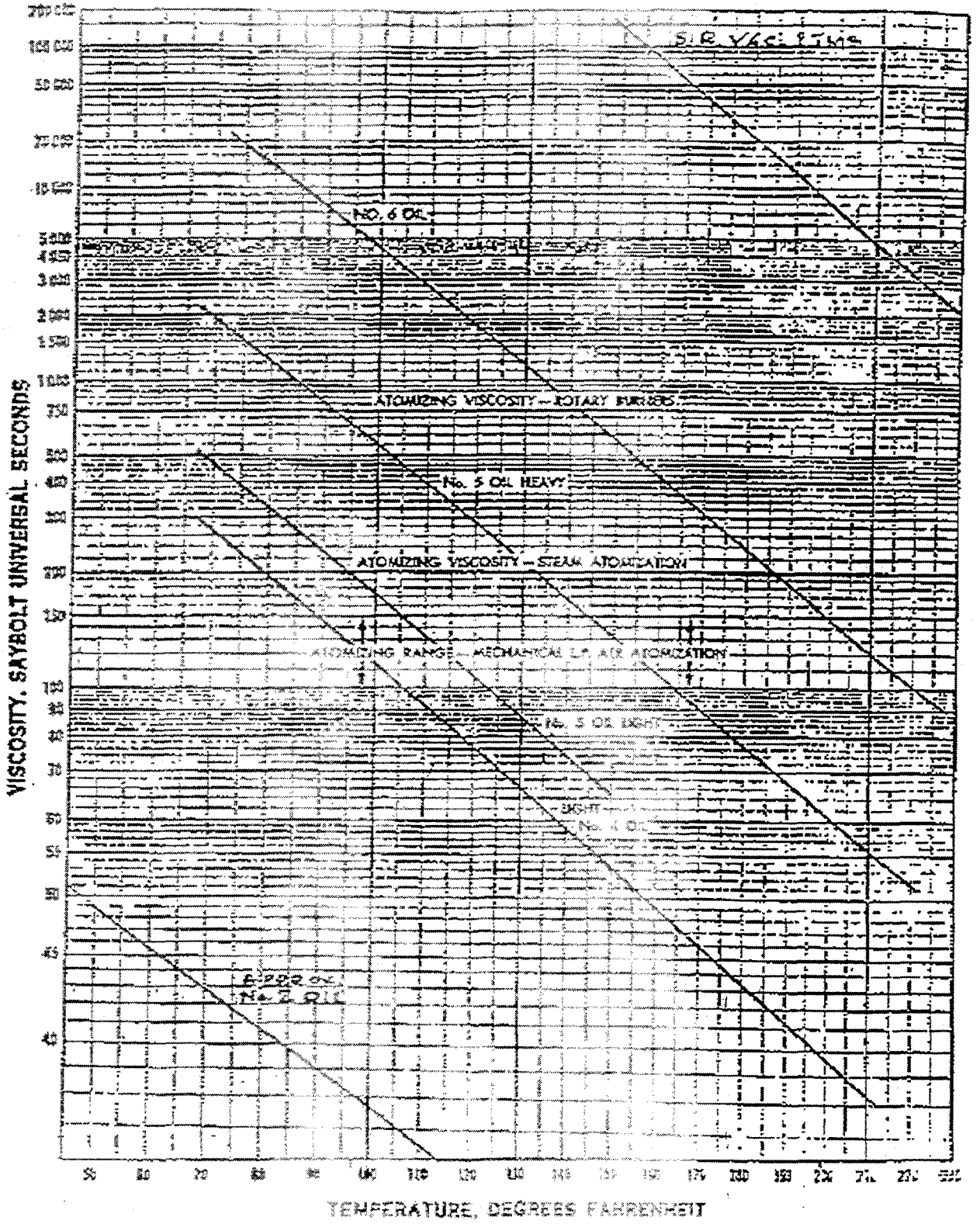


Importance of Fuel Oil Pressure

Fuel oil pressure to the burner is very important. The greater the fuel oil pressure, the greater the flexibility in burner operation. Atomization of fuel oil is obtained through the expenditure of energy. The energy may be supplied by steam suitably injected into the oil or it may be supplied by high oil pressure.

Oil burners will turndown to about 40-45 psig (3.0 Kg/Cm^2) oil pressure. As the oil pressure is lowered on the oil gun, atomization gets worse due to lower pressure. The oil particle sizes increase resulting in slower burning. The slower burning lifts the flame out of the primary tile. Pulsation will occur and the flame will go out.

VISCOSITY – TEMPERATURE RELATIONSHIPS FOR HEAVY FUEL OILS



Fuel Oil - Temperature versus Viscosity

Heaters can be operated just as efficiently burning fuel oil as burning gas; however, this is dependent on burner design and the quality of fuel oil burned.

A most important factor in fuel oil burning is viscosity at the burner. No matter what atomization system is utilized, the viscosity of the oil as atomized is quite critical. Changes in fuel oil viscosity can have a tremendous effect upon how well the oil is atomized or broken into fine particles thus having a tremendous effect upon the flame quality.

It should be noted at this time that the fuel oil burning systems vary quite widely as to their tolerance for viscosity of the oil as atomized. The greatest tolerance for viscosity is characteristic of steam atomization.

Normally, the heavier the fuel oil, the more potential trouble there is in burning it. In general, as the density of the fuel oil increases so does the viscosity of the fuel oil. To reduce the viscosity to a point at which it can be readily atomized requires heating the oil, in some cases to quite high temperatures. In many cases this can mean insulating and even steam tracing the oil lines.

The flash point of the fuel oil is important. If the flash point of the oil is too low, flashing will occur either at the burner or in the fuel tank. When this condition occurs, the proper operating temperature to achieve 200 SSU viscosity cannot be reached. Poor operations result from firing the cold oil.

When the oil is too cold, larger oil droplets are produced in the atomizer. The larger droplets of oil produce sparks or "fireflies" in the oil flame.

Burning Maintenance

Cleaning Oil Burner

Oil burner tips will accumulate coke and other foreign solid particles which will eventually plug the oil tip. The tips, therefore, should be cleaned occasionally to keep them operating at maximum efficiency.

1. Close oil firing valve and oil block valve to oil gun.

2. Open the steam purge valve and steam through the oil line and oil tube for five (5) minutes to clear the oil gun of oil.

3. Close steam purge, steam firing and steam block valves.

4. Disconnect oil and steam lines at oil gun.

5. Loosen clevis (receiver bolt), and remove oil gun from receiver.

CAUTION: Use asbestos mittens, as oil gun will be HOT.

6. Remove oil tip and remove all foreign material from the atomizing chamber and orifice holes.

7. Blow the steam and oil tube with air, then replace oil tip.

8. Replace the copper gaskets in oil body, on oil and steam inlets.

9. Place oil gun back in oil gun receiver and tighten clevis.

10. Hook up steam and oil lines, then open steam firing, steam purge and steam block valves to check for leaks.

11. Put burner back in service by following the burner lighting procedure.

Installation, Operation, Maintenance and Trouble Shooting

John Zink Series "EA" Oil Gun

The John Zink Series "EA" oil gun is a mixing chamber/dispersion nozzle-style oil gun. This type gun utilizes metering orifices for the oil and atomizing medium, entering the mixing chamber (atomizer body) and disperses the mixture into the combustion zone through a spray nozzle (oil tip).

Generally, the atomizing medium is steam or air and is required to be controlled at a 25 to 30 psig differential pressure above the fuel oil pressure. However, other forms of atomizing medium such as high pressure gas, or special pressure requirements may occur. Therefore, always check the John Zink burner capacity curve for the specific operating conditions.

Installation

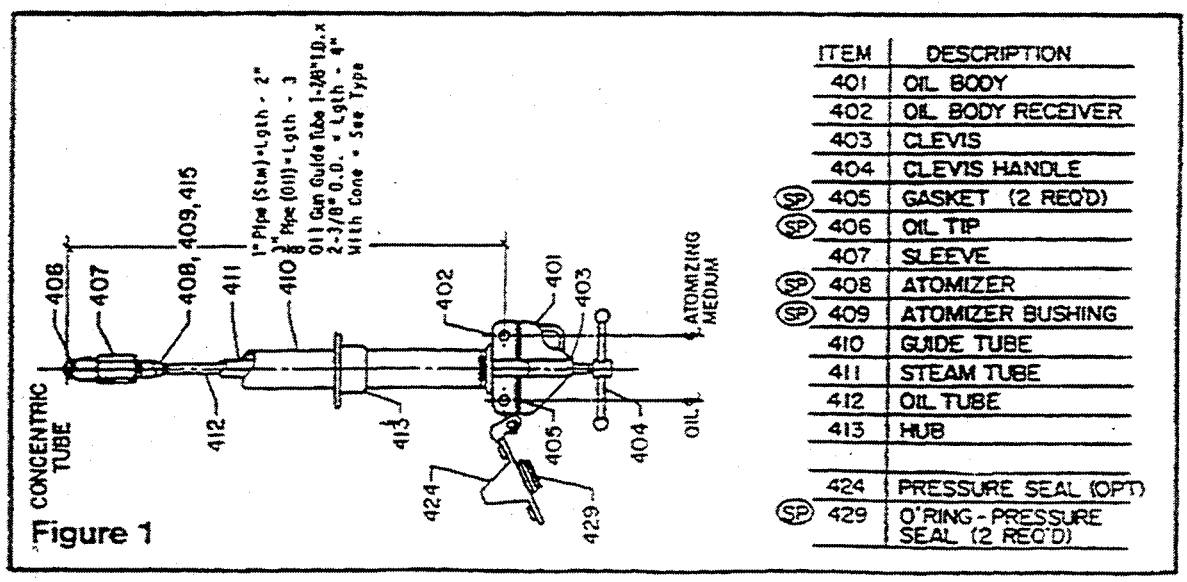
Prior to piping and light off of any John Zink oil gun, five (5) precautions should be observed.

1. Check oil tip part number (stamped an tip) against the burner drawing to ensure proper capacity and flame pattern.

NOTE: New equipment is shipped with a protective cap over the exit ports. This should be removed prior to light-off.

2. Check oil gun insert length against burner drawing (see Figure 1).

3. Position the oil gun oil and steam connections per the John Zink burner drawing using the three set screws on the oil gun hub (see Figure 1).



ITEM	DESCRIPTION
401	OIL BODY
402	OIL BODY RECEIVER
403	CLEVIS
404	CLEVIS HANDLE
(SP) 405	GASKET (2 REQ'D)
(SP) 406	OIL TIP
407	SLEEVE
(SP) 408	ATOMIZER
(SP) 409	ATOMIZER BUSHING
410	GUIDE TUBE
411	STEAM TUBE
412	OIL TUBE
413	HUB
424	PRESSURE SEAL (OPT)
(SP) 429	O'RING - PRESSURE SEAL (2 REQ'D)

Figure 1

4. Inspect and replace if necessary, the oil body gaskets to ensure proper interface seal.

5. Carefully review the John Zink burner assembly drawing to ensure proper oil and atomizing medium piping.

Specifically, two alignment pins, in the oil body receiver, require insertion of the oil body in only one way. If the oil and steam connections are reversed, the operation of the oil gun will be severely affected.

You are now ready to complete the oil and atomizing medium piping. As a general rule, all oil guns should be piped with a manually controlled purge crossover (see Figure 2).

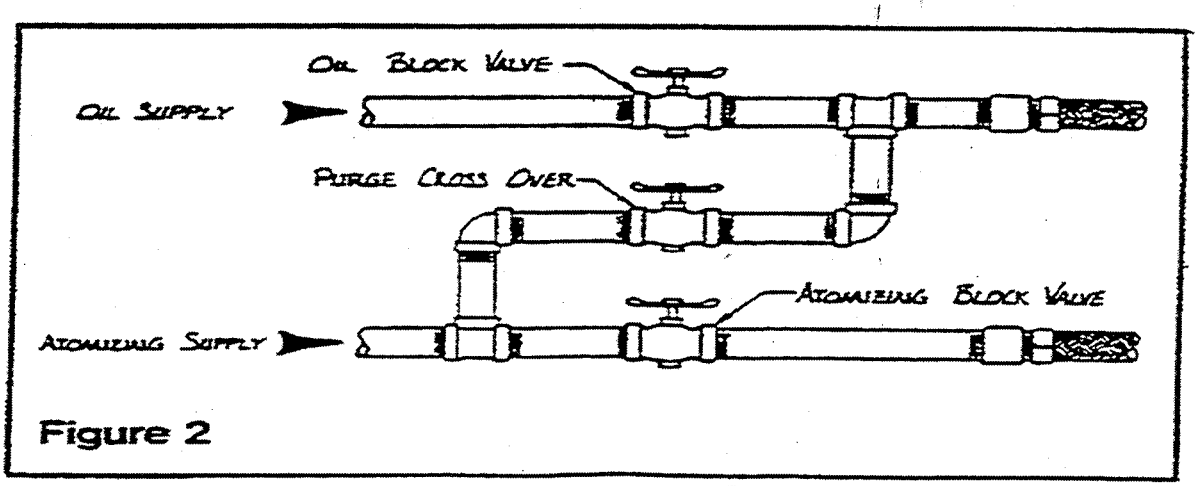


Figure 2

Operation

Three separate modes may be employed for oil gun light-off.

1. Manual ignition using an operator-inserted torch.

2. Direct ignition from gas pilot.

3. Direct ignition from low fire gas-flame.

Prior to attempting oil light-off, atomizing medium should be introduced into the oil gun both through the atomizing piping, and by use of the purge crossover valve, the oil piping. This will ensure the blow down of any water accumulation in the atomizing medium piping, provide visual proof of. atomizing medium flow, and, in the case of heavy oil firing, provide heat to the oil gun piping for the protection against chilling of the oil.

At this time, all oil firing permissives should be positive and oil made available to the final manual block valve.

Manual Ignition Using Operator-Supplied Torch

1A . Prepare a torch which will provide 12 - 24 inches of flame.

2A. Insert the torch into the burner and check for continued torch flame.

This torch may be inserted through the register(s), and/or the lighting and sighting ports of the burner to be lit. Its location can be either of two places.

a. Through the throat of the regen (oil) tile and freely playing across the oil tip.

-or-

b. Through the annular space created by the secondary tile and the oil stabilizer (either a diffuser cone or a regen tile) and freely playing at the edge of the stabilizer.

Direct Ignition From Gas Pilot

1B. Establish pilot flame. This may be achieved by use of a portable ignitor on manual pilots or electric are if electric ignition pilots are supplied.

2B. Adjust pilot flame to 18 to 24 inches long with slight yellow tails by use of pilot gas pressure and/or pilot air door (where supplied).

Direct Ignition from Low-Fire Gas Flame

1C. Establish main flame.

2C. Regulate main flame to approximately one-third (1/3) rate. Generally on process heater burners, this will be at 1 - 3 psig. (Review John Zink Company capacity curve for specific pressure requirements.)

3C. Close/open burner air register(s) to 50% open.

4C. Close purge crossover valve (see Figure 2).

5C. Open atomizing medium hand control valve (Figure 2) establishing proper baseline atomization control.

6C. Slowly open the oil manual block valve (Figure 2) until, oil ignites and oil flame stabilizes.

7C. At this time, the ignition source, either the manual torch or the pilot/main flame, when not required as standing ignition source, may be removed.

8C. Open the oil manual block valve full open placing oil/atomizing medium on control at low fire (see John Zink Company capacity curve for suggested minimum oil/atomizing medium operating pressures).

9C. Following procedures outlined in Steps 1 through 8, light all burners required.

10C. Adjust. burner operating pressures for required heat release and air register(s) or forced draft fan setting (where applicable) for required excess air.

At this time, small operational adjustments on oil tip locations may be required due to variations in oil, atomizing medium and oil tip wear. A full discussion of oil tip positioning in a regen style burner is supplied in the trouble-shooting section.

NOTE: At no time should an oil gun be operated above minimum firing rate, in an oil tile, when the primary air is obstructed, either by foreign materials or closed register(s).

Trouble-Shooting

In general, poor oil firing will exhibit six(6) problems to the field operator, either singly or in combination.

1. Smokey Flame
2. Fire Flies or Sparklers in the Firebox
3. Lazy or Uncontrollable Flame Pattern
4. Instability
5. Coke Formation
6. Oil Spills

Insufficient Air

Insufficient air in a process heater is often one of the most difficult problems to solve because it can be either the main problem whose solution is simply to increase the air/fuel ratio, or it may be. the result of any one or a combination of the other listed causes.

In the case of multiple burner operation, to address air deficiency as a cause, you must review the single burner operation with respect to the total heater operation. A single burner operating with insufficient air will still smoke even though the total firebox may exhibit sufficient excess air for completed combustion. This is due to the additive excess air from all other burners.

Specifically, an overall review of all burners, noting gas pressure on gas-fired burners, oil and atomization pressures on oil fired burners, position of hand block valves on gas, oil and atomization, and register settings should give a good indication of whether any particular burner is being operated at a capacity greater than would seem suitable for its air supply. Neglecting air. flow distribution in any forced draft system and minor tolerance differences in burner throat tile installations, all burners with equivalent fuel supplies should have approximately equivalent register settings. It should be noted that this superficial preliminary review is not valid unless all manual block valves are full open.

If the above review yields no major variations in fuel/air supply systems and air shortage is still suspected, a physical inspection of primary and secondary (and tertiary where applicable) air throats should be made to ensure no foreign materials are obstructing proper air flow. This inspection can usually be made through sight ports and/or register assemblies, however, in some cases, a probe may be required. If no obstruction is evident in the tile throat(s), inspection and manipulation of the air register assembly(s) is indicated to rule out register blockage.

Given the above conditions are shown to be clear of problem, the final step is to make a simple static pressure reading within the register assembly. This reading should entail no less than three(3) locations, one(1) of which should be made within the primary air supply zone, downstream from the primary air supply on a regen style burner and within six(6) inches of the back side of the diffuser in a diffuser-style burner. These static pressure readings are a valid

indicator of plenum air distribution and/or furnace draft variations and should be compensated for by register settings.

In some cases, spot checks of oxygen levels within the radiant box can also be used for register adjustment to achieve balanced, multiple burner operation.

Once the air supply side of the burner(s) in question has been determined to be balanced,, the fuel supply becomes questionable.

Lack of Atomizing Pressure

Check the John Zink capacity curve supplied with the burner. Each oil gun capacity curve should indicate both the type of oil used for design sizing and the suggested atomization pressure for that oil.

Generally, the lower viscosity oils require lower atomizing pressures than the higher viscosity oils. If a change in viscosity has been experienced either through a change in oil from design or a change in operating temperature, it is possible that a revision in atomization pressure may be required.

Since the "EA" series gun uses an internal mixing chamber, this type gun is subject to variations in oil flow at any single oil pressure. A small variation in atomizing medium supply pressure can significantly change the internal pressure of the mixing chamber. Therefore, low atomizing supply pressure will increase the net oil flow, increasing BTU input, thereby causing that burner to be "overfired" at a pressure which it normally would have sufficient-air.

Lack of Atomizing Flow

Pluggage of atomizer steam ports with pipe scale, dirt, and/or oil borne particulates will cause a reduction in atomizing medium flow, effectively reducing the mixing chamber back pressure, resulting in the same effect as low atomizing pressure.

A full discussion of disassembly and cleaning of the "EA" series oil gun is covered in the Maintenance Section.

Wet Steam

Specifically, on those burners designed to use steam as the atomizing medium, it should always be clean, dry steam. A suggestion of 10 - 20°F superheat should ensure this situation. However, all atomizing steam system should be trapped.

Since water is lower in energy than steam, atomization with wet steam is not as effective as with dry steam. This lower energy atomization will result in a larger droplet size and some oil coated water droplets being dispersed into the combustion zone. These larger droplets and water droplets are slower burning and can often be seen floating on the internal firebox currents. These "fireflies" are a major source for ash and soot buildup on radiant and convective tubes.

A second effect is that since this lower grade atomization is slower burning and accumulation (or sludging) of water is incombustible, burner stability may be severely reduced.

Cold Oil (Heavy Oil)

On those burners operating on heavy or viscous oils, the oil should be heated to a temperature sufficient to achieve 200 - 250 SSU. This suggested viscosity range is sufficient for proper atomization and any reduction in oil temperature will increase the oil's viscosity.

Viscosity is a measure of the oil's resistance to break up, and as the viscosity increases the quality of atomization and combustion will decrease.

Tip/Atomizer Failure

The "EA" series oil gun operation is dependent on a number of machined orifices, channels and seals. These pieces are subject to high velocity abrasive flows and corrosive action dependent on the type of oil fired.

Clearly these orifices, channels and seals are subject to some "normal" wear, making them a "maintenance item". Additionally, this condition is aggravated by the common, and some not so common, contaminants found in many oils. Coke or carbon particles, catalyst fines, and silica particles have a highly erosive action on metal parts when subjected to high pressure, high

velocity metering, while sulfur, chloride compounds and, in some cases, anhydrous acids will severely attack, through corrosion, the materials of the atomizer and dispersion nozzle.

The use of hardened tip and atomizer materials for erosive oils and 300 series stainless steel or higher for corrosive oils is suggested.

Some typical effects of tip and/or atomizer deterioration are follows:

1. Enlargement of oil orifice - high oil flow, low atomizing medium ratio, poor atomization, burner overfiring.

2. Enlargement of atomizing orifices - high atomizing medium flow, low oil flow, reduction in oil gun capacity, reduction in low fire stability.

3. Enlargement of atomizier exit - lowered mixing chamber pressure, reduced atomization quality, burner overfiring.

4. Deterioration of atomizer labrinth seal - steam bypassing of atomization chamber, poor atomization, instability, unsymmetrical flame patterns.

5. Deterioration of dispersion chamber - reduction of exit port L/D, deterioration of dispersion pattern, coking, oil spills.

6. Enlargement of exit ports – reduction of exit port L/D.

See the maintenance section for details of disassembly, inspection and cleaning of the oil gun.

Regan/Diffuser Failure

Commonly a failure in either the refractory of the regen tile or the metallurgy of a diffuser cone is the result of some other oil firing problem. However, it should be noted that these parts are integral and necessary to the proper function of their burner. Failure of these parts should be acted upon with replacement immediately.

As a secondary consideration, non-concentricity of these parts with respect to secondary tile throat and oil tip will cause poor air distribution, non-uniform flame patterns, coking and oil spills.

Low Oil Flow

Extreme reduction in oil firing rate and/or plugging of the oil orifice from pipe scale or oil borne contaminants can cause severe burner stability problems, while the lowered exit port velocities can cause dripping or internal oil spills. If for any reason the burner capacity requirements are reduced by any appreciable percentage, new reduced capacity oil guns are suggested.

Low Oil Pressure

See **Low Oil Flow** above.

High Atomizing Pressure

As discussed in **Lack of Atomizing Pressure** above, the converse is true.

High atomizing medium pressure will increase the mixing chamber back pressure thereby, reducing the oil flow. In many cases, this raised atomizing medium/oil ratio can cause severe stability problems.

Leaking Crossover Valve

Since the "EA" series oil gun is commonly operated with the atomizing medium at a higher pressure than the oil, a leaking purge crossover valve can cause severe disruption in oil flow to the oil gun and be detrimental to the atomization of the oil supplied.

The bypassing of atomizing medium into the oil supply is typically characterized by what is commonly called "Motor Boating". This continuous disruption of oil flow is clearly audible, thus deriving its name from the similar sounds.

High Steam Temperature

High temperatures on the atomizing medium cause problems in two separate ways, but these problems can be directly tied to the medium temperature.

Light oils often can be adjusted to very clear, yellow fires, however, this same fuel will often exhibit instability, pulsation in flow, clear blue/bright yellow flame envelope and haze at the flame boundary. These are all indicators of fuel oil vaporization within the oil gun. This two-phase or vapor flow through orifices originally designed for liquid flow will severely reduce the oil gun capacity and stability.

Heavy oils which contain residual or added light oil will exhibit these same problems, as the lights flash, with the added problems of heavy oil slug flow and resulting smoke and poor atomization of the heavy ends.

Oil Tip Misposition

By far the most common oil firing problem and the most detrimental condition to oil firing, is the mispositioning of the oil tip with respect to its air supply/stabilization source. Due to variations in oil, atomizing medium, oil temperature, atomizing medium quality, burner air side P, operating oil/atomizing medium pressures, and furnace requirements for flame pattern, all John Zink oil guns are supplied with oil tip adjustability.

While the tip position is located on the John Zink burner assembly drawing supplied with every job, the final position is a field operator adjustment for optimum operation.

Maintenance

Tools

1. Cooling Water Bath
2. Stationary Vise.
3. Wire Brush
4. 12 or 24 Inch Pipe Wrench
5. 1-1/2 Inch Open End Wrench
6. 1-5/8 Inch Open End Wrench
7. 3/4 Inch Open End Wrench
8. 1 / 4 Inch Allen Wrench (Male Hex Drive)
9. Standard Screw Driver (Slot Type)
10. Copper, Bronze, or Nickel Containing High Temperature Pipe Thread Lubricant with Applicator Brush
11. Hole Gauges - Contact John Zink Company for Specific Sizings

Oil Gun Insert Removal

Two types of oil gun assembly are available from John Zink Company. These oil guns vary only in the method for supply connections.

One style, the Z-39-C oil body style, requires the oil and atomizing medium to be broken for oil gun insert removal, while the most common, the Z-56 oil body style, does not. In both cases, the oil gun guide tube, which provides the tip adjustment, does not have to be removed with the tip/atomizer assembly for cleaning.

In all fuel cases when breaking the closed oil and atomizing medium systems, safety is of high importance. It is highly recommended that a full purge of the oil side be completed prior to insert removal. This will minimize the possibility of oil spills exterior to the burner

Z-39-C

After purging the oil gun, proceed to disconnect the oil and steam piping making sure to shield any possible pressure spray from the connections due to oil gun pluggage. Then loosen the three set screws located at the back of the oil gun guide tube. Oil gun should then be completely retractable without disturbing the guidetube location.

Z-56 "Quick Change"

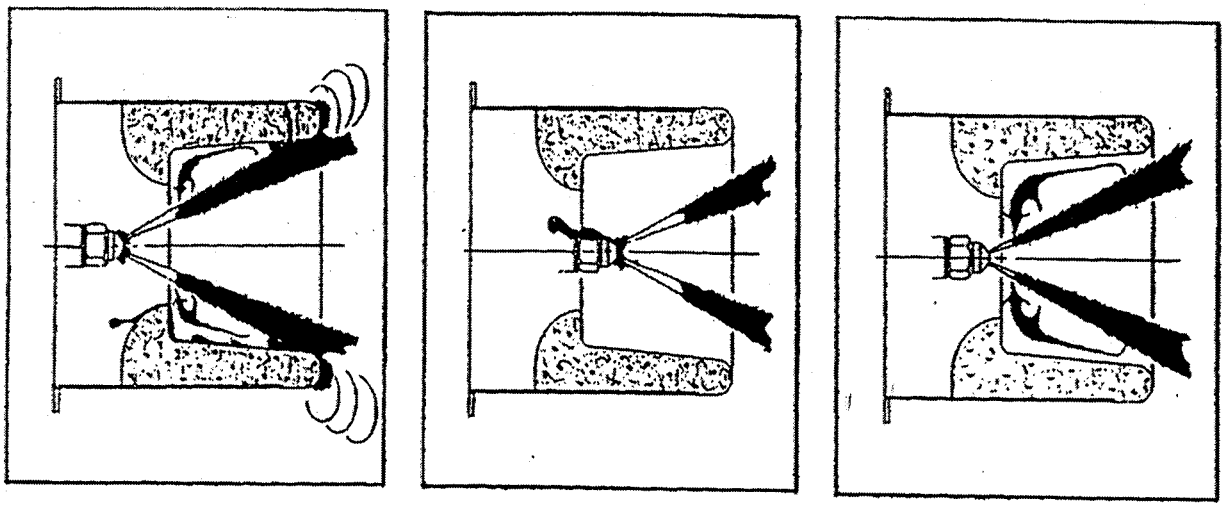
After purging the oil gun., proceed to loosen the oil gun clevis by use of the clevis handle making sure to shield any possible pressure spray from the interface sleeves due to oil gun pluggage. Swing the clevis away and retract the insert.

Regen Tile Operation/Tip Location

Proper oil tip location in the regen tile is dependent on the included spray angle, the number of exit ports and the depth of the regen tile. To achieve optimum regen tile function the main body of the spray pattern should clear the exit of the tile but should be close enough to establish full volume recirculation of burning oil mist for stabilization.

The over-insertion of the oil tip in the regen tile will reduce or eliminate the stabilizing recirculation causing combustion initiation to occur further out in the burner and will cause lifting and flameout in turndown conditions. This type of setting will also result in coke buildup on the oiltip and oil dripping due to the loss of the primary air velocity across the tip.

An oil tip which has been positioned too far behind the regen tile will often impinge on the exit of that tile. As the distance is increased, the burner flame will progress from smokey edges and sooting of the edges of the outlet, through coke buildup on the tile, to oil splashing and major spills.



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Disassembly

1. Using the cooling water bath, quench any residual heat buildup in the tip area until the tip/sleeve stay be gripped, in a bare hand, without discomfort.

2. Position the oil body, Item 401 in the stationary vice and tighten.

3. Using the 1-5/8" end wrench as a backup an the oil gun sleeve, Item 407, remove the oil tip, Item 406, using the 1-1/2" wrench.

4. Using the pipe wrench, remove the steam tube/sleeve assembly, Items 411 and 407, from the oil body.

5. Using the pipe wrench as a backup on the oil tube, Item 412, remove the atomizer assembly, Item 408 and 409, using the 3/4 inch end wrench.

6. Holding the atomizer body with the 3/4 inch end wrench, remove the oil spud using either the Allen wrench or the slotted screw driver.

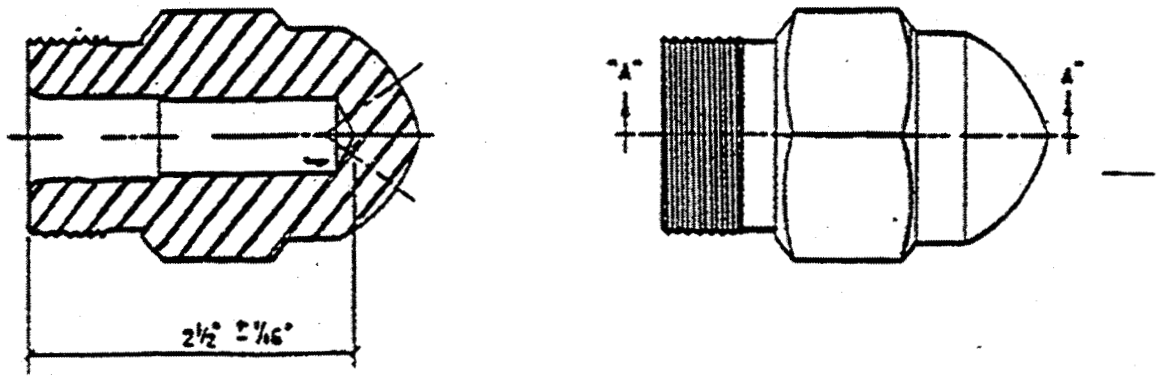
NOTE: Dependent upon the materials of construction, the type of drive supplied on the oil spud may be either style. A quick inspection from the threaded end of the atomizer assembly will determine the proper tool.

Inspection

Oil Tip

External inspection of the port region will reveal much about the condition of the oil gun. Visually, carbon or oil buildup around the exit ports is an indicator of three(3) possible problems: tip inserted too far, low atomizing medium pressure or flow, erosive or corrosive action on tip.

After cleaning all foreign materials from the face of the oil tip, inspect the exit ports for wear. Visual erosion, egg-shaping, of the exit ports will cause disruption of the flame pattern and should be accepted as reason for replacement.



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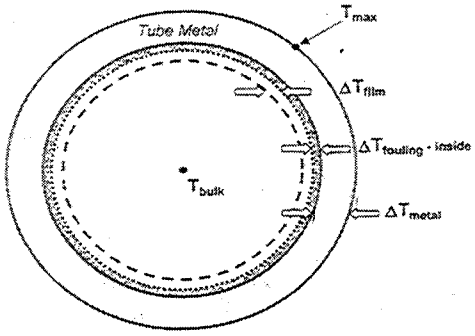
Training Services

Inspection & Turnaround

Skin Thermocouples & Infrared Monitoring

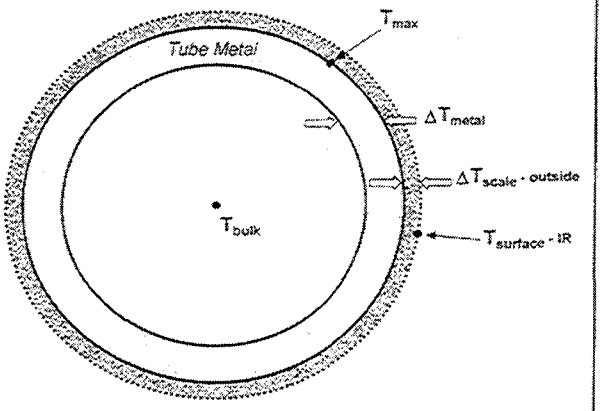
TWT

- TWT= Bulk+Film+Metal+fouling
  - Film= 60-150°F(30-75°C)
  - Metal= 10-20°F(5-10°C)
- Coking-black oil processes-4mm coke raises TWT 100°F (50°C)
- Hydrotreating-FeS scale-0.8mm scales raises TWT 100°F (50°C)



Tube Surface Temperature

- 20 mils (0.51mm) oxide scale raises surface temp. 210°F(105°C)
- Oil ash can increase surface temperatures up to 900°F(500°C)



FH 2004/12-3

Estimated Clean Tube Wall Temperature (TWT)

TWT= Process Outlet + Factor(F)

F = 150°F for Platforming Heaters and Hydrocrackers

F = 60°F for Oleflex Heaters

F = 200°F for Vacuum Heater Outlet Tube

F = 60-100°F for all other heaters

FH 2004/12-4

Tube Wall Temperature (TWT)

Calulational Method

Skin Thermocouples

Infared Monitoring

Skin Thermocouples

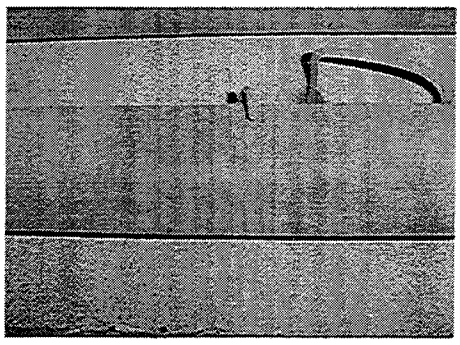
- Trend indicators to measure coking or fouling
- Not intended to be TI points
- Many read 50-100 F high
- Short Life

Skin Thermocouples

- Trend indicators to measure coking or fouling
- Not intended to be TI points
- Many read 50-100 F high
- Short Life

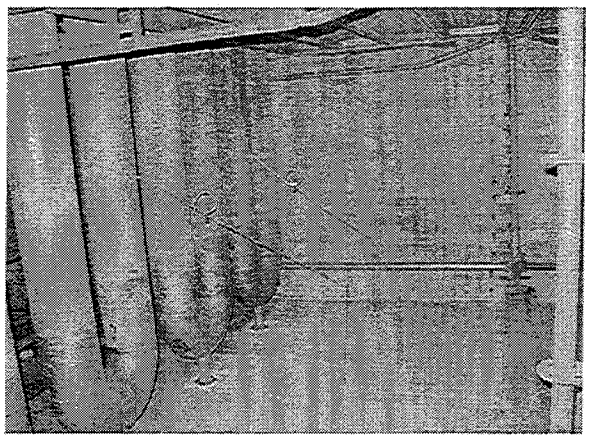
Short skin thermocouple life

- Thermocouple element has snapped from the head
- Insulate elements with moldable ceramic fiber, ceramic fiber cloth, or ceramic beads
- 310 SS elements have short life
- Pyrosil, Hasteloy,Inconel have longer life



Skin Thermocouple

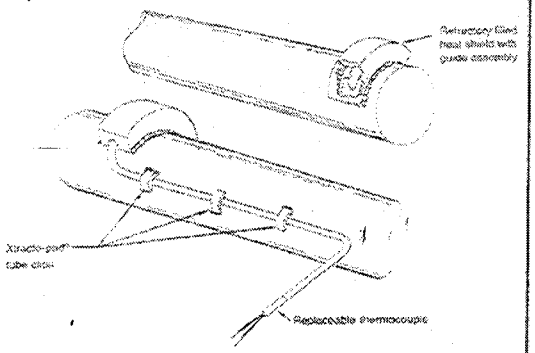
- Poor Installation
- Sheath will burn up
- Sheath should be clipped to tube



FH 2004/12-9

Pad Type Skin Thermocouple

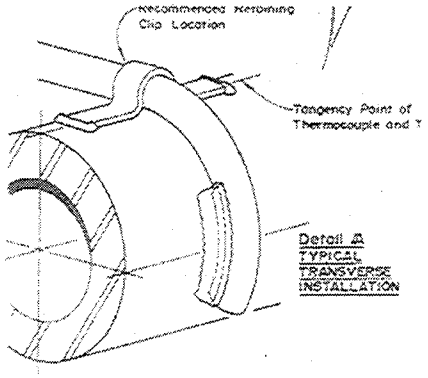
- Gayesco 'Xtractopad'
- Shielded Cover
- Replaceable Sheath
- 310 SS Sheath vs Pyrosil C (73% Ni , 24% Cr) Sheath
- Sheath clip to tubes
- Phone 713-941-8540
- Fax 713-944-3715



FH 2004/12-10

Knife Edge Skin Thermocouple

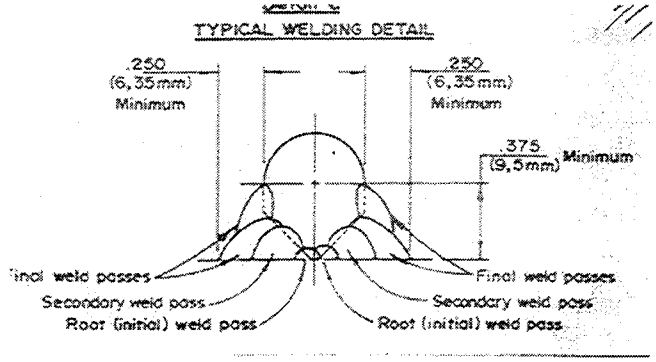
- Rugged and Reliable
- Harder to install than pad
- Reads 50 f higher than shielded
- Phone 708-653-1400
- Fax 708-653-2412



FH 2004/12-11

Knife Edge Skin Thermocouple

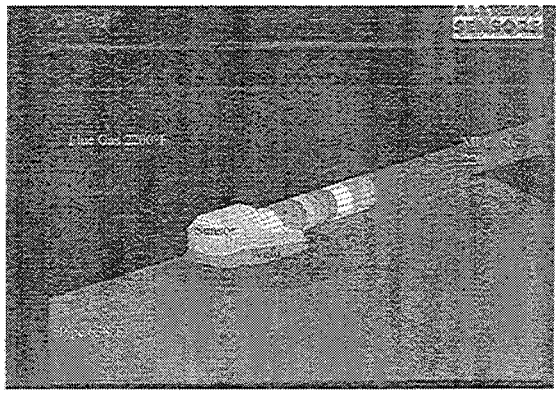
- Multiple weld passes required
- TIG Root Pass
- Reads 50 f higher than shielded



FH 2004/12-12

Skin Thermocouples

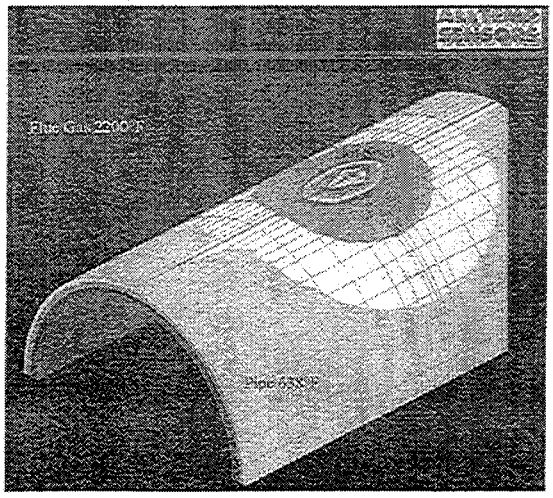
- CFD Model of knife edge
- Skin TI Acts as fin
- Creates Hot Spot



FH 2004/12-13

Skin Thermocouples

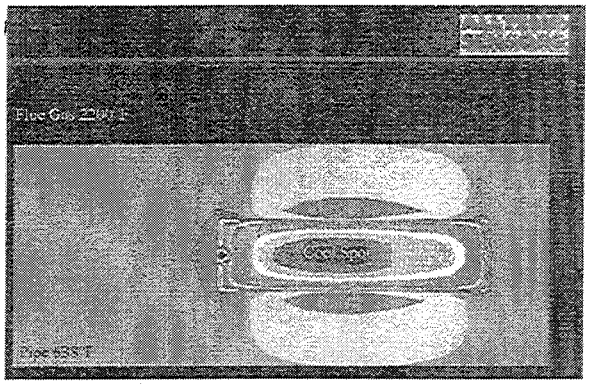
- CFD Model of hot spot



FH 2004/12-14

Skin Thermocouples

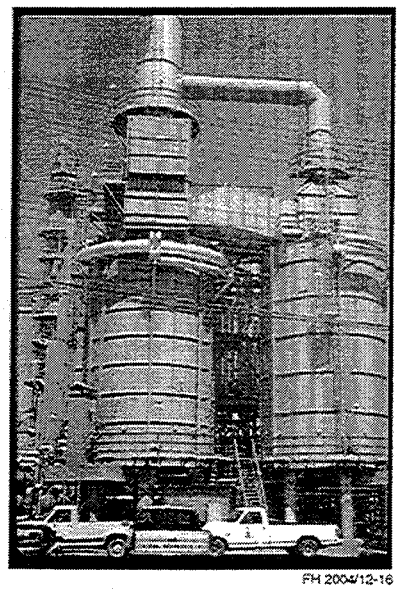
- CFD Model of shielded thermocouple
- Shield moves hot spot
- Accurate tube measurement inside shield



FH 2004/12-15

Skin Thermocouple

- Skin TI's located 1/4 to 1/3 of box height at hot spot of heater
- Expect tube problems in this area
- Focus inspection efforts in this area



FH 2004/12-16

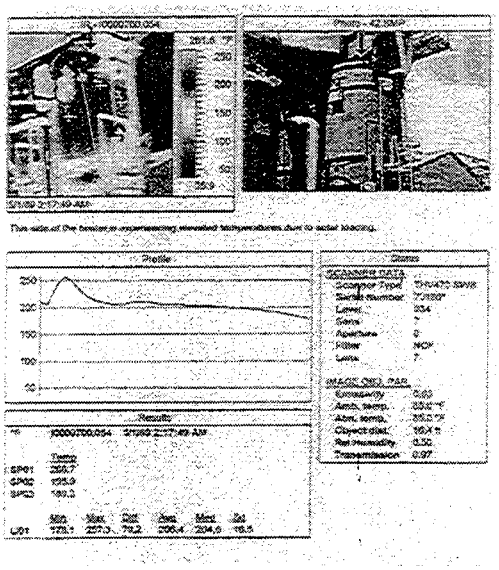
Infrared Monitoring

- Detect Refractory Damage
- Detect coking and hot spots on tubes
- Many refiners shoot critical heaters every 3 months

FH 2004/12-17

Infrared Camera

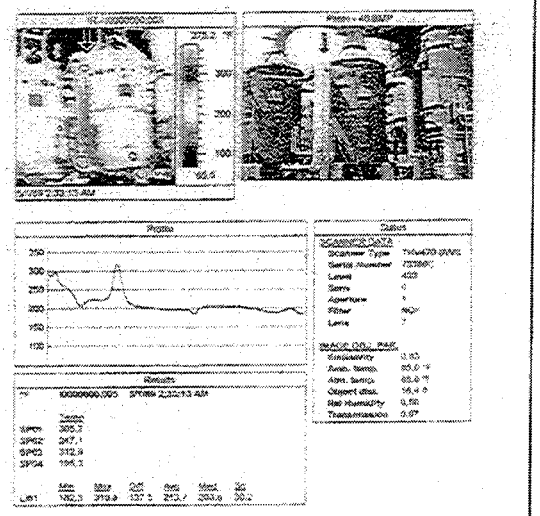
- Inframetrics or Agema
- Good for detecting hot spots
- Heater Shell Temperatures
- Permanent Records



FH 2004/12-18

Infrared Camera

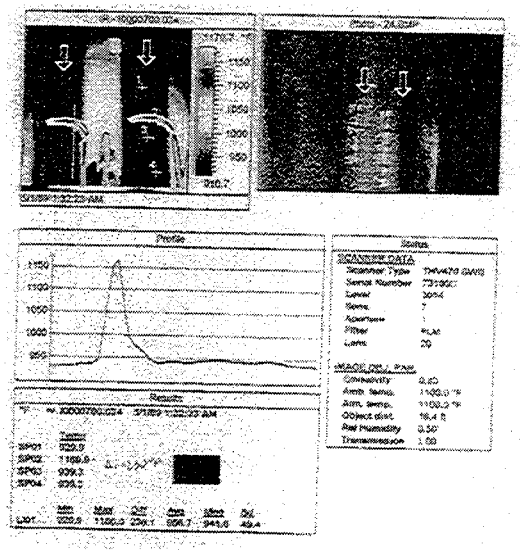
- Inframetrics or Agema
- Good for detecting hot spots
- Heater Shell Temperatures
- Note high heat loss from explosion door



FH 2004/12-19

Infrared Camera

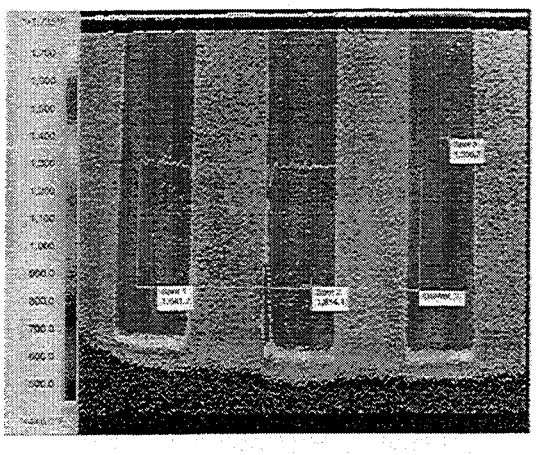
- Skin TI on Platforming Heater
- Skin TI Acts as fin
- Creates Hot Spot
- Skin TI reads High



FH 2004/12-20

Infrared Survey

- Platforming Heater
- Tubes one year after sandblasting
- Tight oxide scale layer

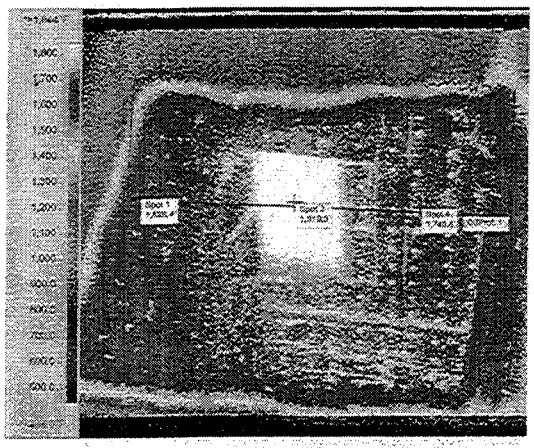


inframetrics ThermMonitor 95 2/22/99 11:25:44 AM

FH 2004/12-21

Infrared Survey

- Platforming Heater
- Outlet flue gas duct with fouled tubes
- Extremely hot outlet- poor process heat transfer- high firing rate



inframetrics ThermMonitor 95 2/22/99 10:58:45 AM

FH 2004/12-22

Portable Infrared Pyrometer

- **Infrared Pyrometer-(Minolta-Land Model 339)**
- **Suitable for furnace unit, (filters H2O and CO2 radiation)**

PH 2004/12-23

Training Services

Inspection & Turnaround

Tube Inspection

API Fired Heater Data Sheet

	Radiant	Convection	Convection	Convection
Coil Design	←	Process	→	Steam
* 4. Design basis for tube wall thickness (code or specification)	←	API RP 530	→	ASME I
* 5. Design basis for rupture strength (minimum or average)				
* 6. Design life, hr	←	100000	→	
* 7. Elastic design pressure, kg/cm²(g)	←	21	→	8.1
* 8. Rupture design pressure, kg/cm²(g)	←	12.34	→	4.68
* 9. Temperature allowance, °C	15	15	15	15
* 10. Corrosion allowance, mm	3	3	3	3
* 11. Stress relieve (yes or no)	yes			no
* 12. Weld inspection requirements, X-ray or other (no. of welds and degree of coverage	←	100%	→	10%
13. Hydrostatic test pressure, kg/cm²(g)				
14. Maximum tube metal temperature (clean), °C	465			506 (no flow
15. Design tube metal temperature, °C	550			371
16. Inside film coefficient, kcal/m² h °C				

Heater Tube Thickness

- **Check Existing Tube With API-530**
- **Vendors Often Report Low Design Temperature**
- **Older Heater Tubes May Not Meet Current API-530 Requirements**

Heater Tube Thickness

- **API 530 'Heater Tube Thickness'**
- **Check Existing Tube Thickness**
- **Determine Retirement Thickness**
- **Creep - High Temperature- Permanent Damage**

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

$$t_m = t_s + CA$$

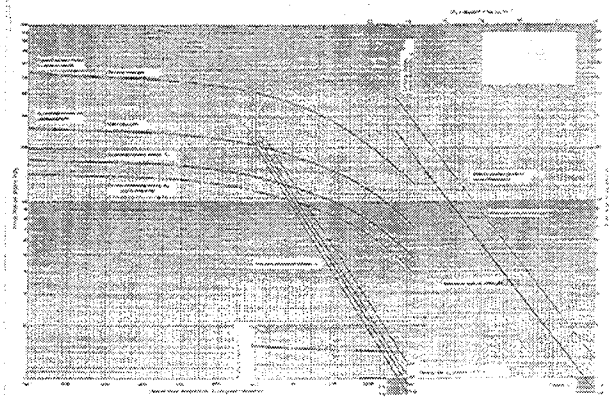
$$t_s = \frac{P_E D_o}{2S_E + P_E}$$

where:

t_m = minimum tube wall thickness required, in
t_s = stress 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

Heater Tube Thickness

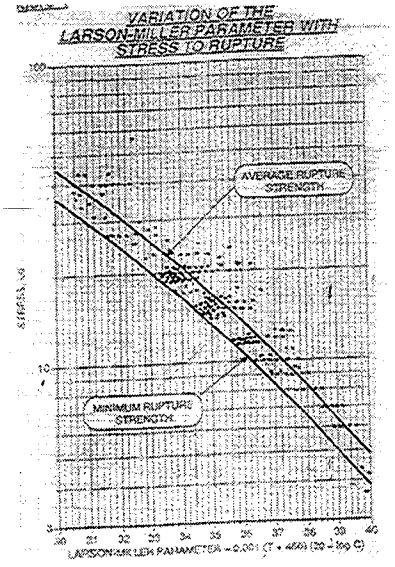
- API 530 'Heater Tube Thickness'
- Elastic - Low Temperature- No Permanent Damage
- Creep - High Temperature- Permanent Damage



FH 2004/12-28

Creep Rupture Stress

- Before 1978- Average
- High Failure Rates
- After 1978- Minimum



FH 2004/12-29

Creep Range

- Permanent Damage
- Tube Diameter Growth
- Replace Visible Bulges

FH 2004/12-30

Creep Temperatures for Tube Metallurgy

■ Carbon Steel	775°F (413°C)
■ 1-1/4 Cr	925°F (496°C)
■ 2-1/4 Cr	895°F (479°C)
■ 5 Cr	885°F (474°C)
■ 9 Cr	960°F (516°C)
■ TP 321	1010°F (543°C)
■ TP 347	1100°F (593°C)

FH 2004/12-31

High Tube Temperatures

- Sagging and Bowing
- Oxidation or Scaling
- Bulging or Creep Leading to Rupture
- Metallurgical Change
- Increased Corrosion
- Thermal Fatigue (Temperature Cycling)
- Coking

FH 2004/12-32

Tube Metallurgy Limits

Metallurgy	Type	Oxidation Limit, °F	Creep Data Limit, °F	Practical Limit, °F
C.S	A-106	1000	1000	780
C-1/2 Mo	P1	1000	1050	Not Used
1-1/4 Cr-1/2 Mo	P11	1050	1100	950
2-1/4 Cr-1 Mo	P22	1075	1200	1075
5 Cr-1 Mo	P5	1150	1200	1150
9 Cr-1 Mo	P9	1175	1300	1175
18 Cr-8Ni	304	1650	1500	1300
18 Cr-10Ni-Ti	321	1650	1500	1300
18 Cr-10Ni-Nb	347	1650	1500	1300

FH 2004/12-33

Process Metallurgy Guide

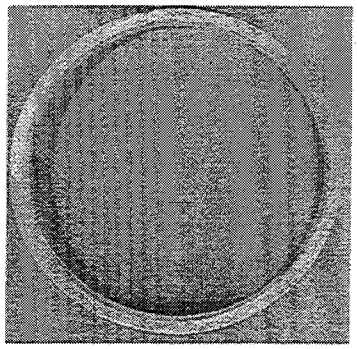
Process	Internal Scale	Typical Metallurgy	Typical Corrosion Allowance	Typical Failure Causes
Crude	Coke	9 Cr	1/8"	1, 3
Vacuum	Coke	9 Cr	1/8"	1, 3
Coker	Coke	9 Cr	1/8"	1, 3
Naphtha Hydrotreating	Inorganic	1-1/4 Cr to 9 Cr 347 SS	1/8"- 3/16" 1/16"	1,2
Kerosine/ Diesel Hydrotreating	Inorganic	347 SS	1/16"	1,2
Hydrocrackers	Inorganic	347 SS	1/16"	1,2,3,4
Catalytic Reformers	Clean	2-1/4 Cr to 9 Cr	1/16"	1,4
Reboilers	Clean	CS	1/8"	1

TYPICAL FAILURE CAUSES

1) Overheating caused by low flow or flame impingment
2) Hydrogen sulfide attack
3) Sulfur attack
4) Long Term Creep Deformation

FH 2004/12-34

Tube Scale



Inorganic scale has a thermal conductivity of 6 Btu/h-in-°F. Coke has a thermal conductivity of 30 Btu/h-in-°F.

At 19,000 Btu/h-ft^2 peak flux (10,000 avg. flux, a 1/32 inch internal scale deposit will raise the maximum TWT by 100°F.

At 19,000 Btu/h-ft^2 peak flux (10,000 avg. flux, a 5/32 inch internal scale deposit will raise the maximum TWT by 100°F.

FH 2004/12-35

Heater Tube Replacement

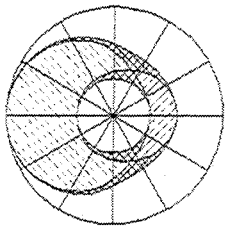
- Loss of Wall Thickness Due to Corrosion or Oxidation (Ultrasonic Testing)
- Replace Visible Bulges (Tube Banding)
- Loss of Strength or Ductility (Tensile Stress Testing)
- Carburization (Micro Hardness)
- Remaining Life Assessment

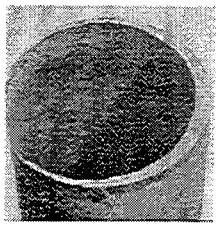
Non Destructive Heater Tube Testing

- Wall thickness (Ultrasonic Testing)
- Diameter Growth (Tube Banding)
- Brinell Hardness (Hardness)
- Replication (Surface Structure)

Corrosion

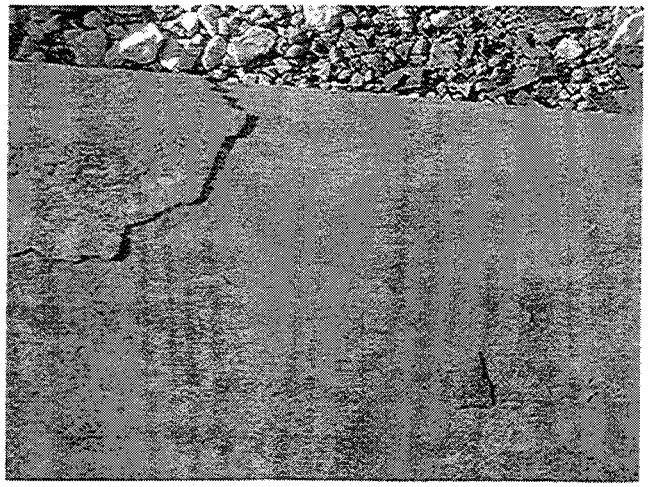
- Corrosion and Tube Thinning Follows Heater Flux Pattern
- Measure Thickness on Front Face





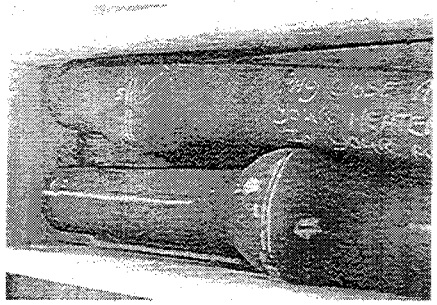
Crude Tube

- High Oxidation Rate is the start of creep failures



Heater Tube Failures

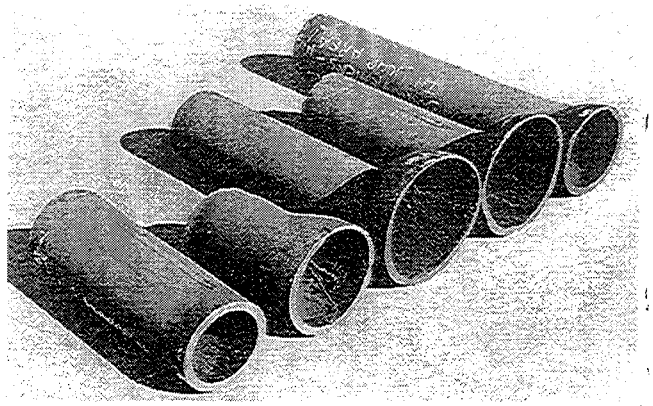
- Bulged Heater Tube
- Replace Visible Bulges
- Flame Impingement



FH 2004/12-40

Heater Tube Failures

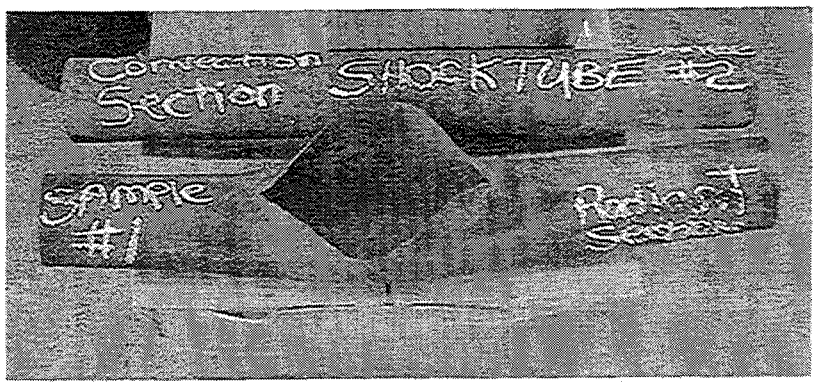
- Bulged Heater Tubes
- Thinnest Section Facing Flame
- Short Term High Temperature Yielding



FH 2004/12-41

Creep Failure

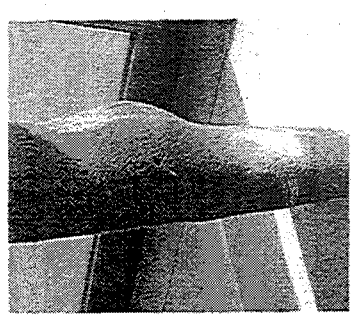
- Caused by flame impingement in convection section
- Smart pig inspection
- Classic 'Fishmouth failure'



Crude Heater –Coking

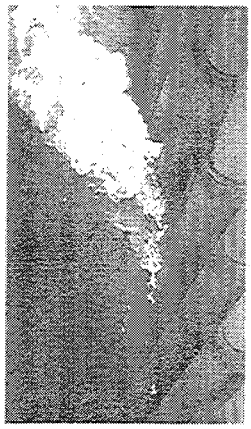
- Asphaltene Precipitation
  - –Opportunity Crudes
  - –Crude Blending Incompatibility
- Salt Precipitation
  - –Poor Desalting
  - –Caustic Injection
- High Oil Film Temperature
  - –High Flux-low mass velocity design
  - –Outlet temperatures above 730°F(388°C)
- Poor flame management
  - –Poor burner maintenance
  - –Keep O2 levels high enough to keep flame off of tubes

Local Hot Spot caused by coking

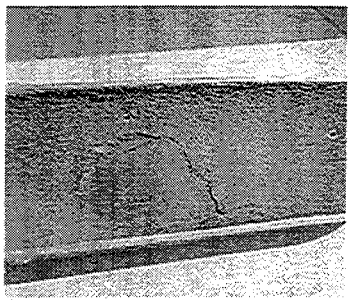


Crude Heaters-Pinhole leaks

Leaking Tubes



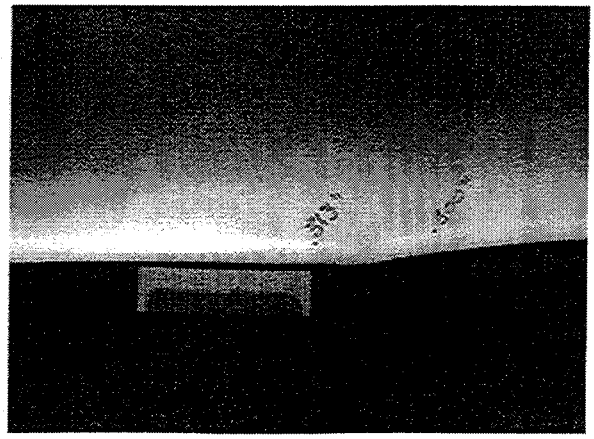
Failed Shock Tube



FH 2004/12-44

Creep Failure (NHT)

- H2S corrosion resulted in .10" metal loss
- Approximately 1" coke in tube
- Radiograph before decoking



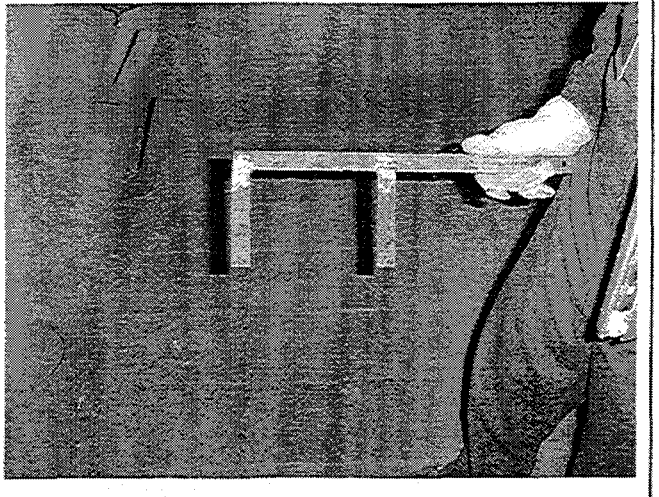
FH 2004/12-45

Tube Diameter Growth Detection

- Banding
- Tube Gauges
- Level and Light Method

Tube Gauge

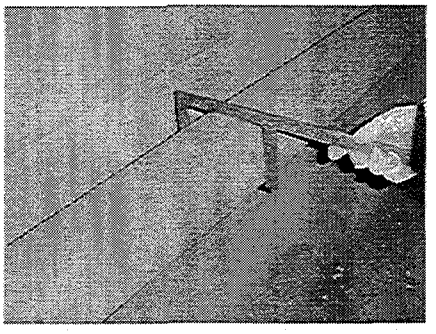
- Replace tubes above these diameter growths
- CS-5%
- 1-5 Cr-3%
- 9Cr-5%



Tube Gauge

- Check both planes

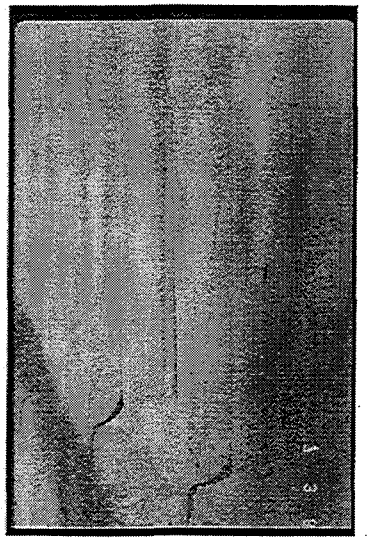




FH 2004/12-48

Heater Tube Failures

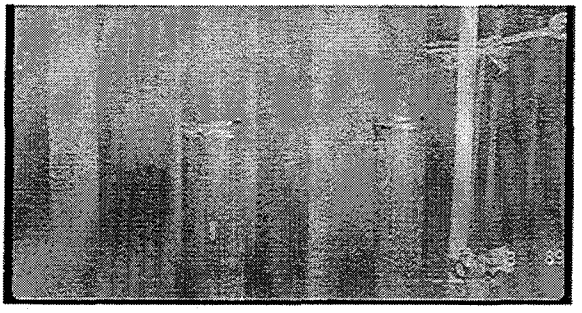
- Burner Tile Damage Caused Oil Spray on Tubes
- High Temperature Caused Tube Failure

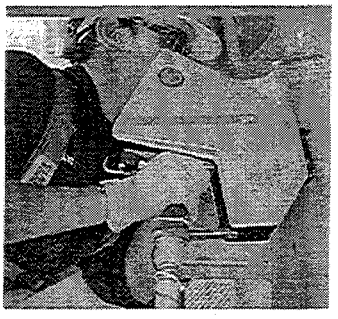


FH 2004/12-49

Heater Tube Failures

- PMI test new tube section
- Spliced in New Tube Section
- TIG Weld Root Pass(100 % Weld X-ray)
- PWHT (If Required)

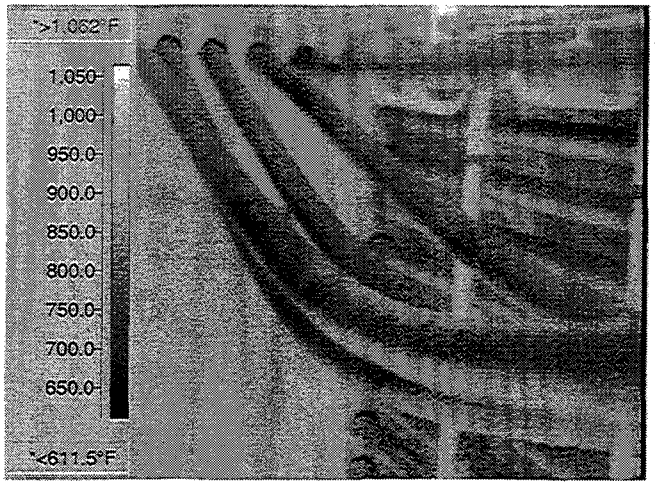




FH 2004/12-50

Heater Tube Failures

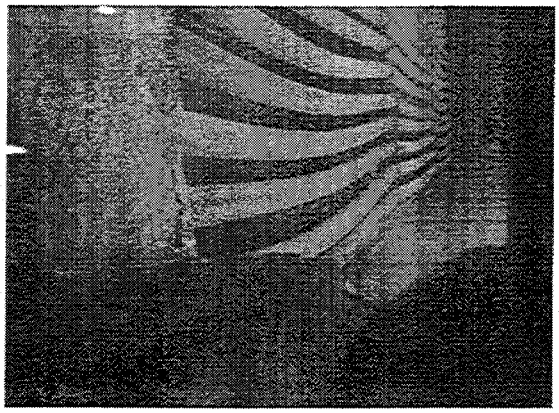
- Tube bowing and sagging
- Replace at 1-1/2 diameter bow



FH 2004/12-51

Heater Tube Failures

- Tube bowing and sagging-often caused during decoking
- Use infrared monitoring to control coke burn
- Replace at 1-1/2 diameter bow



FH 2004/12-52

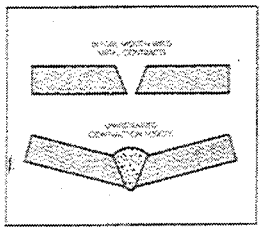
Postweld Heat Treatment

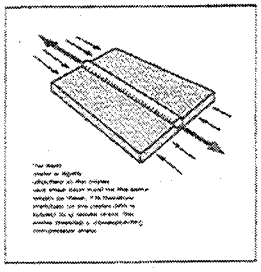
- PWHT 1-1/4 Cr to 9 Cr
- Lowers Hardness and Residual Stresses
- Max. Brinell Hardness = 225-241
- Proper Temperature Range Required Per ASME Section VIII, UCS-56 Requirements



FH 2004/12-53

Postweld Heat Treatment





Increase Ductility and Reduce Hardness:

It is generally recognized that postheat typically lowers the tensile and yield strength of carbon steel and alloy-steel weld metals because softening of the material results. The resultant material is more readily formed and exhibits greater resistance to impact loading.

Diminish Brittle Failure:

Structural steel, pressure vessels, and complex welded components can fail catastrophically even under normal loading conditions.

Steel normally considered ductile can behave in a brittle fashion in the presence of certain defects, such as notches or stress risers. This is particularly true of steels operating at low temperatures.

Postheat treatment, by relieving the residual stresses and tempering the metal, can enhance the fracture toughness of the material and prevent brittle failure.

Increase Resistance to Corrosion:

Stress-corrosion cracking and caustic embrittlement can occur in pressure vessels, boilers, petrochemical piping, and other welded construction exposed to certain environments, such as hydroxides, nitrates, and hydrogen sulfides. Postheated welds exhibit reduced levels of residual stress and typically have a higher resistance to attack by corrosive environments than untreated welds.

Improve Machining Stability:

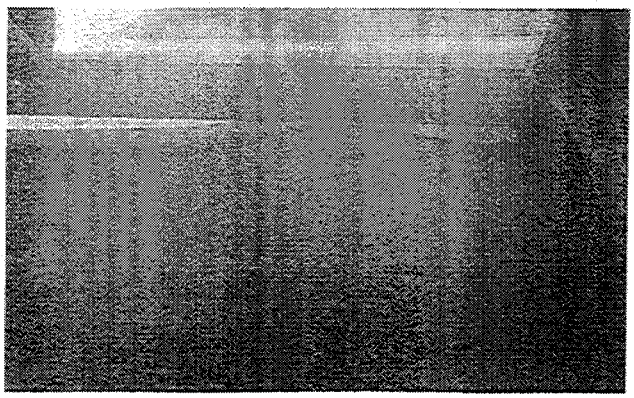
Welding a complex fabrication results in locked-in stresses. If the stresses remain undelieved, distortion after machining can result because the metal helping to balance the tension and compression forces is removed.

Sometimes, intermediate heating is done as the weldment is fabricated, thus reducing excessive build-up of stresses. Such heat treatment, therefore, can enable the weldment to be machined after welding without distortion.

FH 2004/12-54

Manifold Failure

- **1-1/4 Cr Manifold**
- **Damaged During Postweld Heat Treatment**
- **Temperature Was Too Hot**



FH 2004/12-55

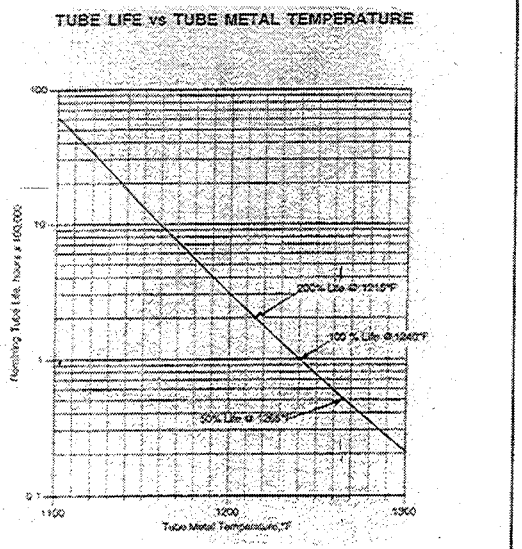
Remaining Tube Life

- Tubes Designed for 100,000 Hour Life
- API 530 Method
- New API 579 Method
- Destructive Testing

FH 2004/12-56

Remaining Tube Life

- API -530 Appendix E" Estimation of Remaining Tube Life"- Not Accurate
- Record Operating TWT and Pressures
- Perform Calculations
- Not Accurate Due to Temperature Sensitivity
- New Method API -579 "Fitness for Service" - More Accurate



FH 2004/12-57

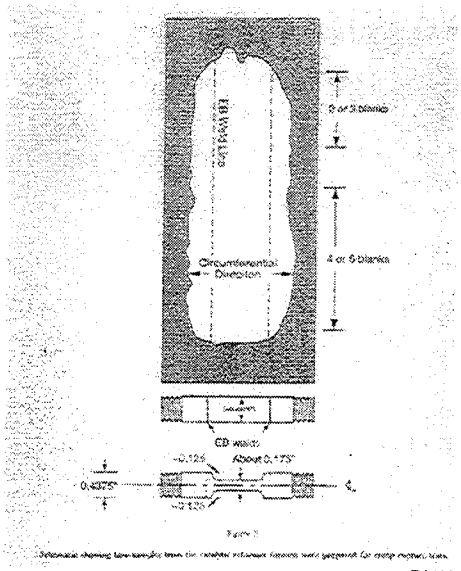
Destructive Heater Tube Testing

- Exposure to High Temperature
- Exceeded 100,000 Design Life
- Omega Testing (Remaining Creep Life)
- Loss of Strength or Ductility (Tensile Testing)
- Creep Voids (Microscope)
- Carburization (Micro Hardness and Microscope)

FH 2004/12-58

Omega Method

- Basis for API -579 "Fitness for Service"
- Determines Remaining Life
- Destructive Samples Required



FH 2004/12-58

Omega Method

- Performed by UOP or Others
- Accurately Predicts Remaining Life
- Used for Tubes Operating in High Temperature Creep Range

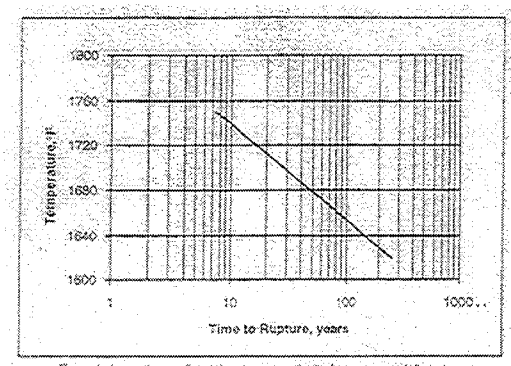


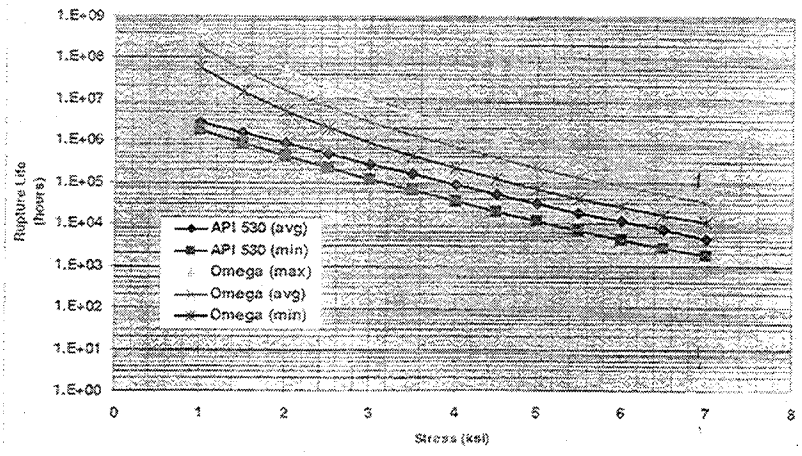
Figure 1 shows the predicted time to rupture the hydrogen furnace tubes at a stress of 700 psi in the tube wall based on the Omega Method creep test results.

FH 2004/12-60

Omega Method

- Longer life for low alloys like 2-1/4 Cr

Rupture Life for 2.25Cr at 1150F



FH 2004/12-61

Tensile Stress Test

- Tensile Stress, Yield Stress, %Elongation, %Reduction of Area
- Replace Tube When Test Are Below Minimum Standards

TENSILE TEST DIR.: LONGIT. ROOM TEMPER. TYPE RECT. 1"

TEST	HEAT	SIZE IN IN	SECTION IN2	LOAD #	Y.S. PSI	LOAD #	T.S. PSI	ELONGAT % GL0
				MIN IN2	30000		60000	30
22	327706	25.30X11.90	301.07	106000	52000	165000	79500	40
25	327706	25.30X11.35	287.15	105000	53000	156500	79000	38
27	324867	25.30X12.15	307.40	106000	50000	160000	75500	40
28	324867	25.30X11.35	287.16	100000	50500	146500	74000	40
31	327706	25.30X11.50	290.95	107000	53300	159000	75100	40
32	327706	25.40X10.65	270.51	100000	52600	148000	79400	36
35	324982	25.40X11.25	285.75	102000	51800	153000	77700	38
36	324982	25.40X11.15	283.21	103000	52700	152000	77800	38
39	327706	25.30X11.25	286.88	107500	54300	158000	79900	38
40	327706	25.30X11.30	288.15	106000	53400	155000	78000	38
41	327706	25.30X10.75	271.98	102500	54700	150000	80000	36
45	327706	25.30X11.70	296.01	104000	51000	151000	75900	40
47	327706	25.40X11.50	293.37	105000	51900	160000	79100	40

FH 2004/12-62

Microscope Testing

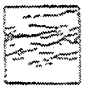
- Destructive Sample
- View Under Microscope
- Destructive Level of Creep Voids
- Microstructural Changes

1 

2 

3 

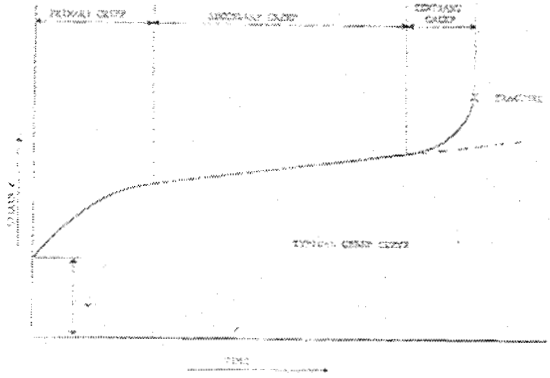
4 

5 

FH 2004/12-63

Creep Stress

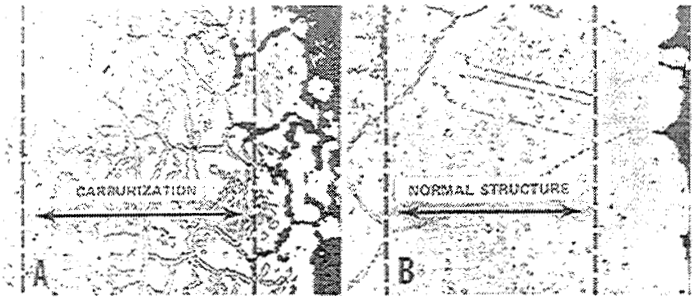
- **Permanent Damage With Time**
- **Not Linear**
- **Cumulative**



FH 2004/12-84

Carburization

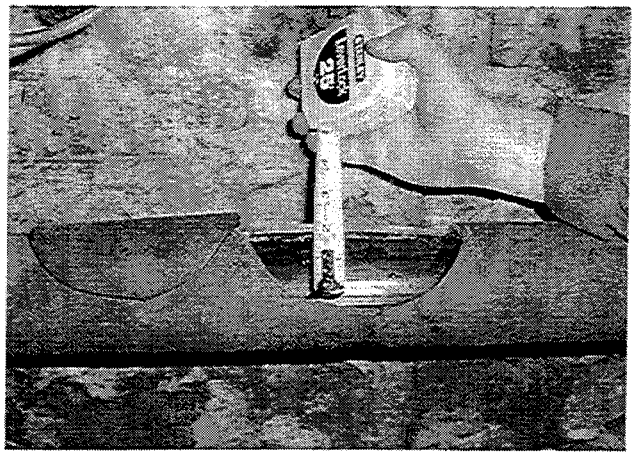
- **Carbon Migrates Into Crystal Structure**
- **Loss of Ductility**
- **Occurs at Elevated Temperatures**



FH 2004/12-85

Carburization Failure

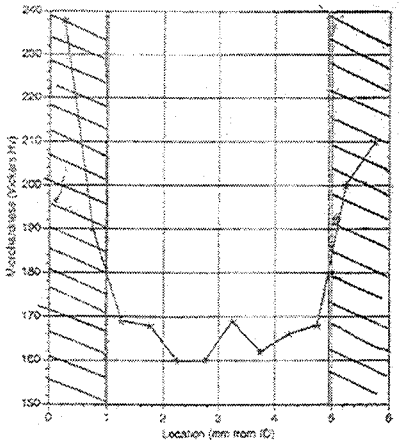
- Coker tube failed during pigging
- Coke thickness is 0.25"



FH 2004/12-66

Microhardness

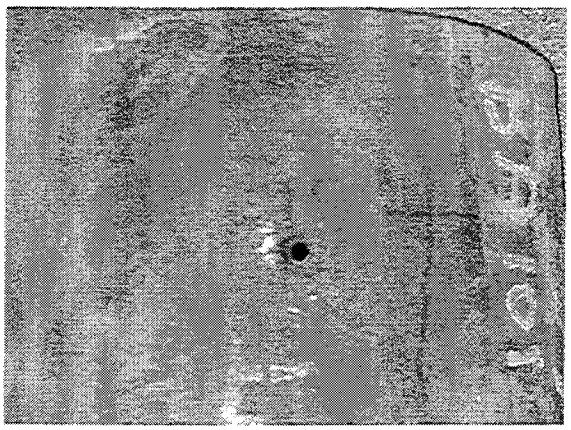
- Determines Carburization Depth
- Requires Tube Sample



FH 2004/12-67

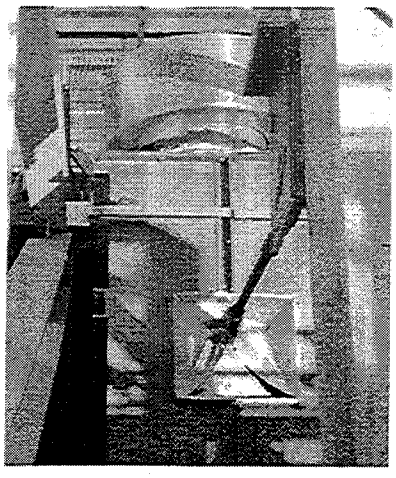
Crude Tube

- Vanadium Corrosion-pitting
- Occurs > 620°C
- Fuel V usually 100-300ppm



Heater Casing Inspection

- Ceramic Fiber Cloth Seal on Manifold Is Torn
- Rain Will Fall on Hot Manifold, Possible Thermal Damage
- Need Weather Shield



Manifold Failure

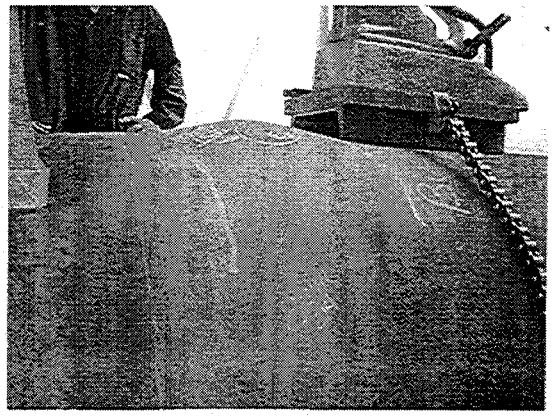
- 347 SS Manifold
- Thermal Fatigue Failure
- Caused by Rain and Bad Header Box Seals



FH 2004/12-70

Manifold Failure

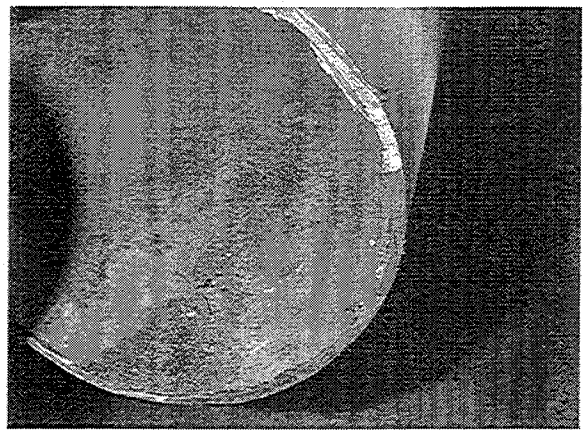
- Bulge
- 3 % Maximum Out of Round ASTM A-450



FH 2004/12-71

Heater Tube Failures

- 347 SS Tubes
- Cl attack from Hydrotesting Water
- Failure Before Startup



FH 2004/12-72

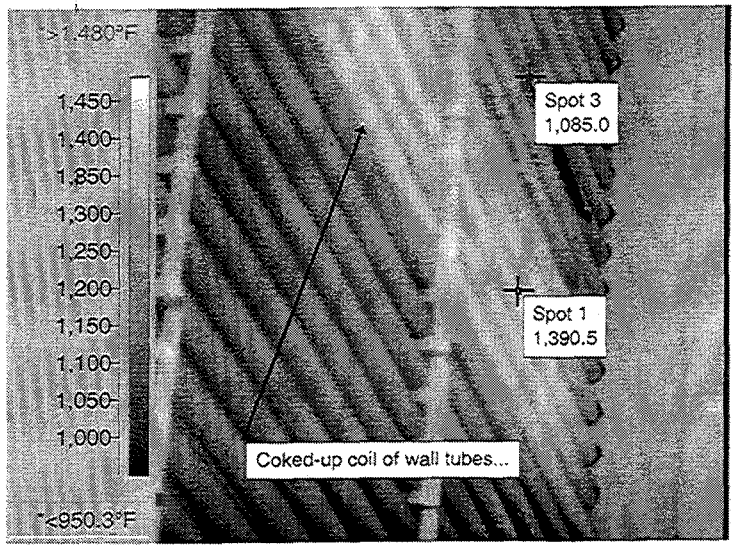
Heater Tube Inspection

- Deposits
  - Internal
    - Coke
    - Iron Sulfide
  - External
    - Oxidation Scale
    - Fuel Oil Ash
    - FCC Catalyst Fines

FH 2004/12-73

Coke in Heater Tubes

- Use Infrared Pyrometer to detect coke in tubes



FH 2004/12-74

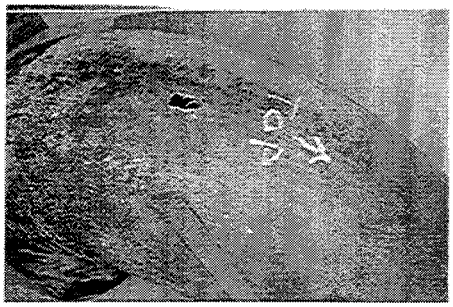
Steam- Air Decoking

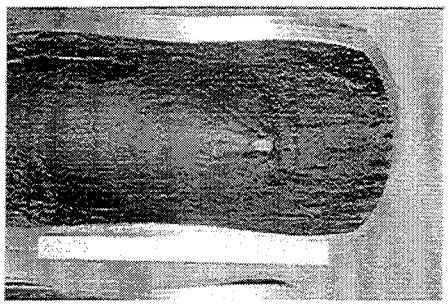
- X-ray to Determine Thickness
- Does Not Work Well on Inorganic Scale - Hydrotreaters
- Spalling Is the Most Important Step
- Control Burn- High Temperatures Can Cause Severe Tube Damage
- Use Infrared Pyrometer to Monitor TWT

FH 2004/12-75

Erosion from Spalling Coker Tubes

- UT return bends
- Thicker bends required
- Pigging eliminates this problem





FH 2004/12-76

Pig Cleaning Heater

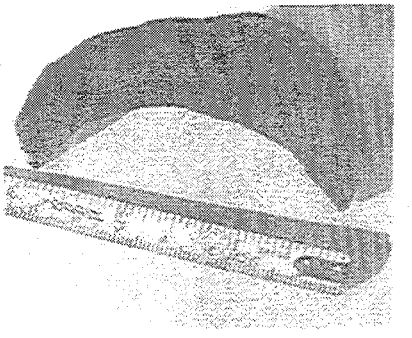
- Less Damage Than Steam-air Decoking
- Good for Inorganic Scale Removal (Hydrotreaters)
- Can Get Stuck in Soft Coke
- Tubes Must Not Be Bulged
- Bend Radius Restrictions
- www.decoking.com



FH 2004/12-77

Pig Cleaning Heater

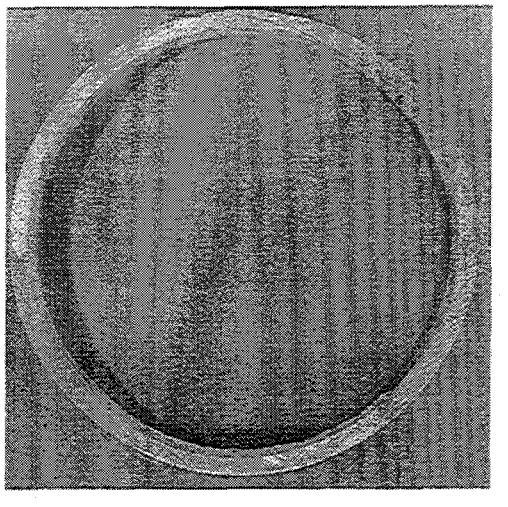
- Sample From Coker Heater
- Thickness Is Approximately 0.1 inch (2.5mm)



FH 2004/12-78

Iron Sulfide Scale

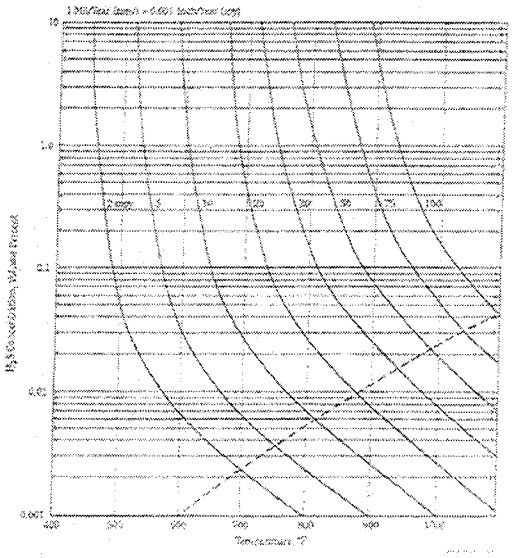
- 5 Cr Hydrotreater Tube
- NPS 5 Tube (0.26" Thick)
- Iron Sulfide Black Inner Layer (0.10" Thick)
- Pig Cleaning Is Preferred
- Acid Cleaning Is Alternate Method



FH 2004/12-79

Iron Sulfide Scale

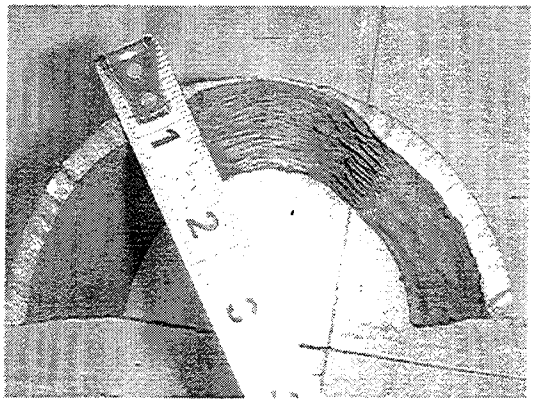
- Hydrotreater Tubes
- Use 100°F Above Process to Predict Corrosion Rate
- 15 mils is High Corrosion Rate
- Curves Have Been Accurate



FH 2004/12-80

Iron Sulfide Scale

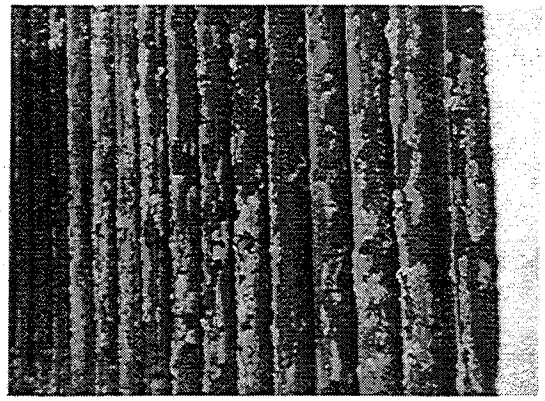
- 5 Cr Diesel Hydrotreater Fractionation heater
- NPS 5" Tube (0.26" Thick)=corroded to failure
- Corroded on top of pipe in convection section
- Caught by smart pig



FH 2004/12-81

Oxidation Scale

- Catalytic Reforming – 2-1/4 Cr Tubes
- Heavy Scale-Tree Bark-Glowing scale has delaminated
- Increased Fuel Firing-Every 100° increase BWT 50°



FH 2004/12-82

Oxidation Scale

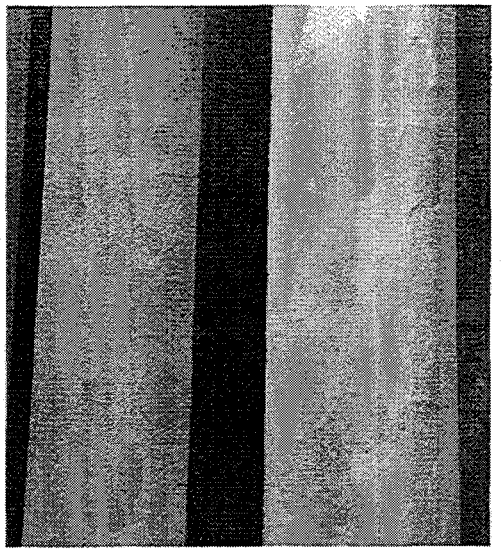
- Use Walnut Shells to Blast Loose Scale off Tubes on Line
- Reduced Fuel Firing



FH 2004/12-83

Oxidation Scale

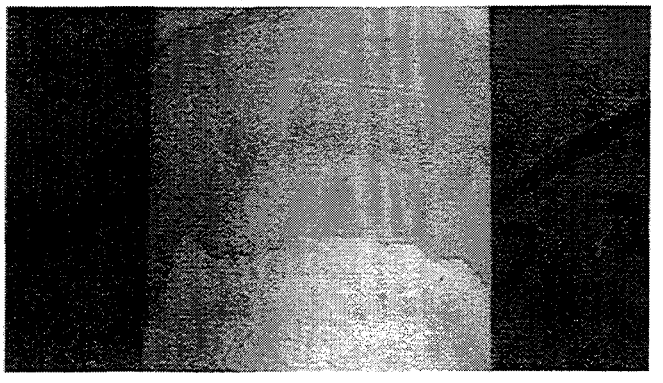
- 9 Cr Platforming Tubes
- 10-40 Mils Oxidation Scale
- 200°F-400°F Hotter Surface Temp



FH 2004/12-84

Oxidation Scale

- 9 Cr Platforming Tubes
- 10-20 mils Oxidation Scale (Cleaned With Black Beauty Blast)
- White Metal



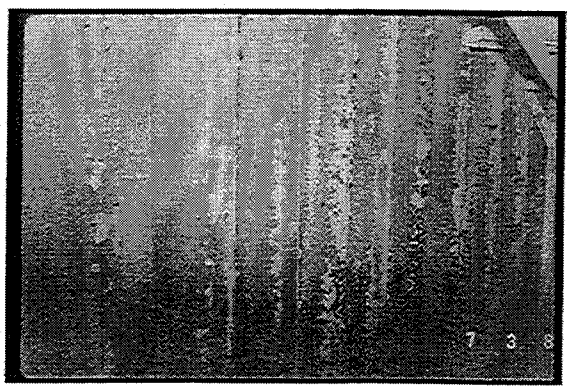
FH 2004/12-85

Heater Tube Fouling

- Fuel Oil Ash Insulates Tube
- Increased Firing Rate
- High Firebox Temperature
- Mechanical Failures
- Unscheduled Shutdowns

Fuel Oil Ash

- Oil Ash Acts As Insulation
- Increased Fuel Firing
- Sandblast Clean



Training Services

TYRI
Refractory Inspection

FH 2004/12-93

Section Outline

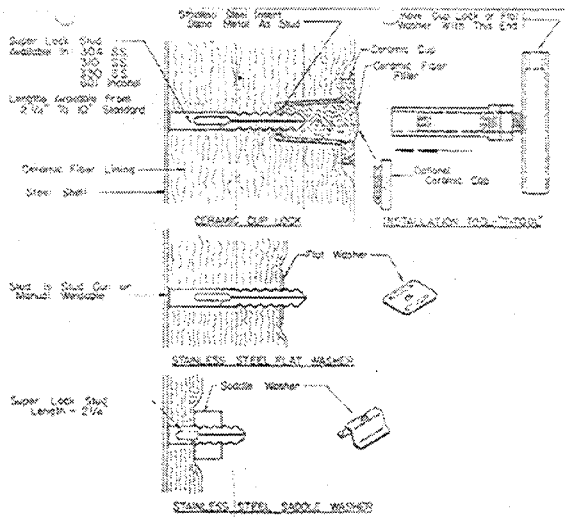
- Castable
- Ceramic Fiber

FH 2004/12-94

Ceramic Fiber

- Better insulator than castable
- Used for gas firing only
- Wallpaper System
- Module System

Ceramic Fiber



Ceramic Fiber Anchor Inspection

- Least reliable component
- Has the anchor tip burned off?
- Has the washer broken or fallen off?
- How deep is temperature damage?
- Stud integrity - Wiggle anchor
- Stud and washer should be insulated to protect against flame radiation

FH 2004/12-97

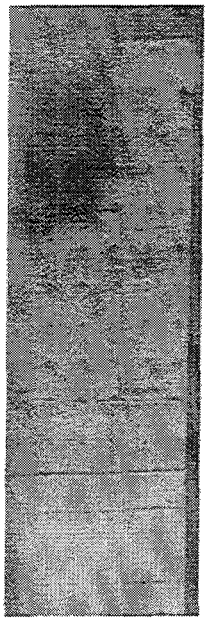
Maximum Temperatures for Anchor Tips

■ Carbon Steel	800 F (427 C)
■ TP 304 SS	1400 F (760 C)
■ TP 310 SS	1700 F (927 C)
■ RA 330	1900 F (1038 C)
■ Inconel	2000 F (1093 C)
■ Ceramic studs	› 2000 F

FH 2004/12-98

Ceramic Fiber

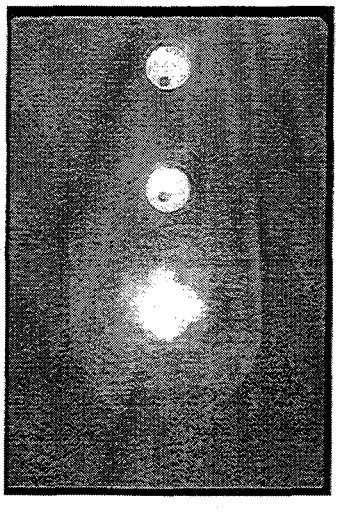
- Wallpaper Construction
- Overlap Design
- Lower Section is Ceramic Fiber Board
- Studs and Anchors should be insulated



FH 2004/12-99

Ceramic Fiber

- Ceramic Patch protects anchors and studs on wallpaper design



FH 2004/12-100

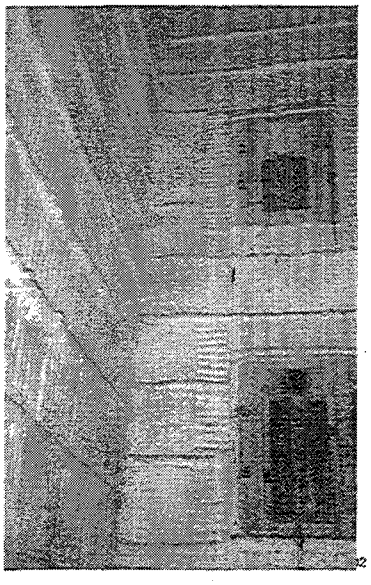
Ceramic Fiber Inspection

- High temperatures ›1800 F (982 C) causes chemical breakdown
- Detected by powdery appearance and loss of strength
- How deep is temperature damage?
- Integrity Test- Scape test
- Shrinkage and Gaps caused by high temperatures

FH 2004/12-101

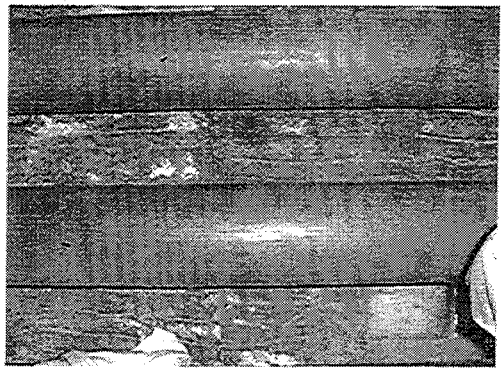
Ceramic Fiber Module

- Module Design
- Protected Anchor
- Best design for high temperature heaters
- Easier Repair
- Check for shrinkage between modules



Ceramic Fiber

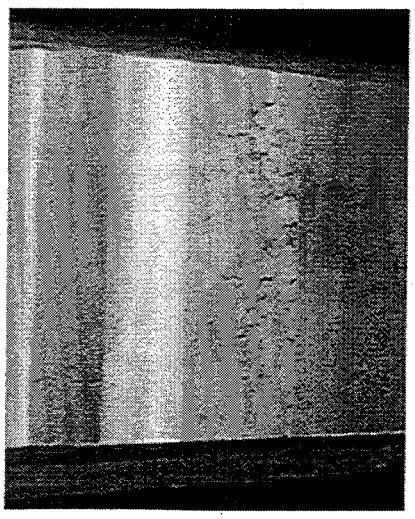
- Oil Fired Module
- White area is broken refractory
- Refractory flakes off when touched
- Gaps between Modules
- Short Life



FH 2004/12-103

Castable Refractory

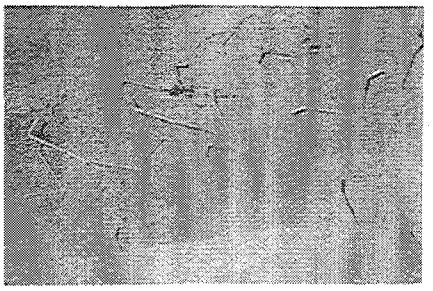
- Refractory Damage
- Repair cracks >3mm (1/8")
- Cutback refractory to heater shell
- John Zink Thermbond



FH 2004/12-104

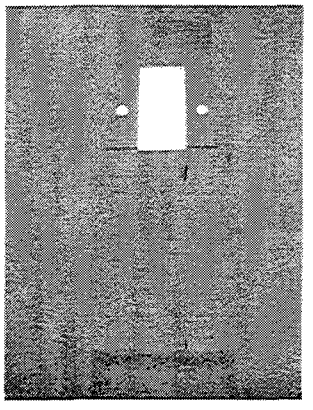
Castable Refractory

- Refractory Repairs
- V Style Anchors
- Penetration Weld Anchor
- Anchor Height and Density
- LHV Or Proprietary Mix



Castable Refractory

- Green Refractory
- Refractory Curing
- Refractory Dryout
- No Flow in Heater Tubes



Training Services

TYRI
Burner Inspection

Section Outline

- Burner Tiles
- Burner Tips

Burner Inspection

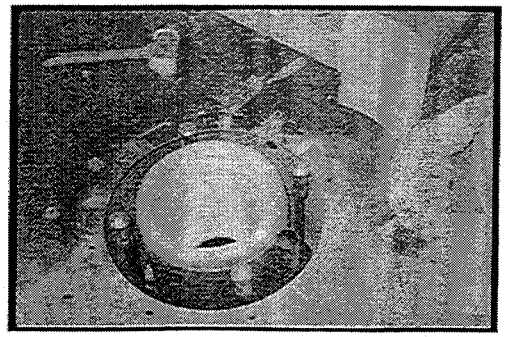
- Sodium vanadate attack on refractory
- Upgrade to 90% Alumina
- Burner tile damage caused oil spray on tubes
- High temperature caused tube failure



FH 2004/12-109

Burner Inspection

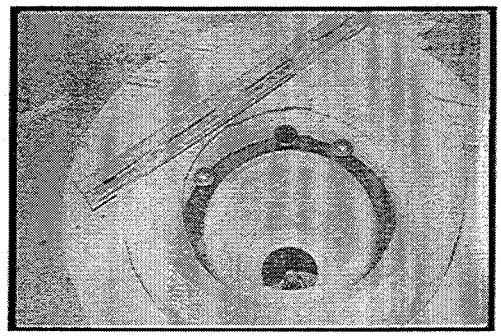
- Tile installation
- Secondary tiles come in eight pieces
- Use template for alignment
- 90% alumina primary tiles
- 60% alumina secondary tiles



FH 2004/12-110

Burner Inspection

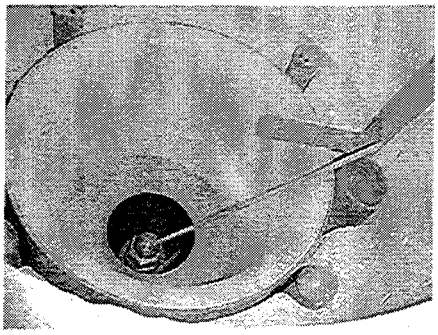
- Tile installation
- Secondary tiles come in eight pieces
- Use template for alignment
- 90% alumina primary tiles
- 60% alumina secondary tiles



FH 2004/12-111

Burner Inspection

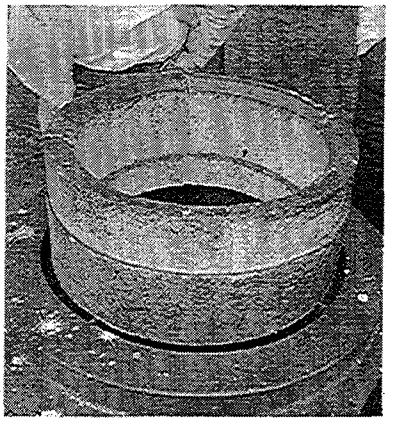
- Tile installation
- Use welding rod to make sure oil tip is proper height and will not impinge on tile
- Use original replacement parts



FH 2004/12-112

Burner Tile Inspection

- Staged Air Round Flame (Low Nox)
- 50% Combustion Air Outside Burner Tile
- Do Not Puck Annular Area



FH 2004/12-113

Burner Tip Inspection

- Correct Tip Drilling
- Do not mix parts from other burners
- Bent Gas Risers?
- Proper Orientation
- Clean Tips
- Assemble tips with high temperature Nickel based Anti-Seize

FH 2004/12-114

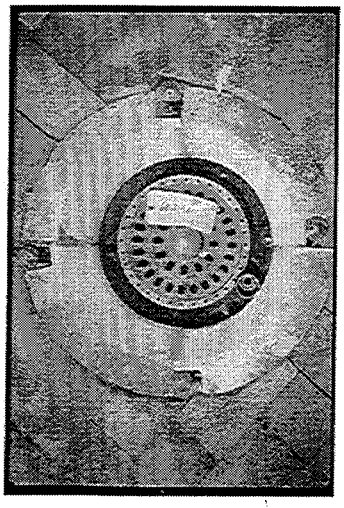
Burner Tip Inspection

- Combination Burner Tips
- Gas Tip burned off
- Gas dump though open pipe



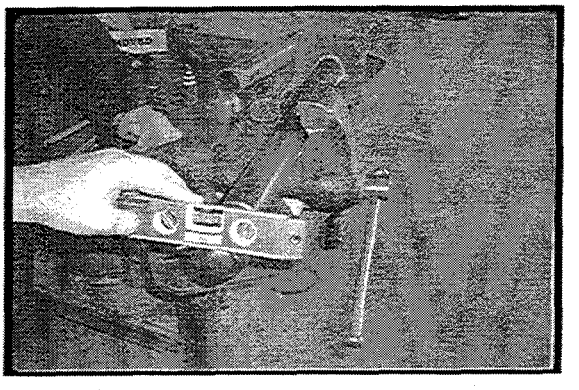
Burner Tip Inspection

- Staged Fuel Round Flame (Low Nox)
- Primary Tips Aligned to Secondary Tips
- Proper Tile Installation
- Uniform Annular Area



Burner Tip Inspection

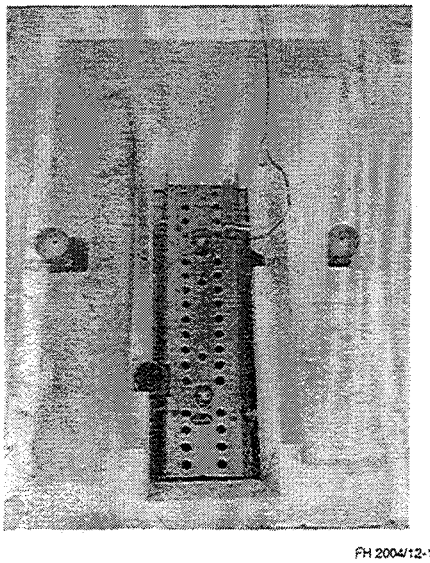
- Staged Gas Burner
- Alignment of primary tip
- Use drill bits to align tips



FH 2004/12-117

Burner Tip Inspection

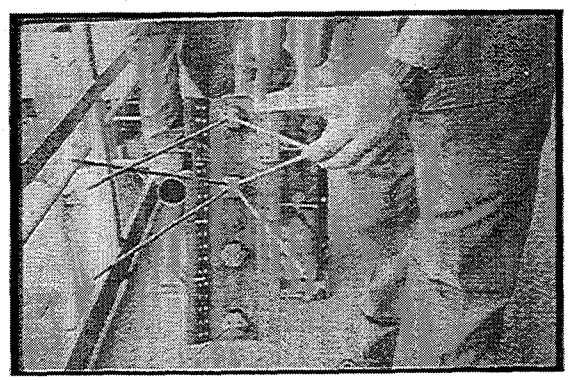
- Flat Flame Staged Fuel Burner
- Secondary Tips pointed to center



FH 2004/12-118

Burner Tip Inspection

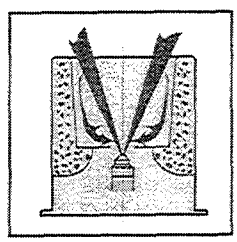
- Gas Burner
- Gas jets intersect
- Use welding rod to align tips



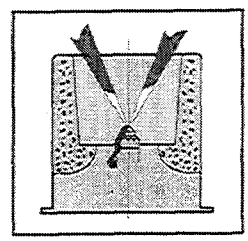
FH 2004/12-119

Oil Gun Location

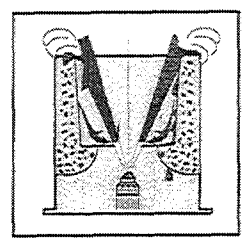
- Proper position is critical to proper spray pattern and air mixing



Proper location



Too far in

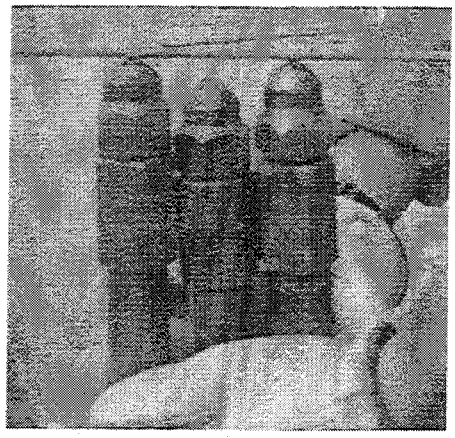


Too far back

FH 2004/12-120

Proper Oil Gun Length

- Zink No.9 EA Gun with 8 firing ports
- Oil Gun Inserts are different Lengths
- Impossible for operator to position properly
- Flame impingement, coking and oil drips result



Training Services

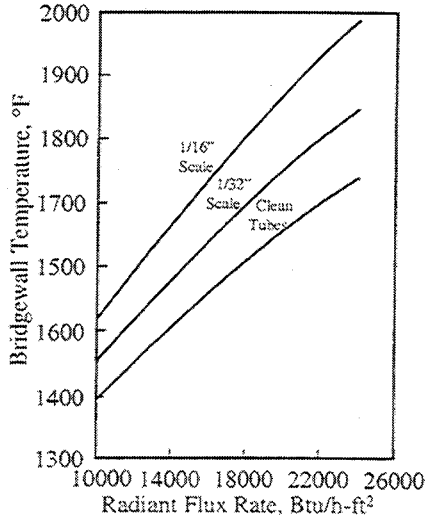
TYRI
Cetek Ceramic Coating

Oxidation Scale

- Platforming Heater- 9 Cr tubes
- Heavy Scale
- Increased fuel firing



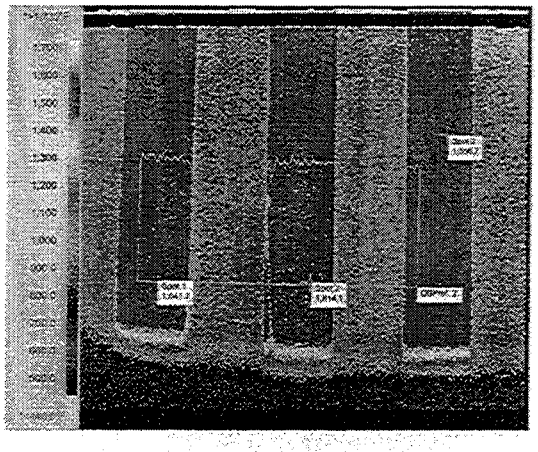
UOP Method Platforming Double Fired U Tube Heater



FH 2004/12-124

Infrared Survey

- **Platforming Heater**
- **Tubes one year after sandblasting**
- **Tight oxide scale layer**

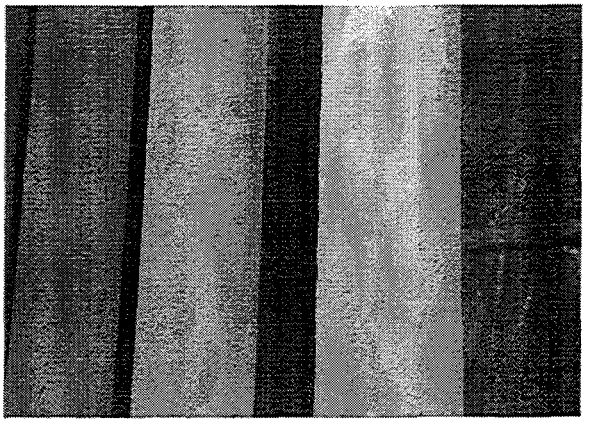


inframetrics ThermMonitor 95 2/22/99 11:26:44 AM

FH 2004/12-125

Oxidation Scale

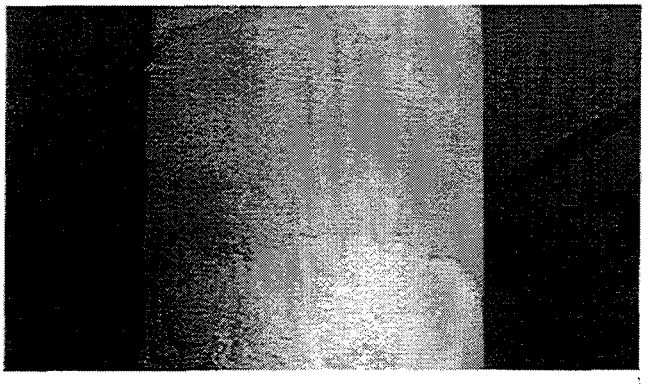
- 9 Cr Platforming Tubes
- 10-40 mils oxidation scale
- 200°F-400°F hotter surface temp



FH 2004/12-126

Oxidation Scale

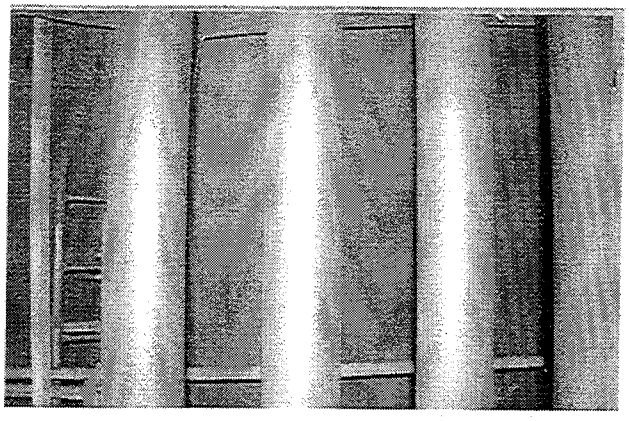
- 9 Cr Platforming Tubes
- 10-20 mils oxidation scale(cleaned with Black Beauty Blast)
- White Metal



FH 2004/12-127

Oxidation Scale

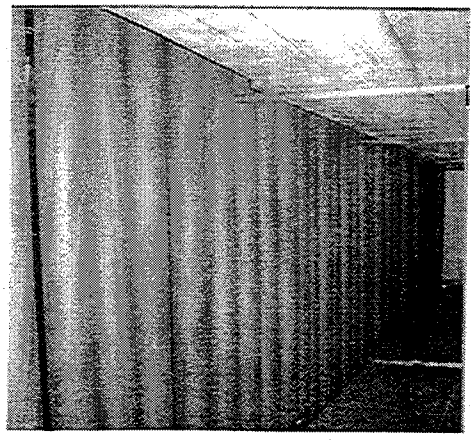
- 9 Cr Platforming Tubes
- Blasted to White Metal



FH 2004/12-128

Ceramic Coating

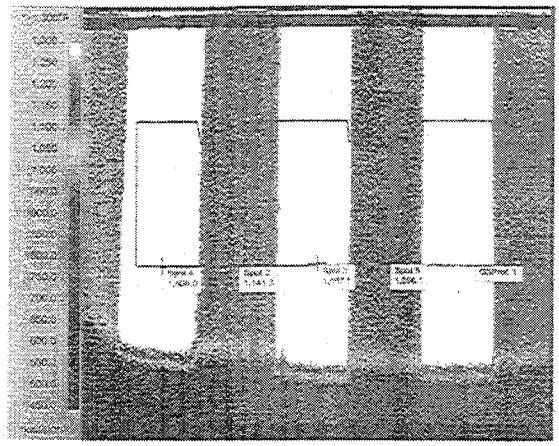
- 9 Cr Platforming Tubes
- Cetek Ceramic Tube Coating



FH 2004/12-129

Ceramic Coating

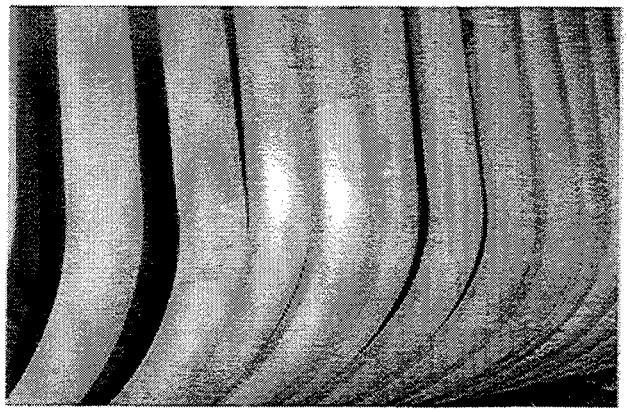
- Platforming Heater
- Tubes after sandblasting and Cetek Ceramic Coating
- More Process duty at same firing rate



FH 2004/12-130

Oxidation Scale

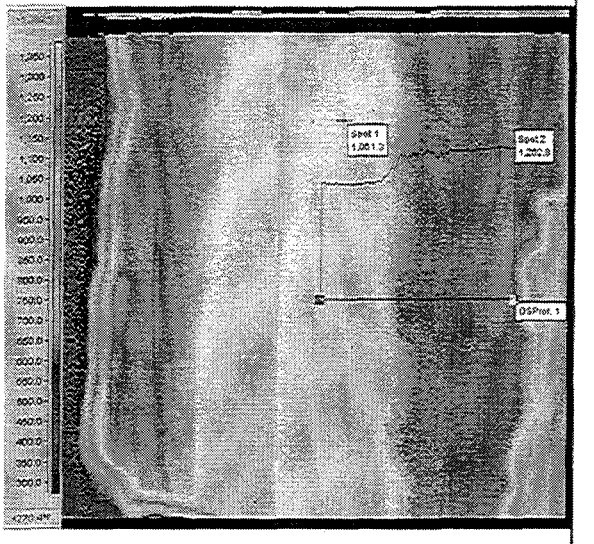
- 9 Cr Platforming Tubes
- White metal vs. Black Beauty Cleaning



FH 2004/12-131

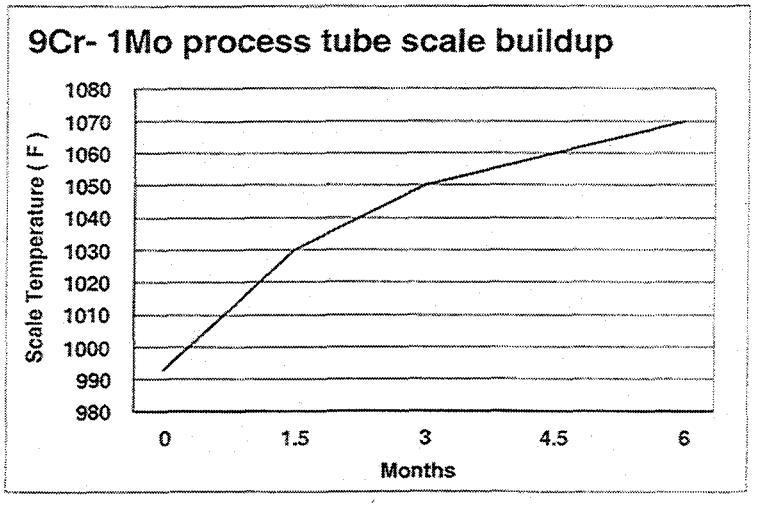
Oxidation Scale

- Thermal Shows White metal 200°F colder than tube cleaned with Black Beauty



FH 2004/12-132

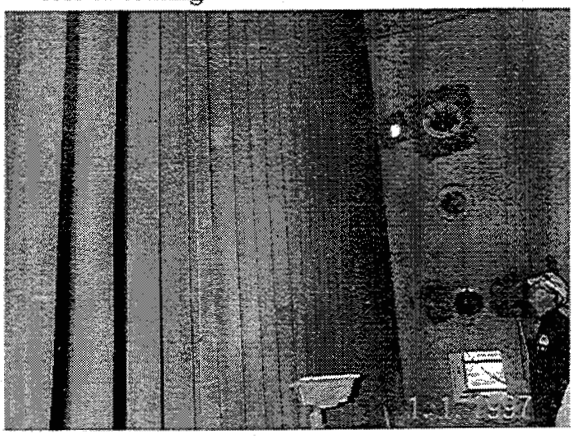
Oxidation Scale



FH 2004/12-133

Ceramic Coating

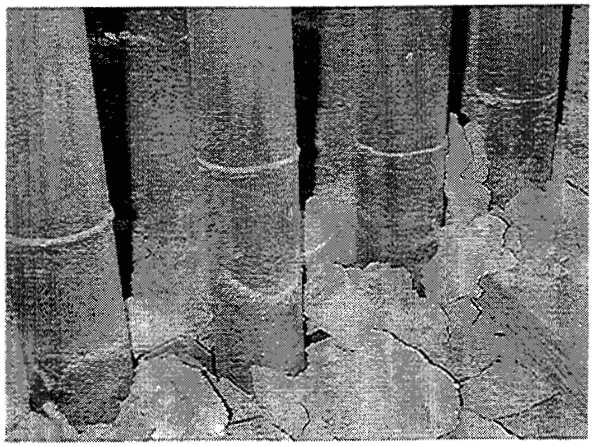
- 9 Cr Platforming Tubes after one year operation with Cetek ceramic coating
- 3 mil original thickness- no degradation or loss of coating



FH 2004/12-134

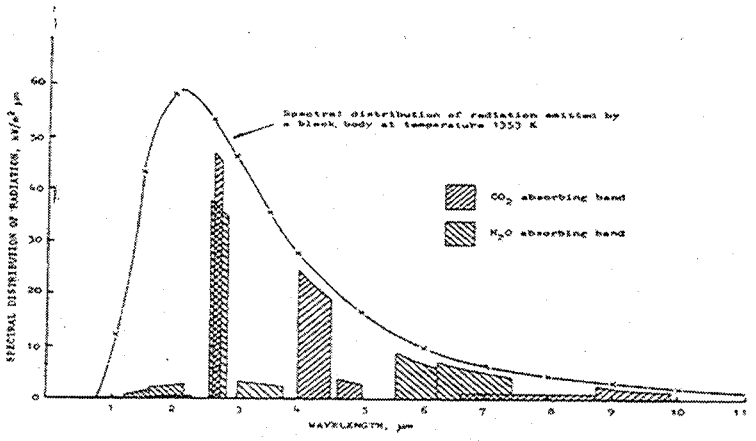
Ceramic Coating

- 9 Cr Platforming Tubes after one year operation with Cetek ceramic coating
- 16 mils oxidation scale on bottom of tube



FH 2004/12-135

Cetek Ceramic Coatings on Refractory

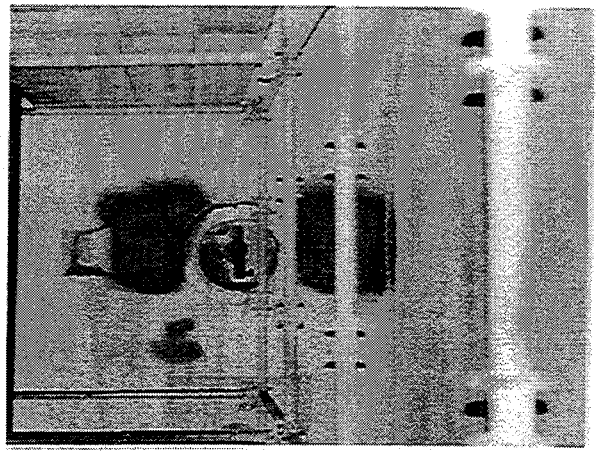


High emissivity surfaces are able to radiate energy across a broad wavelength band lessening the interference of the CO2 and H2O in the flue gas.

FH 2004/12-136

Cetek Ceramic Coatings on Refractory

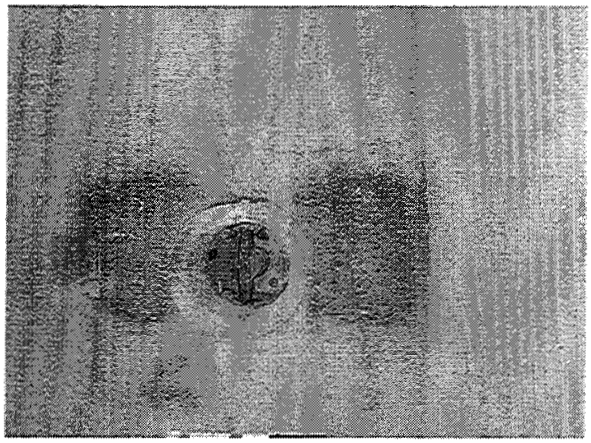
- Cetek Ceramic Refractory Coating



FH 2004/12-137

Cetek Ceramic Coatings on Refractory

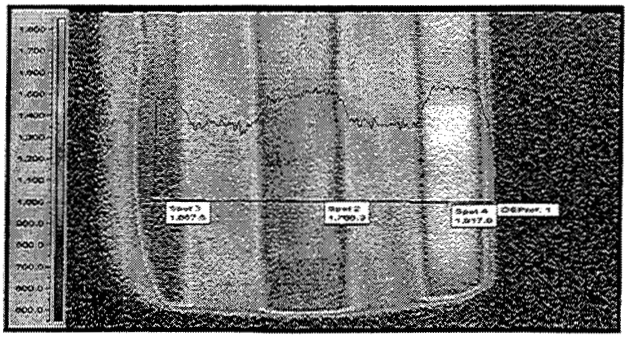
- Cetek Ceramic Refractory Coating after one year in service



FH 2004/12-138

Cetek Ceramic Coatings on Refractory

- Uncoated refractory patch= 1658°F(903°C)
- Coated refractory patch= 1818°F (992°C)

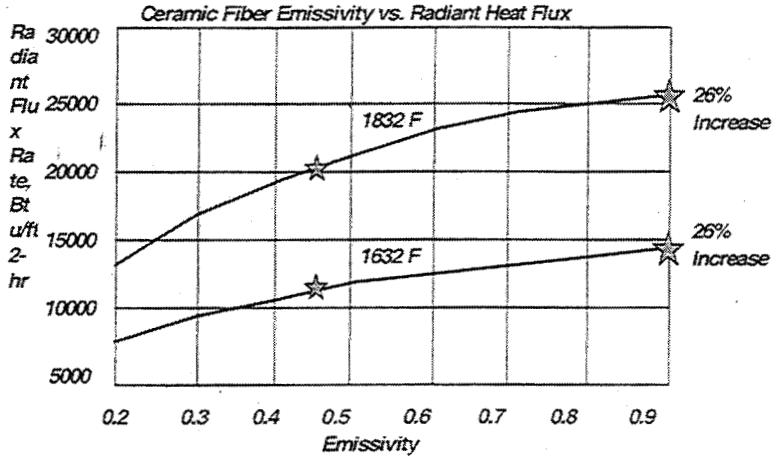


FH 2004/12-139

Cetek Ceramic Coatings on Refractory

Figure 2

Ceramic Fiber Emissivity vs. Radiant Heat Flux



FH 2004/12-140

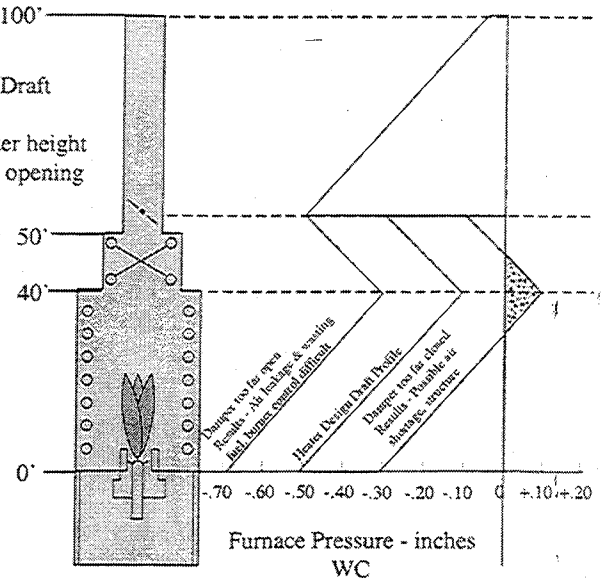
Training Services

TYRI
Heater Casing Inspection

FH 2004/12-141

Draft Profile

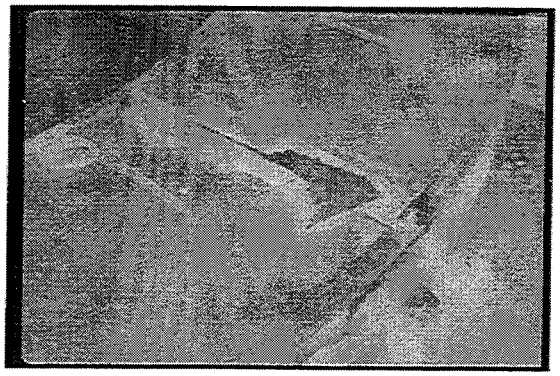
Fired Heater Draft Profiles according to heater height & Stack Damper opening



FH 2004/12-142

Heater Casing Damage

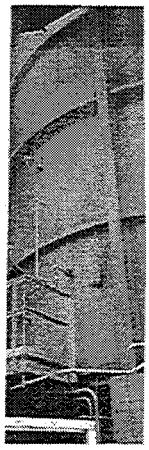
- Operating at Positive Pressure
- Acid Condensation Corrodes Steel
- Operating at negative pressure causes airs leaks into the heater



FH 2004/12-143

Heater Casing Inspection

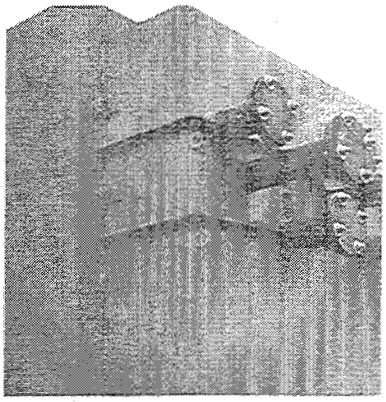
- Seal Welded Construction
- Seals at viewports
- High temperature silicone caulk



FH 2004/12-144

Heater Casing Inspection

- Are Header Boxes Sealed Properly?
- Are tube exit openings sealed?



FH 2004/12-145

Heater Casing Inspection

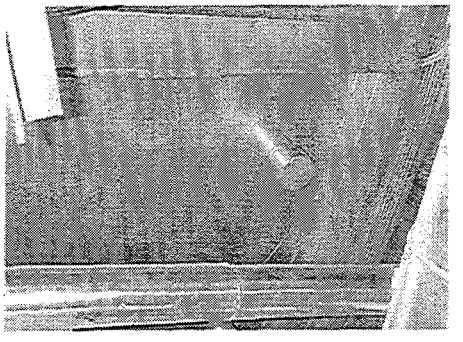
- Sockets are missing
- High tramp air leakage through opening around the pin



FH 2004/12-146

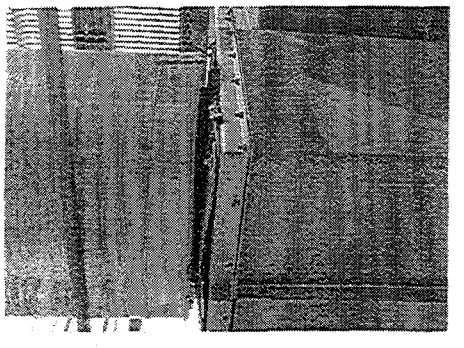
Heater Casing Inspection

- Remove pipe caps and clean debris from sockets at each turnaround on vertical tube heaters
- The installation of the pipe cap is the last item before startup



Heater Casing Inspection

- Manifold Seal is damaged
- Air leakage and potential thermal stress from rain ingress



Heater Casing Inspection

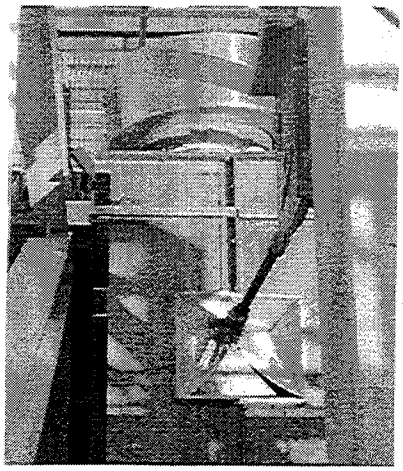
- Ceramic Fiber Cloth Seal
- Similar Seal used on Tube Exit
- Weather Shield Hot Flanges
- Tight Manifold Box (Gaskets)



FH 2004/12-149

Heater Casing Inspection

- Ceramic Fiber Cloth Seal on manifold is torn
- Rain will fall on hot manifold, possible thermal damage
- Need Weather Shield



FH 2004/12-150

REQUIRED TUBE THICKNESS

API Recommended Practice 530' **Calculation of Heater-Tube Thickness in Petroleum Refineries'** details procedures and design criteria for calculating the required wall thickness of refinery heater tubes. These procedures should be used to determine the required tube thickness.

Most heater tubes are purchased on a average wall basis (± 12.5%). The average wall is converted to a minimum wall basis by multiplying the average wall by 0.875. The corrosion allowance is subtracted from the minimum wall to obtain the **minimum thickness (t_m).**

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

$$t_m = t_S + CA$$

$$t_S = \frac{P_e D_0}{2S_e + P_e}$$

where: t_{min} = minimum tube wall thickness required, inches
t_S = stress thickness inches
P_E = design pressure, psig
D_O = tube outside diameter, inches
S_E =elastic allowable stress at maximum tube wall temperature, psi

Typical corrosion allowances range from 1/16" to 1/8".

UOP Practice is use temperature fraction and corrosion fraction = 1.0 in determining the required thickness.

The first edition of API 530 was based on average 100,000 -hour rupture strengths. Numerous companies experienced tube failures using this data. The second edition of API 530 was issued in 1978. The second edition is based on minimum 100,000-hour rupture strengths.

For example, the allowable stress for 2-1/4 Cr tubes was 7800 psi at 1100°F in the first edition. The second edition lower the allowable stress for 2-1/4 Cr tubes to 4550 psi at 1100°F. This large reduction in allowable stress means that many heaters built prior to 1978 have heater tubes that are too thin (usually by one Sch. No.).

PROCESS GUIDE

Process	Internal Scale	Typical Metallurgy	Typical Corrosion Allowance	Typical Failure Causes
Crude	Coke	9 Cr	1/8"	1, 3
Vacuum	Coke	9 Cr	1/8"	1, 3
Coker	Coke	9 Cr	1/8"	1, 3
Naphtha Hydrotreating	Inorganic	1-1/4 Cr to 9 Cr	1/8"- 3/16"	1,2
Kerosine/ Diesel Hydrotreating	Inorganic	347 SS	1/16"	1,2
Hydrocrackers	Inorganic	347 SS	1/16"	1,2,3
Catalytic Reformers	Clean	2-1/4 Cr to 9 Cr	1/16"	1,4
Reboilers	Clean	CS	1/8"	1

Typical Failure Causes

1) Overheating caused by low flow or flame impingment

2) Hydrogen sulfide attack

3) Sulfur attack

4) Long Term Creep Deformation

Tube Metal Temperature

The maximum tube metal temperature is calculated by methods listed in API 530 Appendix C. A rough rule of thumb is the maximum TWT is 75-100°F above the bulk temperature for common heaters on a scale free basis.

Internal Tube Scale

Inorganic scale has a thermal conductivity of 6 Btu/h- in-°F. Coke has a thermal conductivity of 30 Btu/h- in-°F.

At 19,000 Btu/h-ft^2 peak flux (10,000 avg. flux), a 1/32" internal inorganic scale deposit will raise the maximum TWT by 100°F.

At 19,000 Btu/h-ft^2 peak flux (10,000 avg. flux), a 5/32" internal cokedeposit will raise the maximum TWT by 100°F.

External Tube Scale

Flame impingement often results in heavy tube scale. Oil firing deposits an ash coating on the tubes. These deposits act as insulation and should be removed. Radiant heat transfer is T^4_{flame}-T^4_{tube}. The scale increases the surface temperature 200-300°F, requiring additional firing. Scale removal by light sandblasting or power brushing will reduce fuel firing by 5-7% on an all radiant heater.

Tube Skin Thermocouples

UOP specifies knife-edge skin thermocouples on coking or fouling services to be used as trend indicators to monitor the rate of fouling. Coking service includes coking, thermal cracking, and visbreaking heater coils. Fouling service includes desulfurization services including hydrotreating and Unibon heater coils. . The number of skin thermocouples is typically 3 to 6 per pass.

Many refiners install skin thermocouples on heater coils to monitor flame impingement. If there is a history of coking problems on the crude and vacuum heaters, there is justification for installing skin thermocouples on these services.

Skin thermocouples should be considered trend indicators, not absolute temperature indicators. Industrial test have proven that skin thermocouples usually read 30° to 60°C higher than the true wall temperature.

Service life is typically one to four years. Reliability can be improved only by frequent replacement, especially in severe oil firing services.

Burner Inspection

Many heater tube failures are caused by flame impingment. Flame impingment can be caused by improper burner setup. Inspect for the following problems

1) Correct burner tips
2) Correct tip location (height in tile, orientation)
3) Burner tile is concentric with proper diameter.

Tube Gauge Inspection

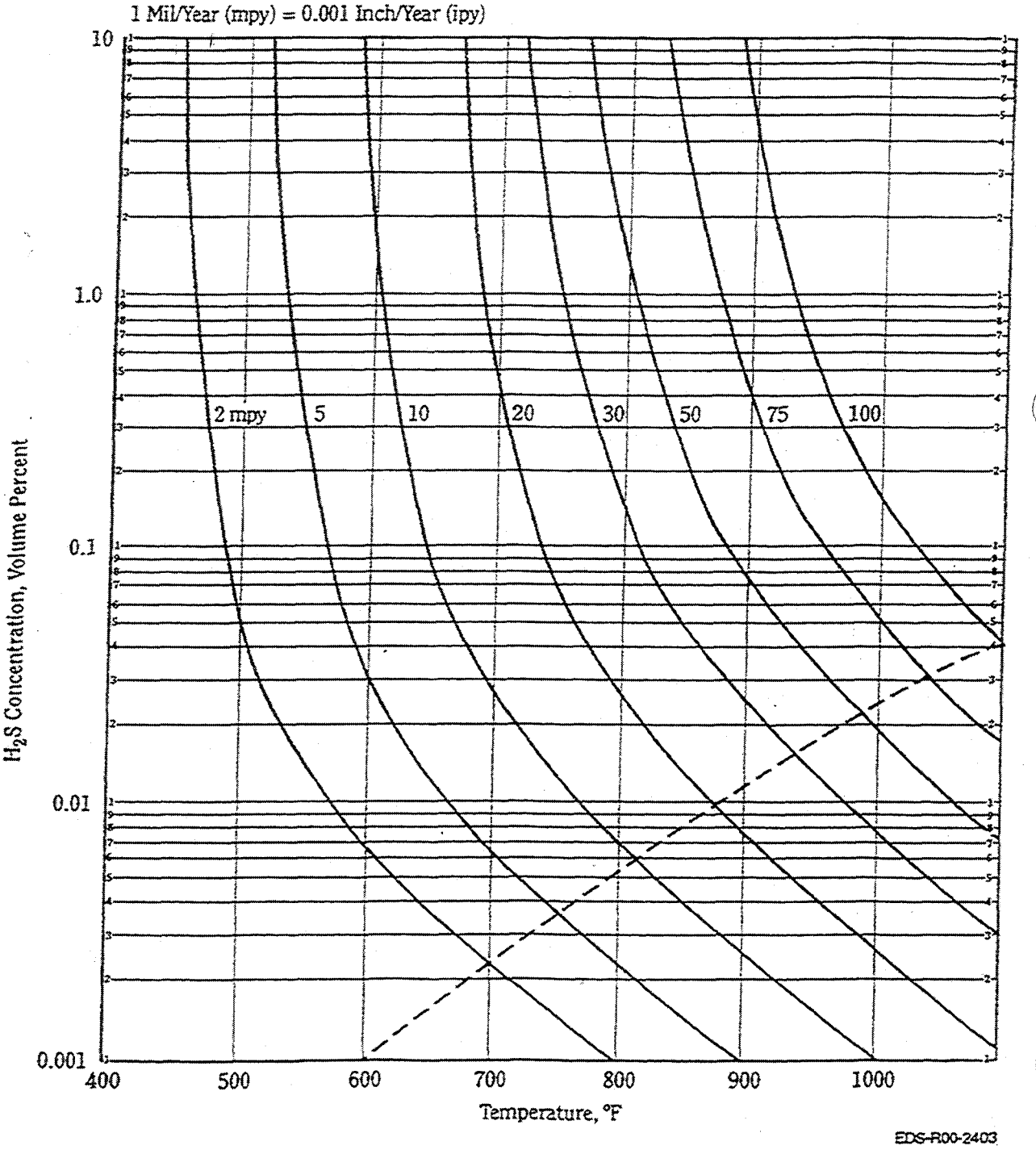
Flame impingement may have caused creep damage to the tubes. Creep damage is accumulated time-high temperature exposure. The refinery has a computer program based on API-530 to monitor remaining tube life. This program cannot account for localized overheating due to flame impingement.

Tube gauges are fabricated from thin SS plate (0.10"-0.125") . The cutout dimension is the sum of tube O.D. (4.5"), mill out of round tolerance per ASTM-450 (.0625"), scale allowance (.0625"), and creep growth. Three gauges should be made on 3%, 5%, and 10% creep growth.

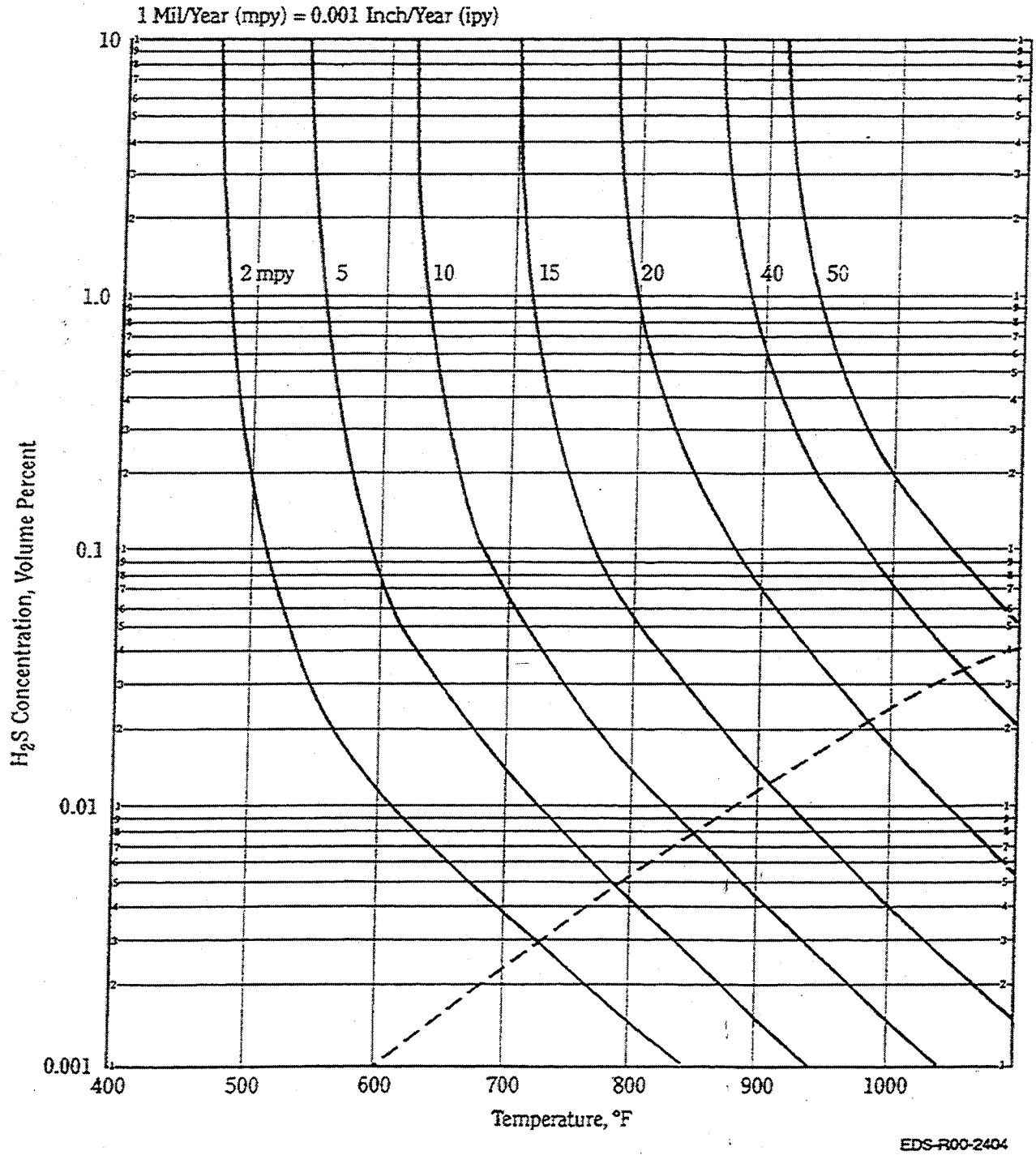
All radiant tubes should be checked with the tube gauges. The gauges will detect diametrical growth that is not apparent through visual inspection. Suspect areas should be checked by banding to determine the exact diametrical growth.

Hydrogen Sulfide

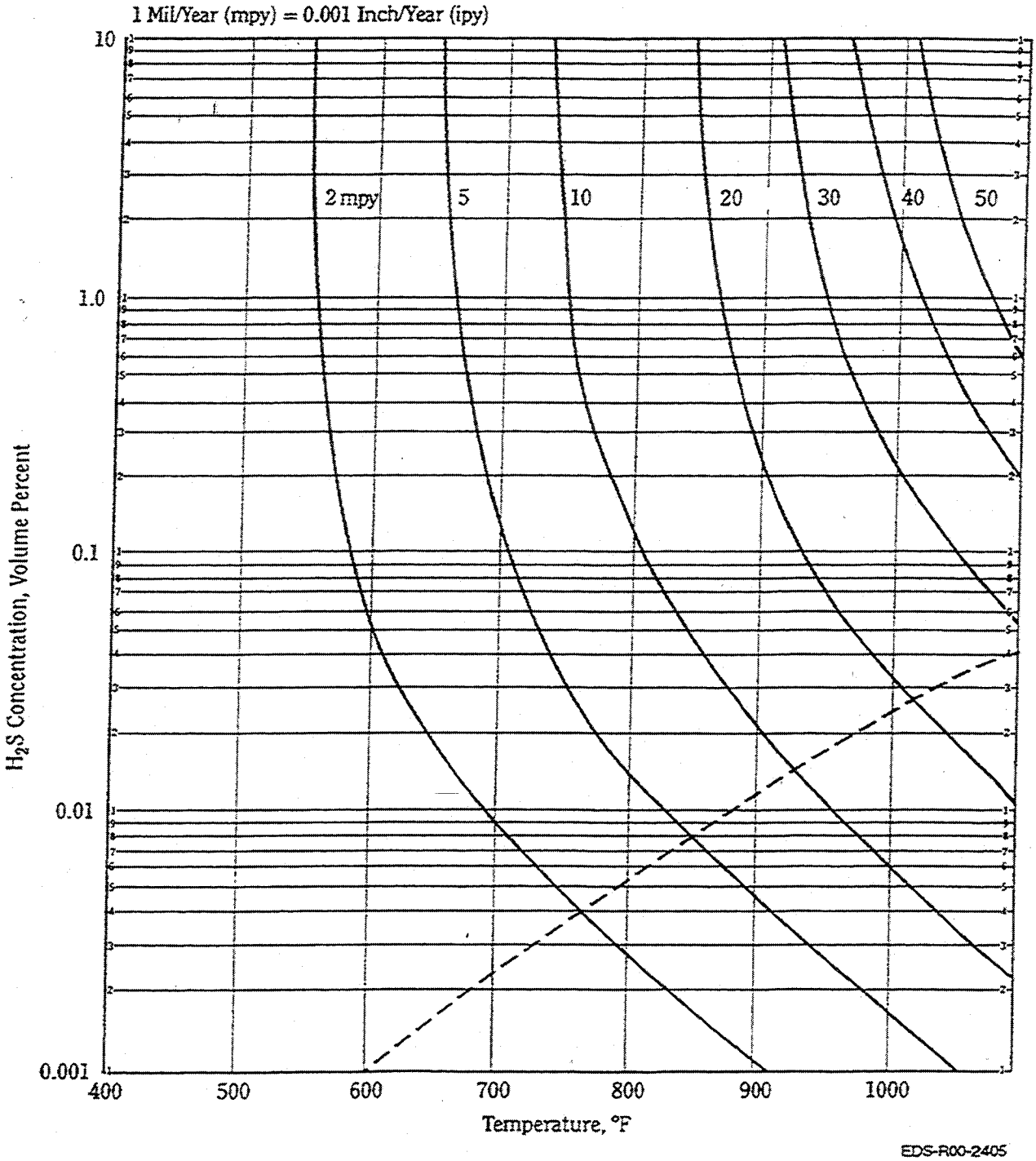
"Trend Curves for Predicting Corrosion Rates"
of 0-5% Chromium Alloy Steel in H_2S
(Predominantly Hydrogen + Hydrocarbon)
Environments at Elevated Temperatures



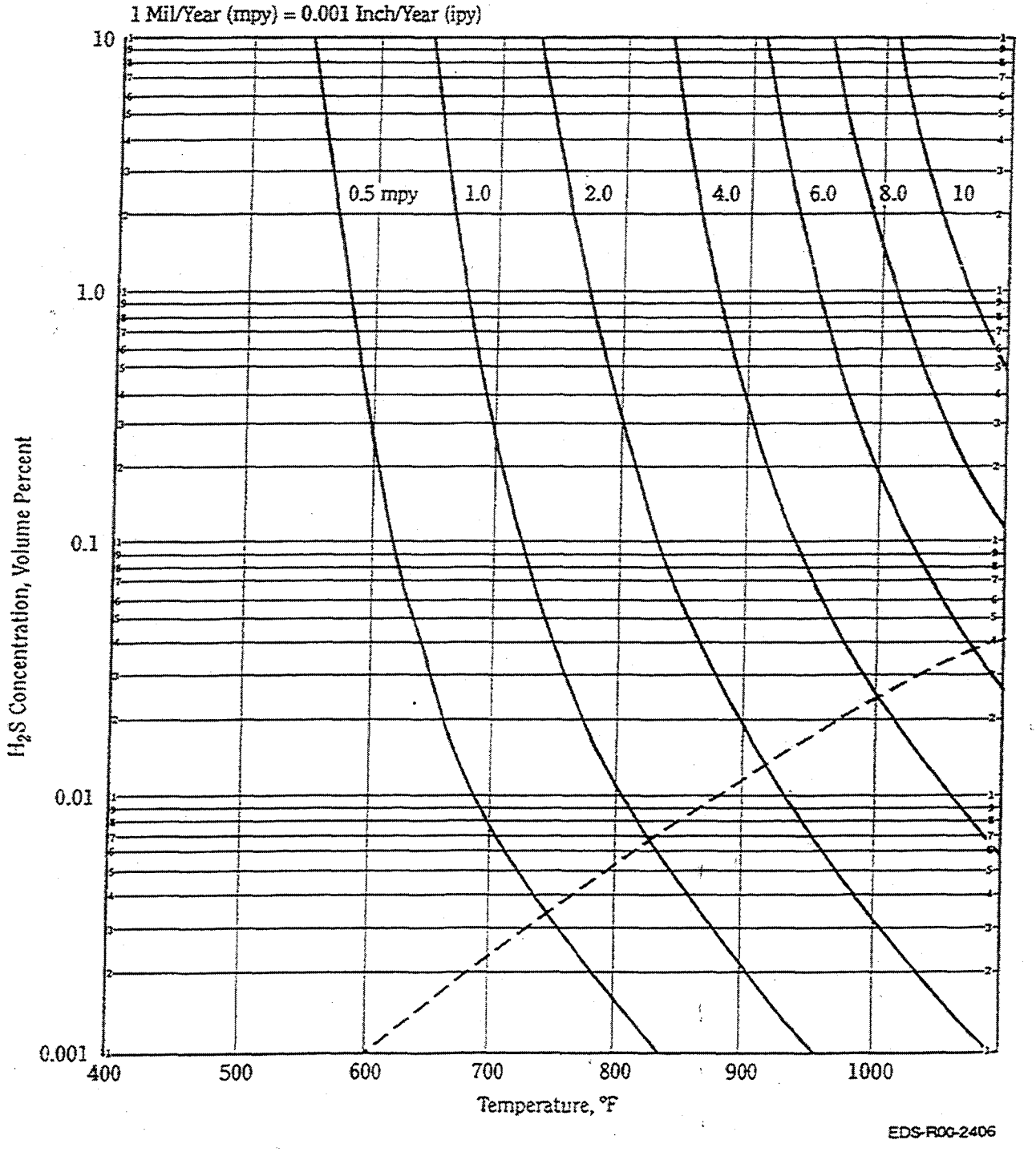
"Trend Curves for Predicting Corrosion Rates"
of 9% Chromium Alloy Steel in H_2S
(Predominantly Hydrogen + Hydrocarbon)
Environments at Elevated Temperatures



"Trend Curves for Predicting Corrosion Rates" of 11 - 13% Chromium Alloy Steel in H_2S (Predominantly Hydrogen + Hydrocarbon) Environments at Elevated Temperatures



"Trend Curves for Predicting Corrosion Rates"
of 18% Cr - 8% Ni Alloy Steel in H_2S
(Predominantly Hydrogen + Hydrocarbon)
Environments at Elevated Temperatures



Burner Installation

Most heater problems can be traced to burner problems and flame impingement. Initial installation mistakes are often the cause. Probably 90% of new heaters do not have the proper burner installation. Be sure to include burner inspection on the heater checkout list.

Obtain the burner drawings from the contractor. It will include the installation tolerances. The following instructions are general rules. Always follow the information on the specific burner drawing.

Burner Tile

The burner tile is an air orifice. It controls the amount of air flow. The installation tolerance is ± 1/8" on the diameter shown on the burner drawing. Measure the diameter in 3-4 locations. Most round tiles are installed as a slight oval shape. This will result in poor fuel -air mixing and a bad flame shape.

Gas Tip Location (Standard Burners)

The installation tolerance is ± 1/8" on horizontal spacing and ± 1/4" on vertical spacing. Measure tip to tip dimensions. Bent gas risers can cause these dimensions to be wrong. Lay a flat edge across the burner tile. Measure down to the gas tip to obtain the vertical dimension.

The VYD burner has one center gas tip. The bottom row of 1/16" ignition ports should be flush with the stabilizer cone. Measure tip to tile dimensions. Most of these burners have tiles installed off-center.

Gas Tips Location (Staged fuel Burners)

The primary gas riser is welded to a rectangular base plate. Align the four holes to the corners of the base plate. This will insure that the primary tips holes point to the secondary tips. The primary tip flames act as ignition sources for the secondary tips.

The secondary tips sit above the burner tile. The secondary tips have 1/16" ignition ports drilled horizontally toward the center of the burner tile. Measure from the top of the tile to the ignition ports. The distance should be 5/8". The installation tolerance is ± 1/8"

Burner Installation

Gas Tip Orientation (Standard Burners)

The tips have arrows or cutouts to aid in tip orientation. The arrows or tips point to the tile. The PMA burner has 4-6 gas tips. The burner is designed for gas jets to intersection in the center of the burner and dissipate the high energy.

Oil Tip Orientation

The tip should be below the primary oil tile. There should be 1" from the flat ledge of the oil tile to the centerline of the oil port.

Pilot Tip Location

The tip is usually level with the flame stabilizer cone. It is either level with the primary tile on oil burners or 1-1/2" below the primary tile.

HEVD Tip Location (Premix gas spider head)

The tip is flush to the top of the tile on straight tile designs. It is located one inch below the top on tiles with a 1" bevel.

PFH Tip Location (Premix gas flat flame head)

The edge of the tip is flush with the side bevel of the tile. This puts the top of the tip ~ 2-1/2" below the top of the tile.

Set Screws

Many of the older designs had set screws on the gas tips, instead of the welded construction used today. The tips would be pushed up into the burner for shipping and reset in the field. Many of these older burners have tip alignment problems, especially when they are hard piped.

HYDROTREATER FURNCES

SANDJET Furnace Tube
Decoking Service

Hydrotreaters are important units in today's refining process due to sour crude feedstocks and the need to upgrade more of the barrel to higher octane fuels. Furnace temperatures in these units thermally decompose some feedstock compounds, forming coke on the inner walls of the furnace tubes. "Tramp" iron from refinery piping and equipment combines with the sulfur impurity to form a non-combustible scale. In heavy oil or residual hydroprocessors, metals from the feedstock can also create non-combustible scale. Growth of these deposits restricts process flow and inhibits heat transfer. Eventually, the hydrotreater must be shut down for decoking.

Decoking with the SANDJET Service removes furnace tube deposits through low angle impacts of particles propelled in a high-velocity stream of pure nitrogen gas. The cleaning material, composed of steel shot or similar particles, fractures the deposit from the furnace tube walls. Deposits and cleaning material are then carried away in the nitrogen stream, leaving clean, dry tube walls.

Less Downtime for More Production

SANDJET Furnace Tube Decoking Service from UCISCO is a fast and effective way to decoke hydrotreater crude furnaces. For example: A typical SANDJET Service decoking operation takes 3 to 4 hours per pass. Thus a two-pass furnace can be completely decoked and descaled within a single shift, shortening turnaround for more production.

Reduces Operating Cost

The non-combustible deposits, which are not removed by steam air decoking, reduce heat transfer efficiency. The SANDJET Service also removes these deposits resulting in improved heat transfer and a lower pressure drop. Better efficiency means less energy dollars and a lower operating cost.

Longer Production Runs

Complete cleaning of coke and non-combustible scale removes the nucleation sites for coke formation. This reduces the rate of coke deposition, yielding longer production runs between decokings.

No Hazardous Emissions

The SANDJET Service is a mechanical process that uses no hazardous chemicals, forms no toxic or polluting gases, and produces no hazardous emissions.

Longer Tube Life for Less Replacement Cost

Decoking using the SANDJET Service does not subject furnace tubes to thermal shock and extreme temperatures, which can cause metallurgical damage and tube failures. With SANDJET Service, furnace tubes are not exposed to temperatures in excess of ambient conditions, therefore tube life is longer and replacement cost is less.

SANDJET Improves Furnace Efficiency in a Midwest Refinery

A hydrotreater furnace in a Midwest refinery had historically been cleaned using steam air decoking followed by chemical cleaning. As the graph shows, stack gas temperatures rose abruptly a few weeks after such cleaning. The refiner responded by reducing excess air below optimum levels. After further temperature rises mandated two more such reductions, the furnace had to be shutdown because of high tube skin temperatures.

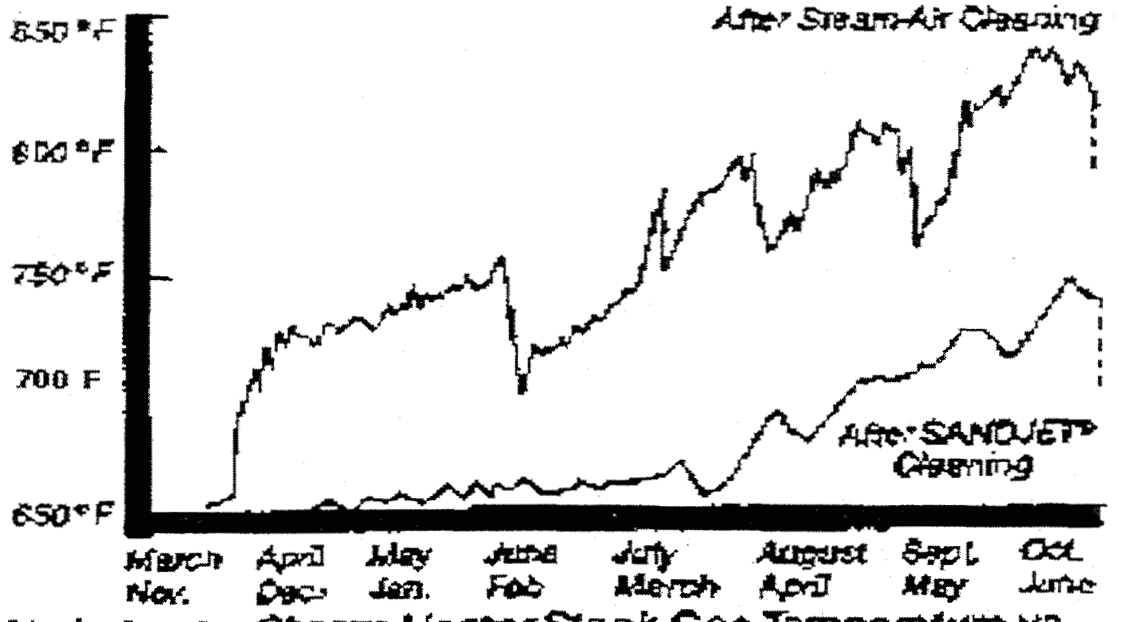
When the same furnace was cleaned with the SANDJET Services, stack gas temperature rose at a much lower rate for the first three months. At the end of this period, final stack gas temperature was only about 40°F higher than the initial temperature after conventional cleaning. This data verifies the SANDJET Service benefits of lower operating cost and longer production runs.

A refinery in Texas had a hydrotreater with a coke deposit that was 75% non-combustible. Thus, steam air decoking was unsuccessful. SANDJET Service was used and the problem deposit was quickly removed. This refiner reported a 14.5% fuel saving after SANDJET as compared to

"Courtesy of UCISCO"

previous cleanings: further proof that the SANDJET Service solves difficult cleaning problems and reduces operating costs.

Backed by the technological expertise of the Union Carbide Corporation, UCISCO's service personnel provide the highest quality cleaning possible with minimum downtime and environmental impact. Years of operational experience and research are available 24 hours a day to meet unscheduled or emergency needs. UCISC0 service representatives are always available at your convenience to provide an effective solution to your cleaning problem.



Hydrotreater Charge Heater Stack Gas Temperature vs. Operating Cycle

Drying and Heat-Up of Monolithic Refractories

Overview

Refractory castables must be cured for a minimum of 24 hours prior to drying out and heat-up of the lining. Longer curing times may be required at cold temperatures.

Since refractory castables have lower permeability and higher moisture content than brick or plastic refractories, they should be heated up slowly after the curing period to allow moisture to escape gradually from the castable. If the castable is heated too rapidly, high pressure steam may be generated within it which could cause the material to rupture and spall. For this reason, refractory castables should never be heated with direct flame impingement on the surface of the lining. Weep holes may be incorporated through the steel casing on the cold face of some installations to open a path for the escape of steam during heat-up of thick castable linings.

Refractory castables containing high purity calcium aluminate cement develop a more stable structure when cured above 70°F (21°C). For this reason, castables made with this binder and exceeding 70°F (21°C) during curing may be heated more safely during drying and heat-up than the same castables cured below that temperature.

Heating a castable lining too fast can also cause cracking on the cold face which extends into the castable lining. Rapid heating causes the hot face to dry out and heat up while the remaining castable still contains considerable water. The differential expansion resulting from a rapid heating rate causes cracks to develop in the castable lining which lowers the stability of the monolithic structure and provides pathways for penetration by hot gases and liquid metals or slags.

The following initial heating schedules, although not all inclusive, will serve as guides to determine the proper heating program for most refractory castable linings. These schedules are based on one-component linings up to 9 inches thick.

"Courtesy of Harbison-Walker Refractories"

Initial Heating Schedules

Schedule 1

The temperature of the gases in contact with the castable are increased 100°F (38°C) per hour to service temperature. This rate should not be exceeded. Hold periods of one-half hour per inch thickness of castable Moth be incorporated at 250°F (120°C) and 500°F (260°C).

Schedule 2

The temperature of the gases in contact with the castables are increased 75°F (24°C) per hour to service temperature with hold times of one-half hour per inch thickness of castable at 250°F (120°C), 500°F (260°C) and 1000°F (540°C).

Schedule 3

The temperature of the gases in contact with the castables are increased 50°F (10°C) per hour to service temperature with hold times of one-half hour per inch thickness of castable at 250°F (120°C), 500°F (260°C), 1000°F (540°C) and 1500°F (815°C).

Schedule 4

The temperature of the gases in contact with the castables are increased 30°F (1°C) per hour to service temperature with hold times of one-half hour per inch thickness of castable at 250°F (120°C), 500°F (260°C), 1000°F (540°C) and 1500°F (815°C).

Consider the following factors when selecting a schedule:

The temperatures referred to in the schedules are the temperatures of the hot gases in contact with the castable and not the temperature of the castable itself. Thermocouples may be placed about 1/2-inch away from the castable surface for an accurate measure of temperature during heat-up.

At least one thermocouple should be placed where the temperature of the hot gases will be the highest to avoid localized overheating.

Castable linings may be heated at a lower rate if the temperature cannot be controlled accurately or if there is a likelihood of localized overheating in large furnaces.

These heat-ups are for castable linings up to 9-inch thick. Linings thicker than 9 inches will require a slower heating schedule. Contact your local Harbison-Walker representative if in doubt as to best recommendation.

Linings composed of two or more castable components may require a slower schedule. This is most important when a lightweight castable is installed behind a dense castable working lining in a vapor tight steel shell. Since all the water in the insulating castable may have to be removed through the dense castable lining, a slower schedule is advised.

If the furnace is to be cooled before being put into service, the temperature should be held at service temperature until all moisture is removed and equilibrium is established in the lining. The hold time required may vary from eight hours for small furnaces with thin linings to several days for large furnaces with thick linings.

The selection of the proper heating schedule is dependent on the design of the refractory lining and the particular type of furnace.

If steaming is observed during any hold period, the hold period should be maintained until steaming subsides.

UOP POLICY ON SKIN THERMOCOUPLES

UOP specifies knife-edge skin thermocouples on coking or fouling services to be used as trend indicators to monitor the rate of fouling. Coking service includes coking, thermal cracking, and visbreaking heater coils. Fouling service includes desulfurization services including hydrotreating and Unibon heater coils. The number of skin thermocouples is typically 3 to 6 per pass.

Many refiners install skin thermocouples on heater coils to monitor flame impingement. If there is a history of coking problems on the crude and vacuum heaters, there is justification for installing skin thermocouples on these services.

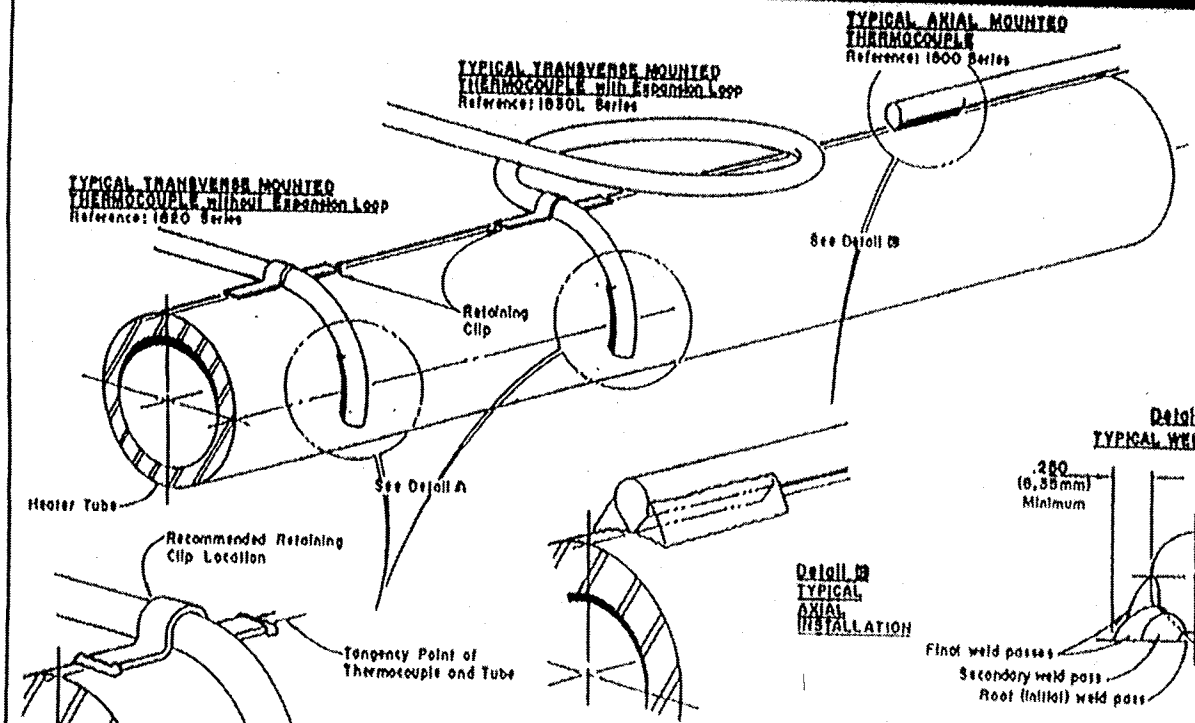
Skin thermocouples should be considered trend indicators, not absolute temperature indicators. Industrial test have proven that skin thermocouples usually read 30° to 60°C higher than the true wall temperature.

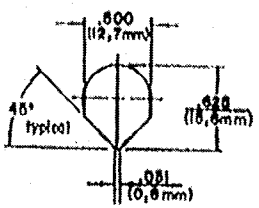
Service life is typically one to four years. Reliability can be improved only by frequent replacement, especially in severe oil firing services.

Most heaters have TWTs that are 75-100°F above the process temperature. These heaters include all reboilers, crude heaters, NHT Charge heaters, and Tatoray Charge.

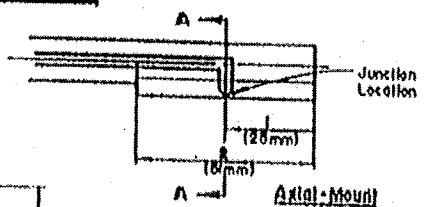
Platforming, and Pacol heaters have TWTs that are ~ 150°F above the process temperature.
Oleflex heaters have TWT's that are ~ 50°F above the process temperature.

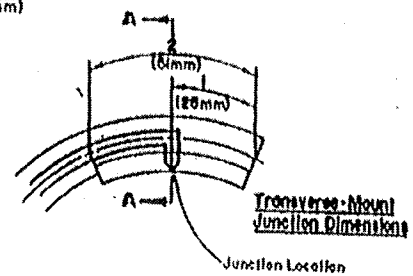
TUBE-TEMP INSTALLATION DETAILS



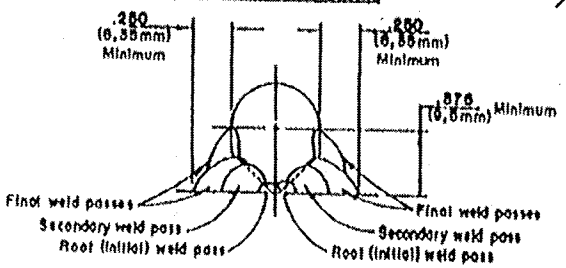


SECTION A-A Typical shape full length of 2" (51mm) long 'vee' pad.





Detail C
TYPICAL WELDING DETAIL



INSTALLATION AND WELDING PROCEDURE

1. Grind surface of Heater Tube in area of thermocouple junction location to remove rust and scale. Clean area approximately 3" (76mm) long, in direction of thermocouple axis, by approximately 2" (51mm) wide.
2. Locate and clamp thermocouple firmly in desired location. Center of 2" (51mm) long 'vee' pad must contact tube.
3. Select proper weld-filler rod from table. Welding method: Tungsten Inert Gas, Metal Arc Process (TIG, Heli-Arc).
4. Perform root weld pass on both sides of 'vee' using .062 (1,57mm) Ø filler rod. Welds must overlap each other at juncture of 'vee' and tube and run full length of 'vee' pad. (See Detail 'C')
5. Perform secondary weld pass on both sides of 'vee' using .093 (2,36mm) Ø filler rod. Welds to run full length of 'vee' pad. (See Detail 'C')
6. Perform final weld passes on both sides of 'vee' using .093 (2,36mm) Ø filler rod. Welds must extend .375 (9,5 mm) minimum above tube surface and run full length of 'vee' pad. (See Detail 'C')
7. Transverse mounted thermocouples: Locate Retaining Clip at tangency point of thermocouple and tube and weld at both ends only using .093 (2,36mm) Ø filler rod. (See Detail 'A')
8. Axial mounted thermocouples: Locate Retaining Clip as desired and weld same as step 7.
9. Assemblies requiring on-site 'field' forming or reforming to adjust fit must not be 'cold' formed. Portions of thermocouple to be formed or reformed must be heated to 1500 to 1800°F (760 to 818°C) during forming operations. 2" (51mm) minimum bend radius recommended. Do not attempt to reform 2" (51mm) long 'vee' pad containing junction.

RECOMMENDED WELD FILLER ROD MATERIALS FOR SHEATHED 'TUBE-TEMP' ASSEMBLIES

HEATER TUBE MATERIAL		FILLER ROD MATERIAL
ASTM A 312, TP 304, TP 309 or TP 310		80999 - AWS A5.9, Class ER309
ASTM A 312, TP 321	(18% Cr, 10% Ni) Ti stabilized	
ASTM A 312, TP 347	(18% Cr, 10% Ni) Cb stabilized	
ASTM A 335, P 11	(1-¼% Cr, ½% Mo)	Inconel 82 - AWS A5.14, Class ER NiCr 3 ⚠1
ASTM A 335, P 22	(2-¼% Cr, 1% Mo)	
ASTM A 335, P 5	(5% Cr, ½% Mo)	
ASTM A 335, P 9	(9% Cr, 1% Mo)	
ASTM A 200, T 22	(2-¼% Cr, 1% Mo)	
ASTM A 161, T 1	(.18% C max., .5% Mo)	
ASTM A 106	Carbon Steel	
HK-40		80999-AWS A5.9, Class ER309
INCOLOY 800		Inconel 82-AWS A5.14, Class ER NiCr 3 ⚠1

REV	DESCRIPTION	DATE	APPROVED
⚠1	TYPICAL ... Rev. 1488-88		
⚠2	...		
⚠3	...		
⚠4	...		

THERMO COUPLE PRODUCTS COMPANY
Montreal (QUE.) Canada

INSTALLATION DETAILS for 'Tube-Temp' THERMOCOUPLES

T.C.P. Design Standard

SPEC-117 ⚠4

Training Services

New Heater Specification

FH 2004/13-1

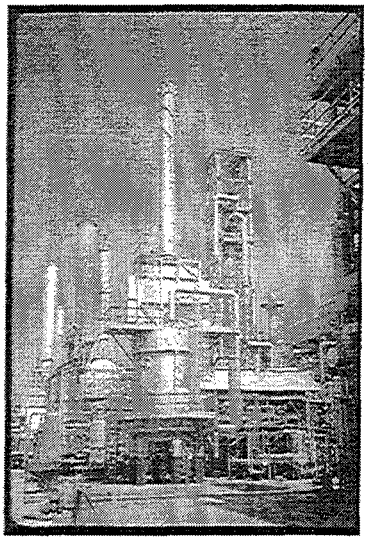
Fired Heater Specification

Process	Radiant Flux, Btu/h-ft^2	Mass Velocity, lb/sec-ft^2	Process ΔP, psi	Coil Metallurgy	Type of Heater
Crude	10,000	200 min	150	5 Cr or 9 Cr	Vertical Tube Box Cabin or V.C.
Vacuum	8,000	250 min	75	5 Cr or 9 Cr	Vertical Tube/Box Cabin or V.C.
Visbreaking	7,000-8,000	350 min (5-6 fps cold oil)	250-300 clean	9 Cr	Cabin with brick centerwall
Coking	9,000	400 min (6 fps cold oil)	300 clean	9 Cr	Cabin with brick centerwall or double fired horizontal tube
Semi-Regn. Platformer	10,000	40-50	7	2-1/4 Cr	V.C., cabin, wicket
CCR Platformer	10,000 SFU 15,000 DFU	23-35	3 per cell	9 Cr	Single Fired U-tube Double Fired U-tube
Reboiler	10,000	150-300	50	C.S.	V.C.
Hot Oil	10,000	400-600	65	C.S.	V.C.
Naphtha Hydrotreater (max. 15 mils/yr)	10,000	70-180	50	9 Cr	V.C.
Naphtha Hydrotreater (with cracked stock)	10,000	70-180	50	347	V.C.
Kerosene Hydrotreater	10,000	150-250	50	347H SS	V.C.
Diesel Hydrotreater	10,000	150-250	50	347H SS	V.C. or double fired vertical tube
Hydrocracker Gas Only	10,000 - 13,000	50-80	30-50	347H SS	double fired vertical tube
Hydrocracker Gas&Liquid	15,000	180-300	50-80	347H SS	double fired vertical tube

FH 2004/13-2

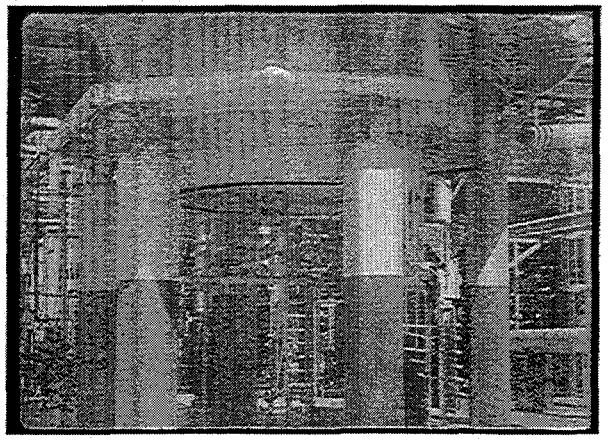
Vertical Cylindrical Heater

- Most Common Heater
- Least Cost
- Smallest Plot
- Easy to Construct
- Good for small duties
- Maximum process duty is 170 MM Btu/h



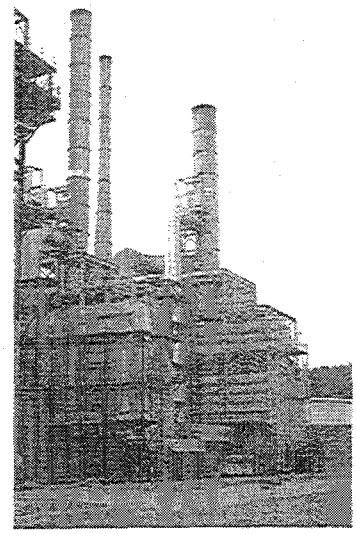
Vertical Cylindrical Heater

- Firing Valves under Heater (American Method)
- Firing Valves at viewport (European Method)



Cabin Heater

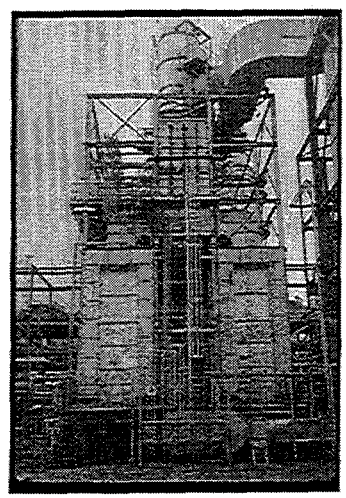
- Horizontal Tubes
- 15-20% More cost
- Large Plot
- Free Draining
- Used in Black Oil Services



FH 2004/13-5

Cabin Heater

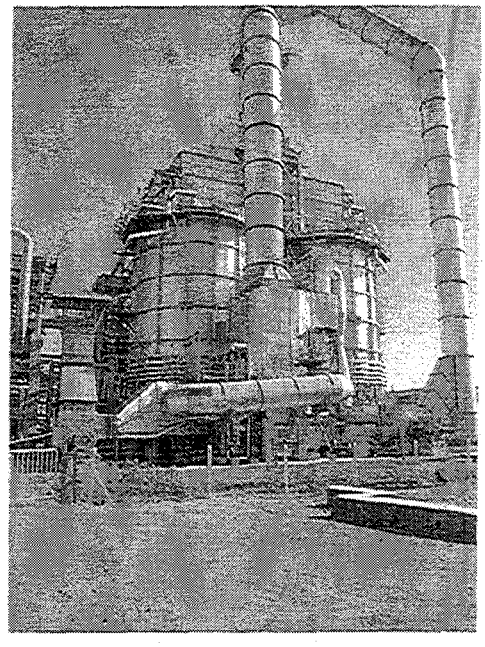
- Two Radiant Sections
- Common Convection
- Geometry Concerns
- L/D Ratio 2-2.5
- Two radiant with one convection is also used with large VC heaters above 170 MM Btu/h (50 MW) process duty



FH 2004/13-6

Large Crude Heater

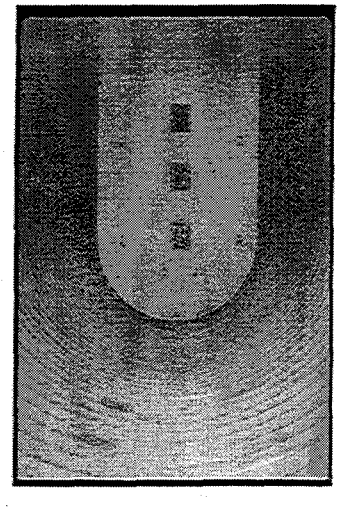
- Two Radiant Sections
- Common Convection
- Geometry Concerns
- L/D Ratio 1.5
- Two radiant with one convection is also used with large VC heaters above 170 MM Btu/h (50 MW) process duty



FH 2004/13-7

U-Tube Platforming Heater

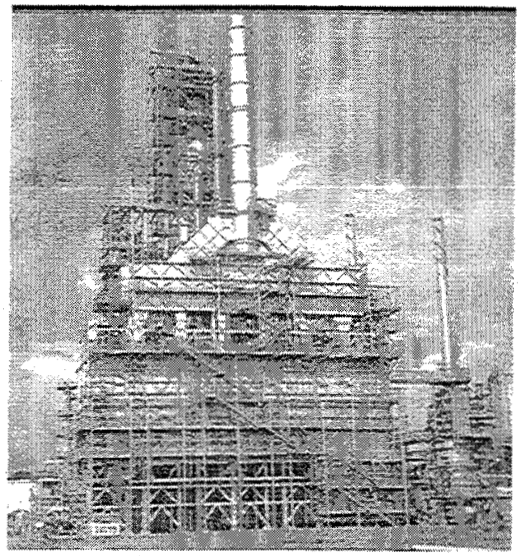
- Low Pressure Drop
- 5'-6' coil radius
- Horizontal Firing
- Single and Double Fired
- 9 Cr Metallurgy



FH 2004/13-8

U-Tube Platforming Heater

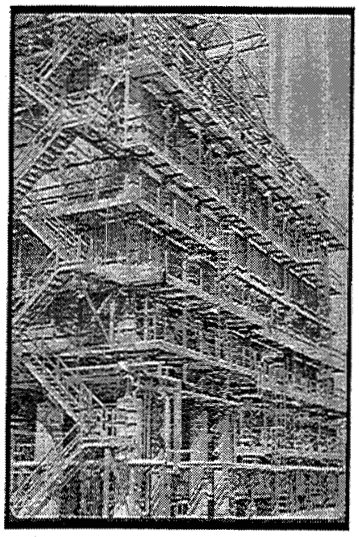
- 4 cells in one box
- Common Convection
- Smallest Plot



FH 2004/13-9

U-Tube Heater

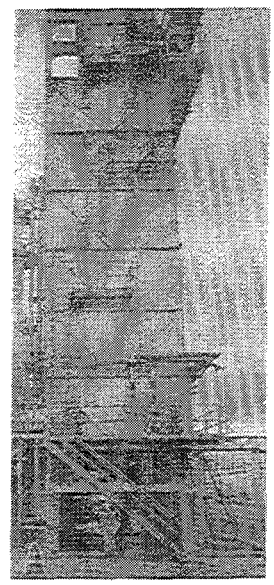
- Top Manifolds
- Staircase to Manifolds
- Platforms at Burners



FH 2004/13-10

Wicket or Inverted U-Tube Heater

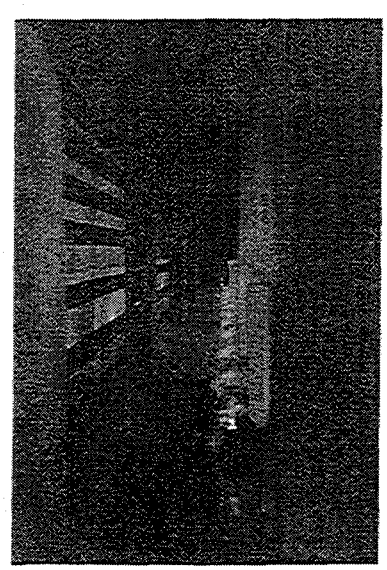
- Bottom Manifolds
- Free Draining Coil
- Used for Oil Fired Low when ΔP is required



FH 2004/13-11

Double Fired Horizontal Tube

- Used in Visbreaking, Coking and Hydrocrackers
- Short time at high temperature
- Free Draining Coil



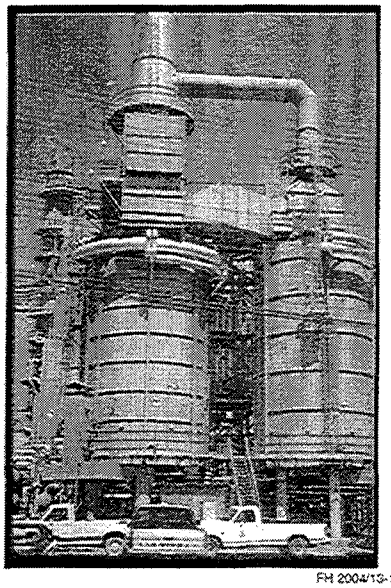
FH 2004/13-12

Training Services

New Heater Specification

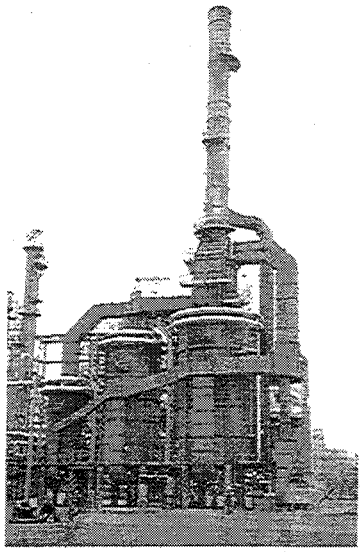
Heat Integration

- All radiant charge heater
- Flue gases ducted to reboiler heater
- 85%-87% Fuel Efficiency
- Blind in duct for regeneration
- Skin TI's located 1/4 to 1/3 of box height



Air Preheat System

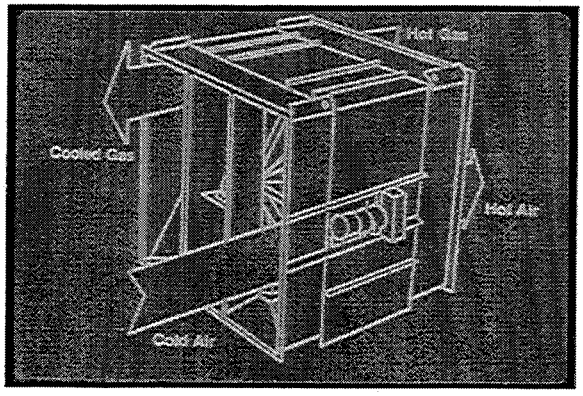
- Process Duty greater than 70 MM Btu/h (21 MW)
- Two year Payout
- Multiple Heaters on one system
- Forced Draft Burners require two combustion Fans



FH 2004/13-15

Regenerative Air Preheater

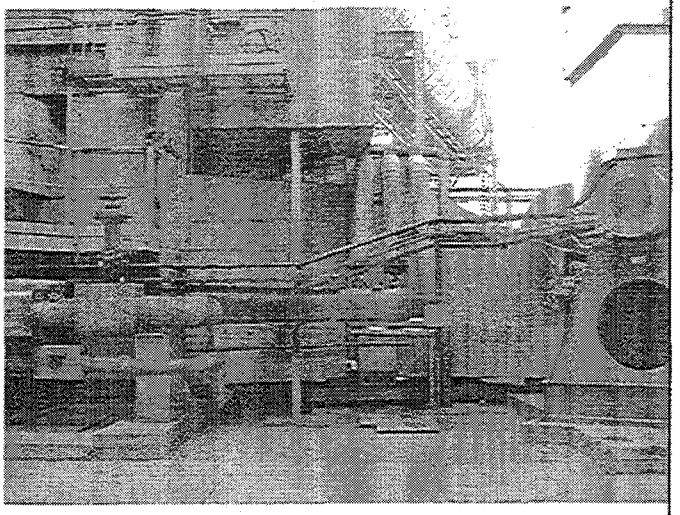
- Lungstrom Rotating Element
- 1970's design
- 10-25% Air Leakage
- Prone to fires
- Replace with Cast Iron Type



FH 2004/13-16

Regenerative Air Preheater

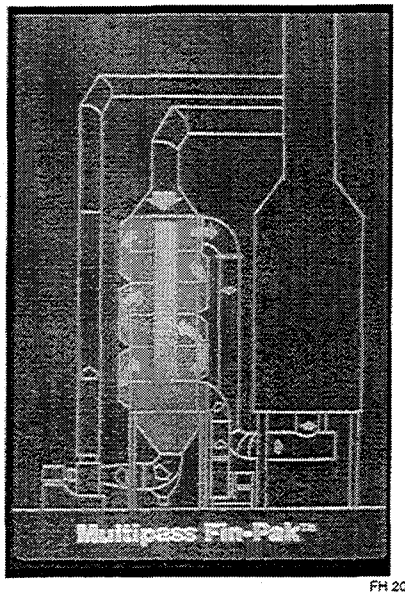
- Lungstrom Rotating Element
- Steam air heater to prevent corrosion



FH 2004/13-17

Recuperative Air Preheater

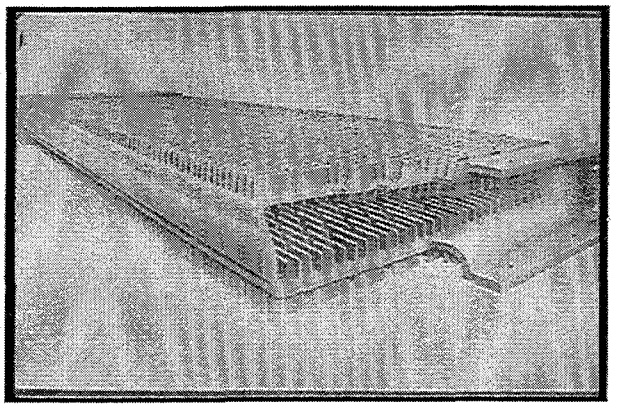
- By-Cast or Deka Cast Iron Elements
- Most Reliable Design
- Use 3 skin TI's at cold end
- Use steam-air heater upstream
- Do not use glass tubes



FH 2004/13-18

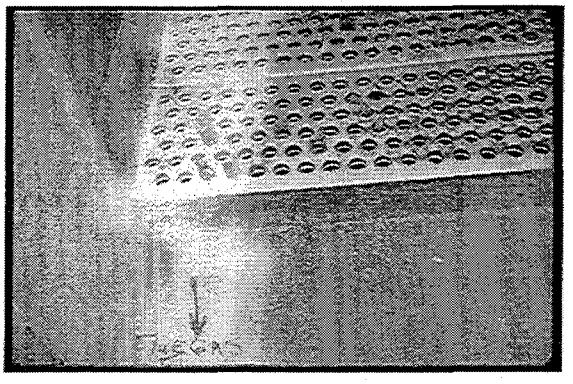
Recuperative Air Preheater

- By-Cast or Deka Cast Iron Elements
- Fin Height controls element Temperature
- Flue gas flows down center
- Air Flows across top fins



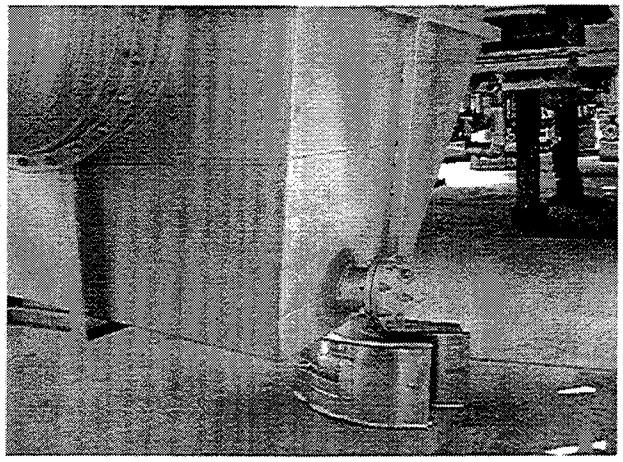
Recuperative Air Preheater

- Glass Tubes below Cast Iron Elements
- Used when steam-air heater is not used
- High breakage Rate
- Teflon Seal Fail
- Acid Condensation fouls the ID Fan
- UOP no longer allows this design



Air Preheater Wash Hopper

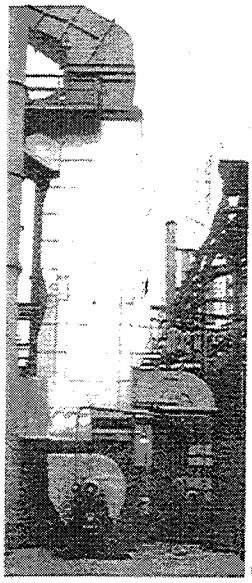
- Water wash when heat transfer drops or ΔP increases (4"clean)
- Check O2 upsteam and downstream of air preheater



FH 2004/13-21

Air Preheat System

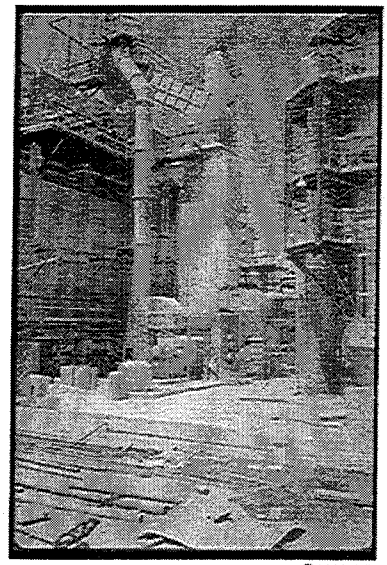
- Glass Tubes below Cast Iron Elements
- Used when steam-air heater is not used
- High breakage Rate
- Teflon Seal Fail (high temp or acid attack)
- Acid Condensation fouls the ID Fan
- UOP no longer allows this design



FH 2004/13-22

Air Preheat System

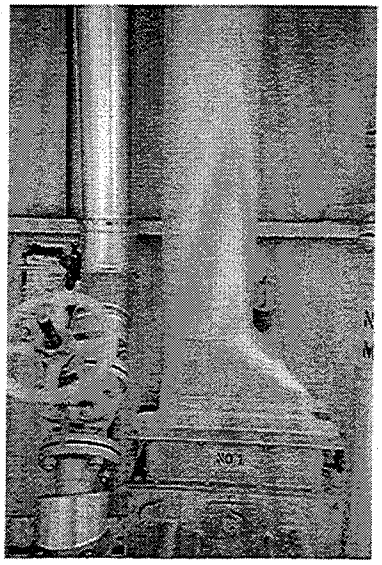
- Bad Air Preheat system
- Single FD fan with natural draft dropout doors
- Bad air distribution to natural draft burners
- No steam air heater
- Passing though acid dewpoint in air preheater results in high maintenance cost



FH 2004/13-23

Air Preheat System

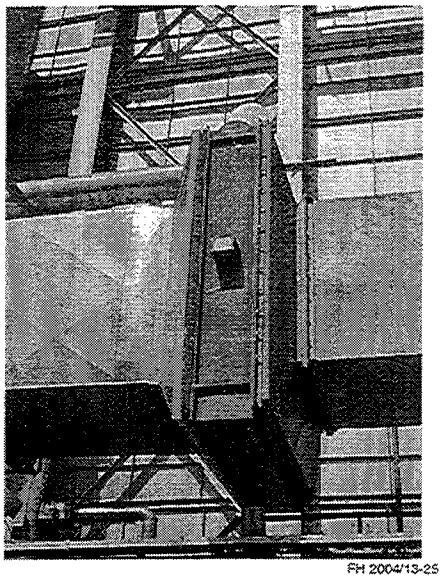
- Bad Air Supply Duct Transition
- Supply Duct should be designed for the lower of 25 fps or duct velocity head shall not exceed 10% of burner drop



FH 2004/13-24

Air Preheat System

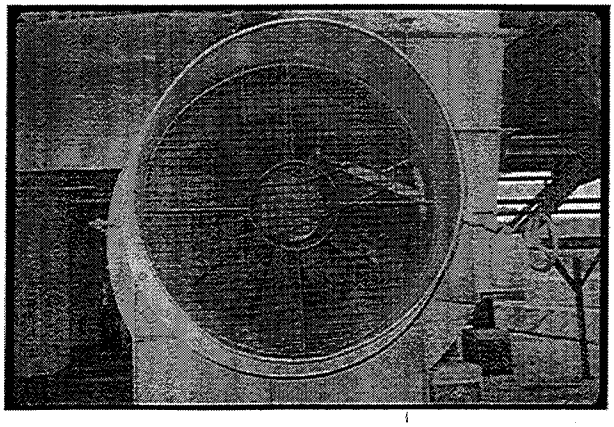
- Steam-Air Heater
- Located in front of flue gas air preheater to heat air to 140°F-170°F (60°C-76°C)
- Eliminates glass tubes
- Eliminates Acid Condensation in Air Preheater and ID Fan
- Standard UOP Design



FH 2004/13-25

Combustion Air Fans

- Two Required (one Spare)
- Variable -Inlet-Vane or Inlet Box Damper for Turndown
- Outlet Damper for switching Fans
- 115% Overage with 132% Head Margin



FH 2004/13-26

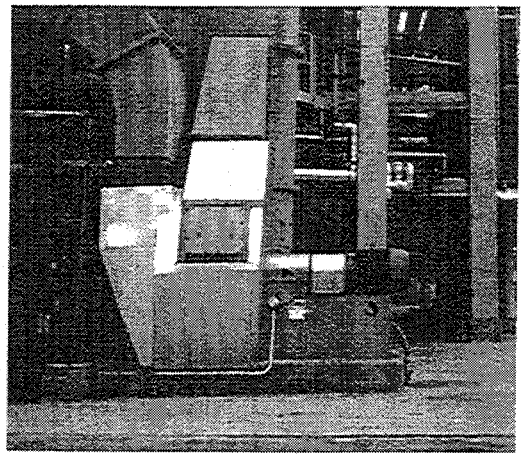
Combustion Air Fans

- When English is not the local language



Induced Draft Fan

- Radial Blade or Radial Tip Blade to prevent fouling
- One Required (stack acts as backup)
- Inlet Box Damper for Control
- 125% Overage with 156% Head Margin
- A 100°F (56°C) Temperature margin for air preheater fouling



New Heater Specifications

General

API 560 *Fired Heaters for General Refinery Services* gives mechanical specifications for purchasing a new heater. It is recommended that this document be supplemented with the following guidelines when purchasing a new heater.

Process

Use the Refinery Heater Summary Chart to pick flux rates, mass velocities, pressure drops, metallurgy and type of heater.

Efficiency

Most heaters should be designed for a 75-100°F cold end approach (process in - flue gas out). This results in 10-12 rows of convection section. See the charts in Section 4.

Consider air preheat when the process duty is greater than 70 MM Btu/h and the process inlet is above 500°F. The simple payout period should be less than 3 years.

Radiant Flux Rates

Do not exceed the flux rates shown in the Refinery Heater Specification Chart.
Very small heaters are designed for lower flux rates. Use the table below.

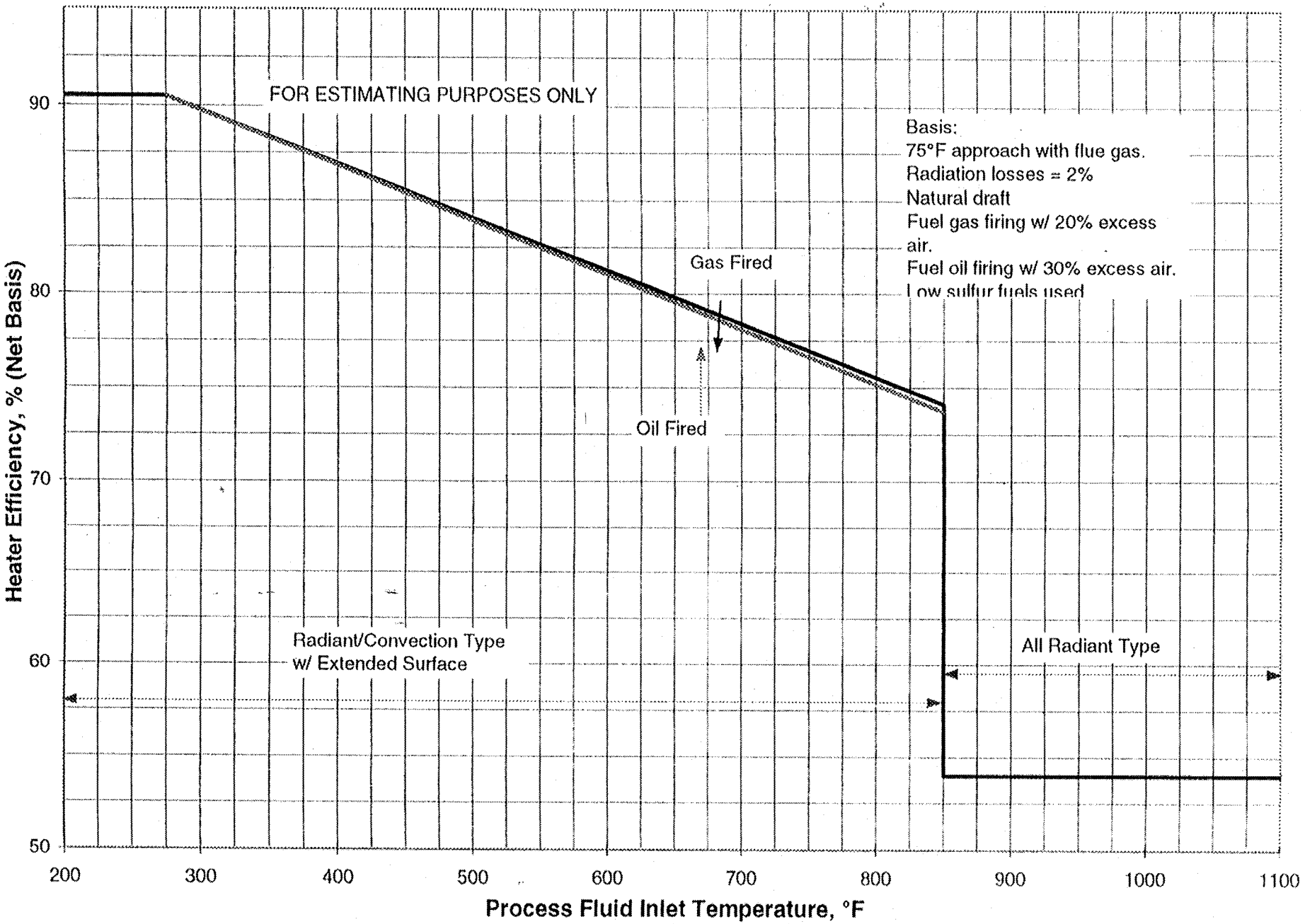
Radiant Duty, Btu/h	**Flux Rate, Btu/h-ft^2**
10	10,000
9	9,000
8	8,000
7	7,000
6	6,000
5	5,000
4 and lower	4,000

The shortest tube length allowed is 18-20 feet to minimize the high BWT problem with short tubes. See Section 3 for discussion.

Refinery Heater Specification

Process	Radiant Flux, Btu/h-ft^2	Mass Velocity, lb/sec-ft^2	Process ΔP,psi	Coil Metallurgy	Type of Heater
Crude	10,000	200 min	150	5 Cr or 9 Cr	Vertical Tube Box Cabin or V.C.
Vacuum	8,000	250 min	75	5 Cr or 9 Cr	Vertical Tube Box Cabin or V.C.
Visbreaking	7,000-8,000	350 min (5-6 fps cold oil)	250-300 clean	9 Cr	Cabin with brick centerwall
Coking	10,000	400 min (6 fps cold oil)	300 clean	9 Cr	Cabin with brick centerwall
Semi-Regn. Platformer	10,000	40-50	7	2-1/4 Cr	V.C. , cabin, wicket
CCR Platformer	10,000 SFU 15,000 DFU	23-35	3 per cell	9 Cr	Single Fired U-tube Double Fired U-tube
Reboiler	10,000	150-300	50	C.S.	V.C.
Hot Oil	10,000	400-600	65	C.S.	V.C.
Naphtha Hydrotreater	10,000	70-180	50	9 Cr	V.C.
Kerosene Hydrotreater	10,000	150-250	50	347H SS	V.C.
Diesel Hydrotreater	10,000	150-250	50	347H SS	V.C. or double fired vertical tube
Hydrocracker Gas Only	10,000 - 13,000	50-80	30-50	347H SS	double fired vertical tube
Hydrocracker Gas&Liquid	15,000	180-300	50-80	347H SS	double fired vertical tube

Heater Efficiency vs. Process Inlet Temperature



New Heater Specifications

Convection Specifications

Gas fired heaters should have at least 10 rows of convection tubes (3 bare and 7 finned).

Oil fired heaters should have 12 rows of convection tubes (3 bare and 9 finned). When firing heavy oils (heavier than a commercial No. 6 oil), use fin spacing of 2-1/2 fins per inch instead of the normal 3 fins per inch. The larger spacing between fins results in more effective sootblowing cleaning.

Studded tubes are used for oil firing only. Studded tubes clean better than fins, but they cost more. Studded surface adds approximate 6 % cost to the heater.

Studded tubes are more efficient than fins. 8 rows of studded tubes will pick up the same amount of heat as 9 rows of fin tubes. The flue gas pressure drop is higher with studded tubes than fin tubes. A taller stack is required with studded tubes.

For natural draft, use a flue gas mass velocity of 0.2-0.3 lb/sec-ft^2.

For induced draft systems, use a flue gas mass velocity of 0.3-0.4 lb/sec-ft^2.

Many heater vendors use high velocities which result in flue gas stack hydraulic limitations. (A polite way of saying the stack is too short). Remember the example of increasing the flue gas mass velocity from 0.3 to 0.44 lb/sec-ft^2. A 45% increase in mass velocity increases heat transfer by 25% at the expense of increasing pressure drop 210%.

Type of Heaters

Up to 170 MM Btu/h process duty, the vertical cylindrical heater is the most economical choice. From 170-340 MM Btu/h, a vertical tube box heater is approximately 5% less expensive than two vertical cylindrical heaters.

Hydrotreaters and Hydrocrackers with Sch 120 or Sch 160 tubes should be double fired to obtain even heating and minimize thermal stresses within the tubes.

New Heater Specifications

Refractory Specifications (General Service Heaters)

All heaters have castable refractory floors. UOP uses 8" (200 mm) thickness. Oil fired heaters have an additional layer of high duty firebrick 2-1/2" (65 mm) on top of the castable.

Ceramic Fiber Blanket

Gas fired heaters typically have ceramic fiber insulation for sidewalls and roof.

Shielded walls (covered by tubes) have 4" (100 mm) thickness. The backup layer is 2" (50mm) of 6lb/ft^3 density blanket. Provide two 1" top layers of 8 lb/ft^3 density blanket suitable for 2300°F. Joints shall be offset. The top layer shall be installed with overlap joints in the direction of gas flow. Studs and washers are 310 SS covered with 1" patch of ceramic fiber

Unshielded walls (not covered by tubes) have 5" (125 mm) minimum thickness. The backup layer is 3" (75mm) of 6 lb/ft^3 density blanket. Provide two 1" top layers of 8 lb/ft^3 density suitable for 2300°F Joints shall be offset. The top layer shall be installed with overlap joints in the direction of gas flow. Studs and washers are Inconel covered with 1" patch of ceramic fiber

Interior steel wall are coated with 1/8" (3 mm) High Temperature Stalastic Asphalt suitable for 500°F (260°C). Epoxy mastic is not an acceptable substitution. It does not adhere as well to the wall as the Stalastic. Several steel casing failures are known to have occurred due to acid attack after the epoxy mastic coating has flaked off.

When sulfur level in the fuel gas exceeds 500 ppm H2S or total sulfur, a 4 mil foil layer of 304 SS is inserted approximately 2" (50 mm) from the steel wall to prevent acid condensation. Joints are lapped and sealed with high temperature cement.

Ceramic Fiber Modules

Modules are better than layered blanket because the supporting hard ware is shielded from the heat source. Repairs are easier with modules. Modules have better velocity erosion resistance than blankets. Modules cost more than a blanket application. Modules are commonly use in high temperature (BWT above 1800°F or 982°C) applications such as ethylene cracking furnaces and steam methane reformers.

Castable Insulation

Oil fired heaters typically have castable insulation. It is much more resistant to sodium-vanadium ash attack than ceramic fiber. Castable insulation is often used with vertical cylindrical heaters with 347 SS tubes to provide protection during soda ash neutralization on the tubes.

API 560 requires that the temperature of the outside casing shall not exceed 180°F (82°C) at an ambient temperature of 80°F (27°C) in still air. UOP does not follow this practice. It was intended for ceramic fiber systems. To meet API requirements requires a dual layer castable with an ultralite backup layer. This backup layer has little strength. Field repairs will be difficult with dual layers construction.

UOP specifies 6" (150 mm) single layer on shielded walls.
UOP specifies 8" (200 mm) single layer on unshielded walls and roof

If dual layer construction is used, be sure to have independent anchoring systems. One set of Y-type anchors is not acceptable. This was missed in API 560.

If the total metals in the fuel oil exceeds 250 ppm, a dual layer castable is used. The hot face layer in a dense castable high in alumina. This is required to resist oil ash attack on the refractory.
Castable refractory is typically a 2200°F service limit, nominally a 60 lb/ft^3 density material. The typical castable used is 1:2:4 LHV (Lumnite -Haydite- Vermiculite).
UOP specifies castable insulation in convection sections, breechings, and stacks. Flue gas ducts to and from air preheat systems are castable insulation.

Air Preheater Specification

UOP specifies the cast iron type heat exchanger (Deka or By-Cast). This type of exchanger has thicker elements than the plate type exchangers. The exchanger can withstand some corrosion without perforating. The exchanger elements should be designed for a 25°F above acid dewpoint. Specify 3 skin thermocouples at the cold end junction to measure element temperature. The air preheater should be insulated. A typical specification is included below:

> Provide a combustion air preheater of the recuperative (stationary) type. Cast iron type air preheater shall be supplied. Plate type air preheater is not acceptable. Preheater shall be capable of handling flue gases at an inlet temperature not less than 56°C above the maximum anticipated inlet temperature when the heater is operating at design case duty. Air preheater shall be insulated.
>
> The air preheat system shall be designed in a manner to prevent acid condensation. The calculated acid dewpoint is XXX°C. The preferred minimum air preheater metal element temperature is 14°C above the acid dewpoint resulting in a minimum metal temperature of XXX°C. A steam/air preheater shall heat the combustion air from 15°C to XXX°C. Flue gas exiting the air preheater shall be 42°C above the acid dewpoint resulting in a flue gas to stack temperature of XXX°C.
>
> Approximate air preheat system conditions are as follows. Flue gas from Item No. XXX enters the combustion air preheater at XXX°C and exits the combustion air preheater at XXX°C. Combustion air from the steam/air preheater enters the combustion air preheater at XXX°C and exits the combustion air preheater at XXX°C.
>
> Provide a minimum of three skin thermocouples on the flue gas side outlet near the air inlet to measure metal surface temperature. Each thermocouple assembly shall consist of a mineral insulated Type K grounded junction thermocouple element in a 316 stainless steel sheath. Thermocouple attachment to the metal tubes shall be per manufacturer's preferred method. Each sheathed thermocouple shall exit the air preheater per vendor design practice, and shall be connected to a standard terminal head assembly.
>
> A water wash system shall be provided to clean the air preheater. The wash system shall include a pipe grid with spray nozzles at the air preheat flue gas inlet. A water collection hopper (adequately sized) with low point drain shall be installed on the air preheater flue gas outlet.
>
> Provide means for positive isolation of the air preheater. Provide access to elements for inspection and cleaning.

New Heater Specifications

Fan Specifications

It is important to have overage on the fans. Most hydraulic circuits in the plant are designed for 10-20% overage with the control valve 70% open. Most vendors do fan hydraulics with the dampers fully open at design (poor control). A typical specification is provided below:

COMBUSTION AIR FAN

Provide two direct coupled, motor driven air fans (one operating and one spare), each sized to operate at 115% of design flow conditions at 132% of the design pressure differential when the heater is being fired at xx% excess air. Contractor is to determine the fan pressure differential necessary to overcome the system drop (burners, venturi, duct work, air preheater drop, etc.) at the rated air rate.

Each fan shall be supplied with an automatically controlled inlet box damper. Inlet guide vanes or radial blade dampers are acceptable substitutions. Provide an isolation blind in the discharge of each fan.

Supply one common vertical suction pipe at least 5 meters above ground level. The inlet shall be adequately protected against ingress of rain and solid matter. Fans shall have noise attenuation to meet 85 dB(A) at a distance of one meter from the fan.

Fans shall be designed per API 560

The inlet box damper provides turndown.

New Heater Specifications

INDUCED DRAFT FAN

Provide one motor driven induced draft fan sized to operate at 56°C above the calculated air preheater outlet temperature at 125% of design flow conditions at 156% of the design pressure differential when the heater is being fired at xx% excess air with the air preheater on-line. Contractor to determine the fan pressure differential necessary to overcome the system drop (air preheater, dampers, duct work, restrictions, etc.) at the rated flue gas rate. Induced draft fan shall be mechanically designed for operation at 260°C. Provide an automatically controlled inlet box damper with parallel multi-blade design on the suction side of the fan.

Induced draft fan wheel type shall be radial-tipped or modified radial self-cleaning design. Induced draft fan shall have AMCA Standard 2404 arrangement 3 or 7 configuration with inboard/outboard type bearings. Arrangement 7 is preferred. Fan shall have noise attenuation to meet 85 dB(A) at a distance of one meter from the fan.

The fan shall be designed per API 560.

COMMENTS:

1. The ID Fan is designed for 100°F/56°C above calculated preheater outlet temperature to allow for fouling in the preheater. When the preheater fouls and the outlet temperature increases, the flue gas SCFM increases while at the same time the heater firing rate increases (due to lower combustion air temperature to burners). This effect causes the unit to become fan limited.

2. The ID Fan is mechanically designed for 500°F/260°C because the flue gas entering the ID Fan should be below that temperature (75°F above the dew point).

3. Arrangement 7 is preferred for easier field installation. The difference between 3 and 7 is that 7 is fully assembled while 3 requires some field assembly. Other than that, the two arrangements are essentially the same.

4. Radial tip blades have fouling tendency and lower efficiency than airfoil blades.

New Heater Specifications

Burner to Tube Clearance

UOP prefers large burner to tube clearances and smaller heat release per burner than most vendors. This helps prevent flame impingement and hot spots. Tables are listed in Section 8.

Upgraded Burner Specifications

The burners are one of the most important parts of the heater, yet they only account for 5-10% of the heater cost. Upgrading the burner is the best investment a refiner can make. The upgrades will only add 1-2% to the cost of the heater. The upgrades will result in lower excess air levels, lower maintenance and better flame quality. A typical specification is included below:

Low NOx Gas Only Burner (John Zink PSFG-M) - Natural Draft

The heater shall be furnished with low NO_x, staged fuel, natural draft type gas burners with noise attenuation and individual plenum box design with damper control of air (John Zink PSFG-M or equivalent). Rotary air control registers are not acceptable. At burner design conditions, a minimum of 85% of the available draft (air damper fully open) shall be utilized across the burner. In addition, 75% of the air side pressure drop (air damper fully open) shall be utilized across the burner throat. Damper blades shall have seal strip construction to obtain an approximate leakage rate of 3% in the fully closed position when subject to design draft conditions. Burners shall be sized for 120% heat release with the heater operating at design case duty. Burners shall be designed for approximately 20 psig (239 kPa (a)) fuel pressure at burner with 15% excess air firing fuel gas. A minimum tip drilling of 2.3 mm is preferred.

Each burner shall be supplied with a gas fueled, inspirating type pilot burner having a flame retention head (John Zink ST-1S or equivalent). Inspirator shall be located outside of burner plenum. Flame retention head shall be 25Cr-20Ni metallurgy. Pilot shall be designed for natural gas.

Gas tips and stabilizer cones shall be either 25Cr-20Ni or 19Cr-9Ni metallurgy.

Burner blocks (burner tiles) based on silica and/or calcium bonders shall be pre-dried to 260°C. Chemically bonded tiles do not require drying. Burner blocks shall be furnished with a tile alignment device to insure proper tile installation. Vendor shall provide contractor with procedures and tolerances for burner and burner block installation. Casing of all burner equipment shall be a minimum of 10 gauge steel. Provide firm seal, pivoting slide plate type sighting and lighting ports on each burner. Provide a minimum of one sighting and one lighting port for the pilot, and one sighting port for the main gas tip.

COMMENTS:

1. The minimum preferred tip drilling of 3/32" is a No. 42 drill bit. Smaller tip drillings are prone to plugging problems.

New Heater Specifications

Upgraded Burner Specifications-continued

The 85% utilization of draft ensures that the burner tiles are sized properly, allowing low excess air operation.

The damper gives good air control. Rotary registers give bad control.

The seal strips prevent air leakage when the burner is out of service.

The tile alignment device results in proper installation, better performance and lower maintenance cost

If the burner tiles are green refractory (not dried or chemically bonded), they will tend to crack and break up at initial firing

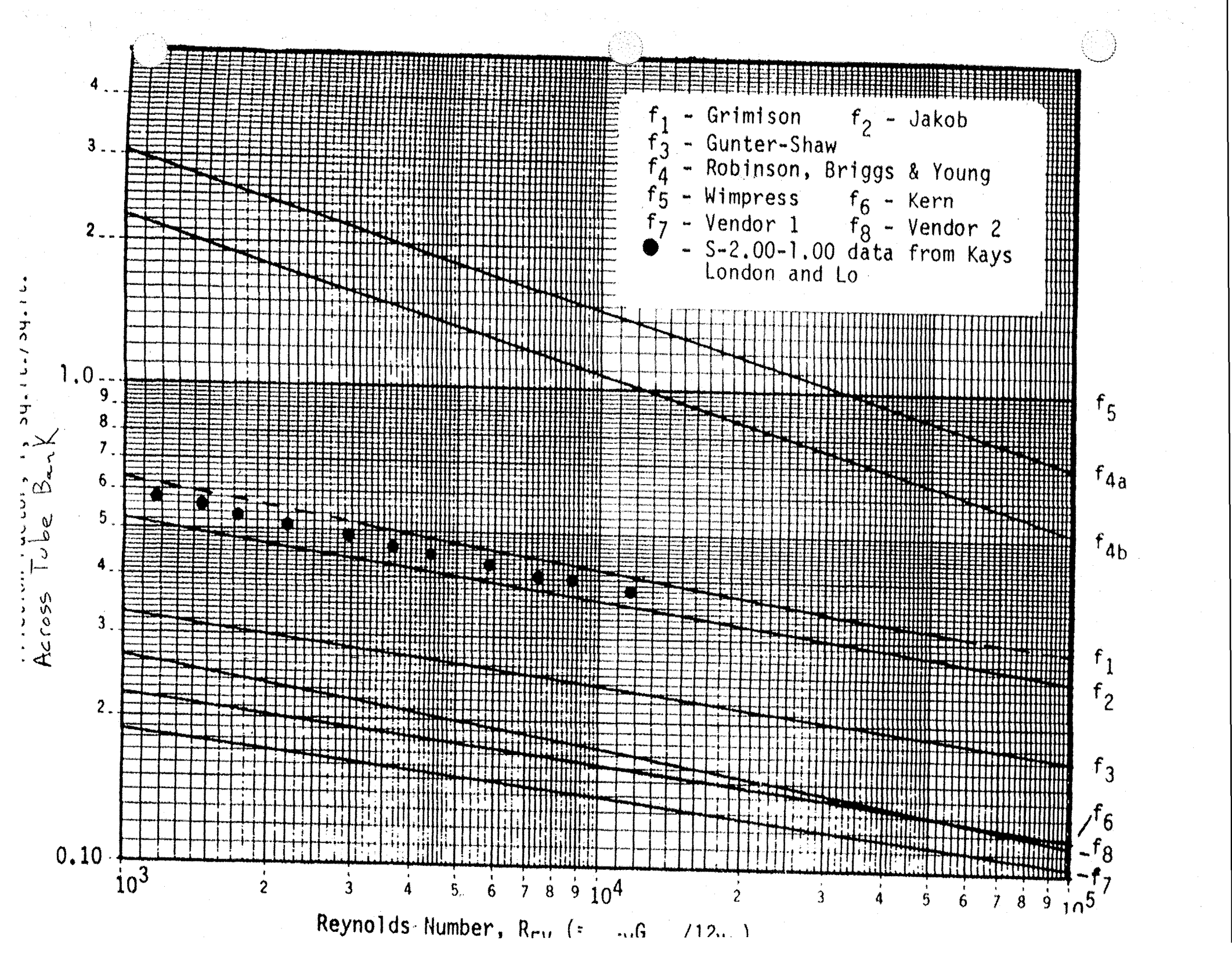
The ST-1S is an upgraded high stability pilot, also suitable for forced draft applications.

Stack Specification

UOP specifies the stack design so that a negative pressure of at least 0.10 inches of water (3 mm) column is maintained at the top of the radiant section at 130% of normal heat release with design excess air and maximum ambient temperature. API 560 *Fired Heaters for General Refinery Services* only requires 120%.

There are many ways to calculate pressure drop across the convection section. Most heater vendors use a method which gives a very low pressure drop which results in a short stack.

Refractory lining is either 1-4 LH (lumnite-haydite) or 1-2-4 LHV (lumnite-haydite-vermiculite). 1-4 LH has more strength and is less prone to cracks. 1-2-4 LHV is a better insulator.


f_1 - Grimison
f_2 - Jakob
f_3 - Gunter-Shaw
f_4 - Robinson, Briggs & Young
f_5 - Wimpress
f_6 - Kern
f_7 - Vendor 1
f_8 - Vendor 2
● - S-2.00-1.00 data from Kays London and Lo
Across Tube Bank K
Reynolds Number, R_{ev} (= ..G /12..)
4
3
2
1.0
9
8
7
6
5
4
3
2
0.10
10^3
2
3
4
5
6
7
8
9
10^4
2
3
4
5
6
7
8
9
10^5
f_5
f_{4a}
f_{4b}
f_1
f_2
f_3
f_6
f_8
f_7

Training Services

Published Heater Articles

Published Heater Articles

- Herb Berman - *Fired Heater*
- R. Norman Wimpress - *Fired Heater Performance*

FIRED HEATERS—1

Finding the basic design for your application

Herbert L. Berman, Caltex Petroleum Corp.

☐ Energy conservation is not new. The economic success of any competitive process requires the efficient use of energy. Of the energy consumed in typical chemical processing or petroleum refining plants, approximately 75% is burned in the form of hydrocarbon fuel in fired heaters and steam boilers. Conservation, therefore, affords a strong and timely incentive for scrutinizing the design criteria and construction features commonly used in fired heaters.

What is a fired heater?

A fired heater, for our purposes, will include a number of devices in which heat liberated by the combustion of fuel within an internally insulated enclosure is transferred to fluid contained in tubular coils. Typically, the tubular heating elements are installed along the walls and roof of the combustion chamber, where heat transfer occurs primarily by radiation, and if economically justifiable, in a separate tube bank, where heat transfer is accomplished mainly by convection.

Industry identifies these heaters with such names as process heater, furnace, process furnace and direct-fired heater, all of which are interchangeable.

The fundamental function of a fired heater is to supply a specified quantity of heat at elevated temperature levels to the fluid being heated. It must be able to do so without localized overheating of the fluid or of the structural components.

Fired-heater size is defined in terms of its design heat-absorption capability, or duty. Duties range from about a half-million Btu/h for small, specialty units to about one billion Btu/h for superproject facilities such as the mammoth steam hydrocarbon-reformer heaters. By and large, the vast majority of fired-heater installations fall within the 10- to 350-million-Btu/h range.

Process industry requirements for fired heaters are divided, in the main, into a half-dozen general service categories. These categories can be designated and described as follows:

Column reboilers. This is normally considered one of the mildest and least critical of fired-heater applications. The charge stock taken from a distillation column is a recirculating liquid that is partially vaporized in the fired heater. The mixed vapor-liquid stream reenters the column, where the vapor condenses and releases the heat of vaporization. Reboiler applications are characterized by relatively small differentials between the inlet and outlet fluid temperatures across the fired heater, and by substantial vaporization (typically, 50% or more of the charge stock is vaporized). Depending on the particular application, reboiler heater outlet temperatures generally fall in the range of 400°F to 550°F.

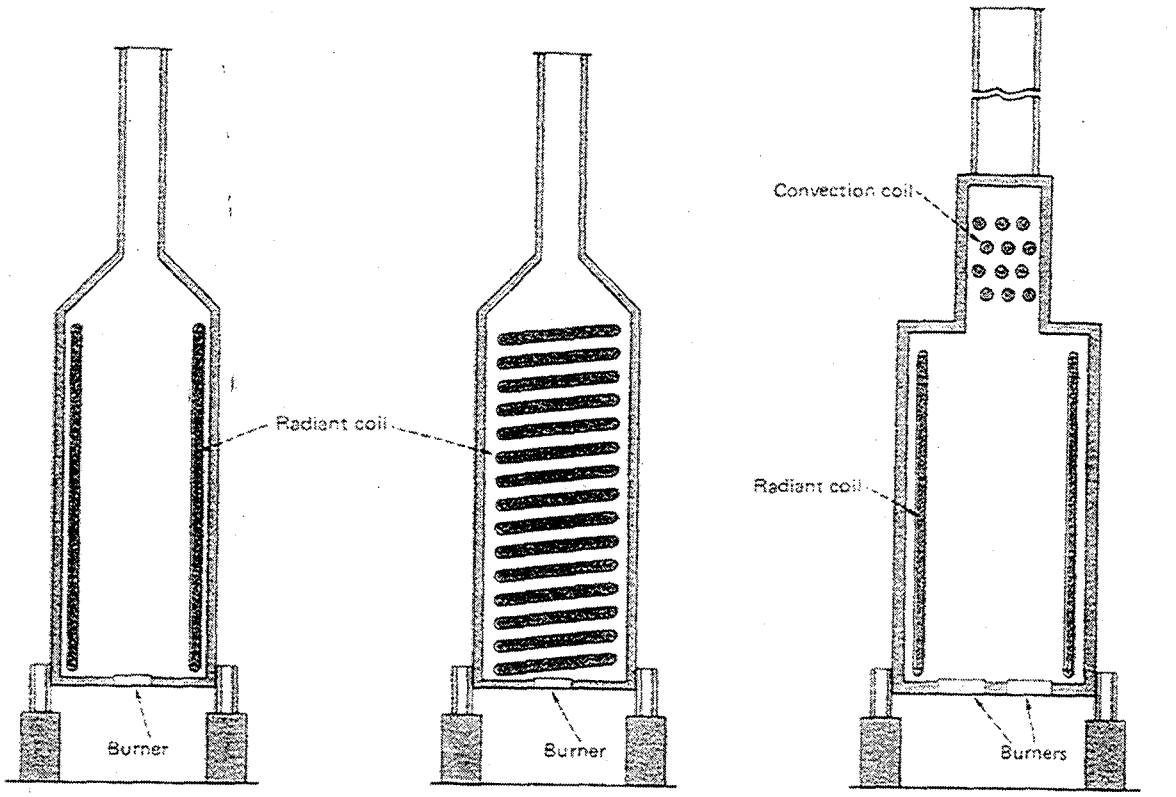
Fractionating-column feed preheaters. Fired heaters in this service tend to be the workhorses of many process operations. The charge stock (usually all liquid, although some feeds may contain a nominal amount of vapor at the inlet) is sent to the fired heater following upstream preheating in unfired equipment. In the fired heater, the fluid temperature is usually raised high enough to achieve partial vaporization of the charge stock.

A typical example of this service is the feed heater for an atmospheric distillation column in the crude-oil unit of a petroleum refinery. Here, crude oil entering the fired heater as a 450°F liquid might exit near 700°F with about 60% of the charge stock vaporized.

Reactor-feed preheaters. Fired heaters in this application raise the charge-stock temperature to a level necessary for controlling a chemical reaction taking place in an adjoining reactor vessel. The nature of the charge stock and the heater operating temperatures and pressures can vary considerably, depending on the process. The following examples illustrate the diversity of the applications performed by reactor-feed preheaters:

- Single-phase/single-component heating such as steam superheating in the reaction sections of styrene manufacturing processes. In this service, the fluid temperature across the fired heater increases from an inlet temperature of about 700°F to an exit temperature of approximately 1,500°F.
- Single-phase/multicomponent heating, such as the heating of mixtures of vaporized hydrocarbons and recycle hydrogen gas prior to catalytic reforming in a refinery. In this service, the charge stock enters the fired heater at about 800°F and exits at approximately 1,000°F.

In reformers, the fluid pressure may range from about 250 to 600 psig. Severe restrictions on fluid pressure drop are normally associated with this service.



a. Vertical-cylindrical, all radiant

b. Vertical-cylindrical, helical coil

c. Vertical-cylindrical with crossflow convection section

Vertical-tube fired heaters can be identified by the vertical arrangement of radiant-section coil. Fig. 1

■ Mixed-phase/multicomponent heating, such as the heating of mixtures of liquid hydrocarbons and recycle hydrogen gas for reaction in a refinery hydrocracker. Fluid temperatures typically run from 700°F at the inlet to 850°F at the outlet. Operating pressures may reach 3000 psig, depending on the process.

Heat supplied to heat-transfer media. Many plants furnish heat to individual users via an intermediate heat-transfer medium. A fired heater is generally employed to elevate the temperature of the recirculating medium, which is typically a heating oil, Dowtherm, Therminol, molten salt, etc. Fluids flowing through the fired heater in these systems almost always remain in the liquid phase from inlet to outlet.

Heat supplied to viscous fluids. Oftentimes heavy oil must be pumped from one location to another for processing. At low temperatures, where the oil may have so high a viscosity as to render pumping infeasible, a fired heater is employed to warm the oil to a temperature that will facilitate pumping.

Fired reactors. In this category are heaters in which a chemical reaction occurs within the tube coil. As a class, these units represent the fired-heater industry's most sophisticated technology. The following two applications typify the majority of installations:

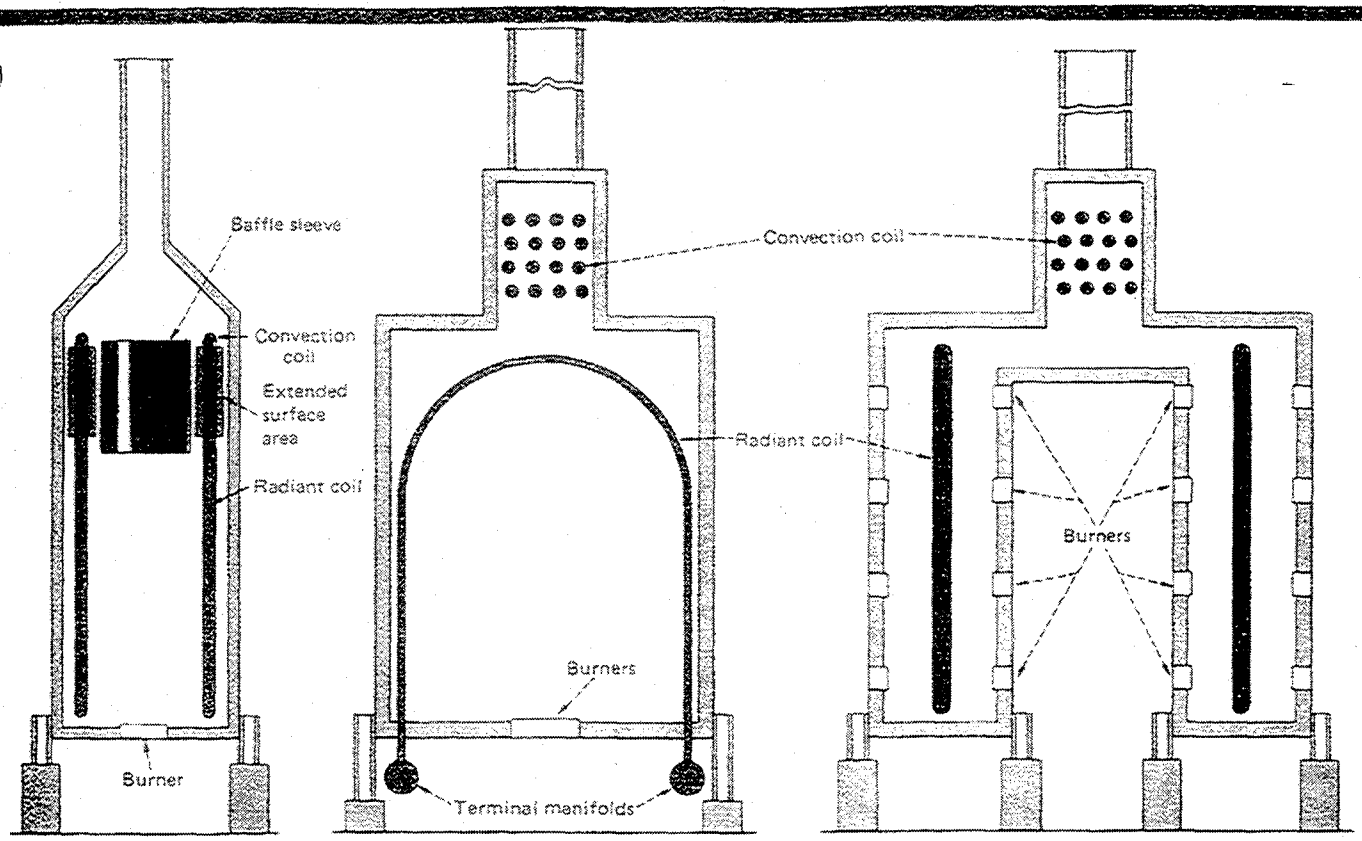
■ Steam hydrocarbon-reformer heaters, in which the tubes of the combustion chamber function individually as vertical reaction vessels filled with nickel-bearing catalyst. In reformers that yield hydrogen, fluid outlet temperatures range from 1,450 to 1,650°F.

■ Pyrolysis heaters, used to produce olefins from gaseous feedstocks such as ethane and propane and from liquid feedstocks such as naphtha and gasoil. In cracking heaters, where chemical reactions occur in the coil, the tubes and burners are arranged so as to assure pinpoint firing control. Fluid outlet temperatures in heaters designed for liquid feedstocks are in the 1,500°F to 1,650°F range.

Many variations in design and layout

There are many variations in the layout, design, and detailed construction of fired heaters. A consequence of this flexibility is that virtually every fired heater is custom-engineered for its particular application.

The simplest type of fired heater follows the so-called "all radiant" design, wherein the entire tube coil is arranged along the walls of the combustion chamber or radiant section. This design is characterized by low thermal efficiency and normally represents the lowest capital investment for a specified duty. The terminology "all radiant" is somewhat of a misnomer. Convection currents do exist, due to the flow of flue gases through the combustion chamber, and these currents



d. Vertical-cylindrical with integral convection section

e. Arbor or wicket type

f. Vertical-tube, single-row, double-fired

(continued) Fig. 1

account for a portion of the total heat absorbed in the radiant section.

In addition to the radiant section, most modern fired heaters include a separate convection section. The residual heat of the flue gases leaving the radiant section is recovered in this section, primarily by convection. Using this heat for preheating the charge stock, or for other supplementary heating services, increases the thermal efficiency of the fired heater.

The first few rows of tubes in the convection section are subject to radiant heat transfer, in addition to convective transfer from the hot flue gases as they flow across the tubes. Because these tube rows are usually being subjected to the highest heat-transfer rates in the fired heater, they are aptly termed "shield" or "shock" tubes.

Horizontal vs. vertical

The principal classification of fired heaters, however, relates to the orientation of the heating coil in the radiant section; i.e., whether the tubes are vertical or horizontal. Vertical arrangements are shown in Fig. 1; horizontal arrangements in Fig. 2. Salient features of each configuration are noted here:

Vertical-cylindrical, all radiant. Here the tube coil is placed vertically along the walls of the combustion chamber. Firing is also vertical, from the floor of the heater.

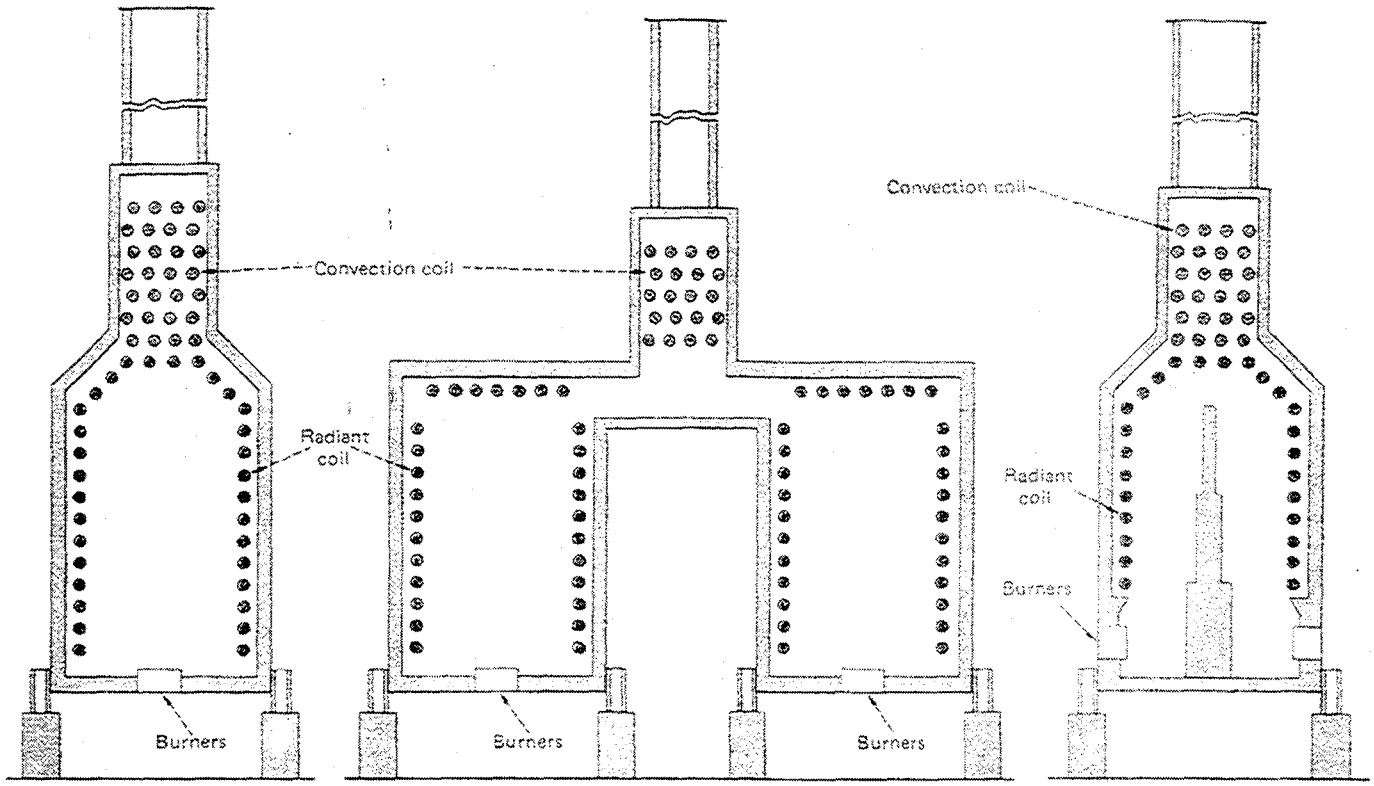
Heaters of this type represent a low-cost, low-efficiency design, which requires a minimum of plot area. Typical duties are 0.5 to 20 million Btu/h.

Vertical-cylindrical, helical coil. In these units, the coil is arranged helically along the walls of the combustion chamber, and firing is vertical from the floor. Although these heaters are grouped with others having vertical tube designs, their in-tube characteristics resemble those of horizontal-tube fired heaters.

This design also represents low cost, low efficiency, and requires a minimum of plot area. The tube coil is inherently drainable. One limitation on these units is that generally only one flow path is followed by the process fluid. Heating duties run from 0.5 to 20 million Btu/h.

Vertical-cylindrical, with crossflow convection. These heaters, also fired vertically from the floor, feature both radiant and convection sections. The radiant-section tube coil is disposed in a vertical arrangement along the walls of the combustion chamber. The convection section tube coil is arranged as a horizontal bank of tubes positioned above the combustion chamber.

This configuration provides an economical, high-efficiency design that requires a minimum of plot area. The majority of new, vertical-tube fired-heater installa-



a. Cabin b. Two-cell box c. Cabin with dividing bridgewall

Six basic designs used in horizontal-tube fired heaters. Radiant-section coil is horizontal Fig. 2

tions fall into this category. Typical duty range is 10 to 200 million Btu/h.

Vertical-cylindrical, with integral convection. Although this design is rarely chosen for new installations, the vast number of existing units of this type warrants its mention in any review of fired heaters.

As with the previous types, this design is likewise vertically fired from the floor, with its tube coil installed in a vertical arrangement along the walls. The distinguishing feature of this type is the use of added surface area on the upper reaches of each tube to promote convection heating. This surface area extends into the annular space formed between the convection coil and a central baffle sleeve. Medium efficiency can be achieved with a minimum of plot area. Typical duty for this design is 10 to 100 million Btu/h.

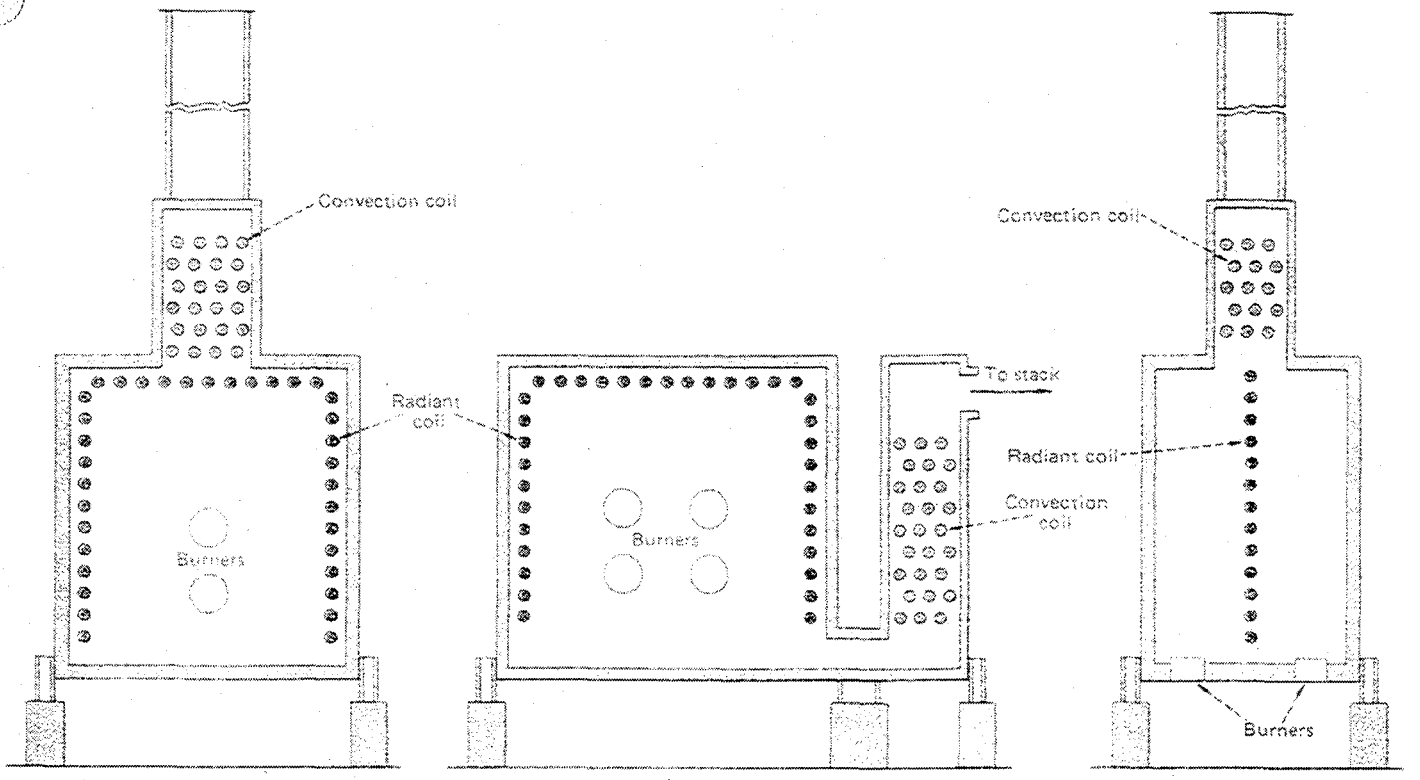
Arbor or wicket. This is a specialty design in which the radiant heating surface is provided by U-tubes connecting the inlet and outlet terminal manifolds. This type is especially suited for heating large flows of gas under conditions of low pressure drop. Typical applications are found in petroleum refining, where this design is often employed in the catalytic-reformer charge heater, and in various reheat services. Firing modes are usually vertical from the floor, or horizontal between the riser portions of the U-tubes.

This design type can be expanded to accommodate several arbor coils within one structure. Each coil can be separated by dividing walls so that individual firing control can be attained. In addition, a crossflow convection section is normally installed to provide supplementary heating capacity for chores such as steam generation. Typical duties for each arbor coil of this design are about 50 to 100 million Btu/h.

Vertical-tube, double fired. In these units, vertical radiant tubes are arranged in a single row in each combustion cell (there are often two cells) and are fired from both sides of the row. Such an arrangement yields a highly uniform distribution of heat-transfer rates (heat flux) about the tube circumference.

Another variation of these heaters uses multilevel side-wall firing, which gives maximum control of the heat-flux profile along the length of the tubes. Multilevel side-wall firing units are often employed in fired-reactor services and in critical reactor-feed heating services. In addition to the twin-cell furnaces already mentioned, single-cell models are available for smaller duties. As a group, these represent the most-expensive fired heater configuration. The typical duty range for each cell runs from about 20 to 125 million Btu/h.

Horizontal tube cabin. The radiant-section tube coils of these heaters are arranged horizontally so as to line the sidewalls of the combustion chamber and the sloping roof or "hip." The convection-section tube coil is posi-



d. End-fired box

e. End-fired box with side-mounted convection section

f. Horizontal-tube, single-row, double-fired

(continued) Fig. 2

tioned as a horizontal bank of tubes above the combustion chamber. Normally the tubes are fired vertically from the floor, but they can also be horizontally fired by sidewall-mounted burners located below the tube coil. This economical, high-efficiency design currently represents the majority of new, horizontal-tube, fired-heater installations. Duties run 10 to 100 million Btu/h.

Two-cell horizontal tube box. Here the radiant-section tube coil is deployed in a horizontal arrangement along the sidewalls and roof of the two combustion chambers. The convection-section tube coil is arranged as a horizontal bank of tubes positioned between the combustion chambers. Vertically fired from the floor, this is again an economical, high-efficiency design. Typical duties range from 100 to 250 million Btu/h. For increased capacity, the basic concept can be expanded to include three or four radiant chambers.

Horizontal tube-cabin with dividing bridgewall. Again the radiant-section tube coil is arranged horizontally along the sidewalls of the combustion chamber, and along the hip. The convection-section tube coil takes the form of a horizontal bank of tubes positioned above the combustion chamber. A dividing bridgewall between the cells allows for individual firing control over each cell in the combustion chamber. Available options permit horizontal firing with sidewall-mounted burners (as shown), or vertical firing from the floor along both sides of the bridgewall. A typical duty range for this design is 20 to 100 million Btu/h.

End-fired horizontal tube box. The radiant-section tube coil is disposed in a horizontal arrangement along the sidewalls and roof of the combustion chamber. The convection-section tube coil is arranged as a horizontal bank of tubes positioned above the combustion chamber. These furnaces are horizontally fired by burners mounted in the end walls. Typical duty range for this design is 5 to 50 million Btu/h.

End-fired horizontal tube box, with side-mounted convection section. Here the radiant section tube coil is disposed in a horizontal arrangement along the sidewalls and roof of the combustion chamber. The convection-section coil is arranged as a horizontal bank of tubes positioned alongside the chamber. The unit is horizontally fired from burners mounted on the end wall.

These furnaces are found in many older installations, and occasionally in new facilities that burn particularly poor grades of fuel oil containing a high ash concentration. This relatively expensive design provides duties ranging from 50 to 200 million Btu/h.

Horizontal-tube, double-fired. Horizontal radiant tubes are arranged in a single row and are fired from both sides to achieve a uniform distribution of heat-transfer rates around the tube circumference. Such heaters are normally fired vertically from the floor. They are often

Evolution of the modern fired heater

The development of the design concepts used in the modern fired heater was spurred by numerous adverse operating experiences.

The pot still of the late nineteenth century was the first example of industrial direct-fired heating. A major drawback with the pot still was its unsuitability for heating viscous or flammable liquids. With such fluids, intense localized overheating, product degradation, and coke formation were frequently followed by disastrous failures and fires.

The horizontal shell still was the next development in direct-fired heating. The chief advantages of the shell still over the fired pot were improved equipment reliability and safety. However, heat transfer rates and thermal efficiency remained poor.

At the beginning of the twentieth century, industry moved to direct-fired heating of fluids contained in tubular elements. The occurrence of tube ruptures and fires remained commonplace, largely due to a lack of engineering knowhow in preventing the formation of undesirable coke and salt deposits, which resulted from overheating of the tubes.

The need for more-even heating led to the concept of separating the heat-absorbing tubes from the combustion zone by means of a partial wall, or "bridgewall." Consequently, these early tubular fired heaters were primarily convection types.

Despite this more-conservative approach, which removed the tubes from the flame zone, overheating, coke deposition, and tube failures persisted. Because combustion was carried out with about 40% excess air, flue-gas temperatures reached 3000°F as the gases left the combustion zone and entered the heat-absorbing section. Intense convection at this temperature level, coupled with a substantial radiative component, resulted in heat transfer to the tubes facing the combustion zone at better than 100,000 Btu/(h)(ft^2).

To reduce the heat-transfer rate in the combustion zone to tolerable levels, vast amounts of excess air (several hundred percent) were required. Not only was thermal efficiency low at these high excess-air rates, but also the quenching effect of large volumes of cold air resulted in poor combustion characteristics.

As the technology of radiant heat transfer has improved, fired-heater designs have reverted more and more to the inclusion of radiant-heat-absorbing tubes, resulting in the modern, primarily radiant, fired heater of today.

selected for critical reactor-feed heating services. For increased capacity, the concept can be expanded to provide for a dual combustion chamber. A typical duty range for each cell of this design is about 20 to 50 million Btu/h.

Air supply and flue-gas removal

Aside from the major classifications according to service and configuration, fired heaters can also be grouped according to their methods of combustion-air supply and flue-gas removal.

The capability for inducing the flow of combustion air into a fired heater exists when hot flue gas of relatively low density is confined in a structure and isolated from higher-density air at ambient temperature. The buoyancy of the hot flue gas contained in the fired heater creates "draft" (less than atmospheric pressure), which induces flow of air into the combustion chamber. Since this draft results from a natural stack effect, it is termed natural draft. The majority of fired-heater installations are such natural-draft types, where a stack effect introduces the combustion air and removes the flue gas.

Obstruction to the flow of flue gas through a fired heater can result in a condition of pressure greater than atmospheric (positive pressure) in the structure. It is the function of the stack in a natural-draft heater to generate draft sufficient to overcome such obstructions and to maintain a negative pressure throughout.

An induced-draft fired heater incorporates an induced-draft fan, in lieu of a stack, to maintain a negative pressure and to induce the flow of combustion air and the removal of flue gas.

A forced-draft fired heater is one wherein the combustion air is supplied under positive pressure by means of a forced-draft fan. It is to be noted that even with air supplied under positive pressure, the combustion chamber and all other parts of the fired heater are maintained under negative pressure, and the flue gas is removed by the stack effect.

A forced-draft/induced-draft heater uses a forced-draft fan to supply combustion air under positive pressure; an induced-draft fan maintains the combustion chamber and all other parts of the fired heater under negative pressure and removes the flue gas.

Most fired heaters equipped with air preheaters are of the forced-draft/induced-draft type.



The author

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FIRED HEATERS—II

Construction materials, mechanical features, performance monitoring

This second of four parts looks closely at some of the design options and mechanical features that are available in today's fired heaters. Items covered include refractories, tube coils, extended-surface devices, tube supports and burners.

*Herbert L. Berman, Caltex Petroleum Corp.**

☐ A variety of process, structural and environmental factors influence the choice of materials and mechanical design features used in fired heaters. For example, high operating temperatures, or poor fuel quality (excessive trace metals and ash), may force the selection of costly, highly alloyed materials. Environmental considerations may necessitate extremely tall stack heights, and available plot-plan area may restrict dimensions.

An economic factor that has assumed greater importance in recent years is the shift toward shop assembly and fabrication in order to reduce field construction time and costs. A consequence of this shift is that provision for shipping clearance along the route from fabricator to jobsite has assumed the importance of a primary design criterion.

Fig. 1 shows a schematic for a typical fired heater, depicting the major structural components and their relationship to each other. These components will be taken up here.

Casing and structural framework

Typically, the outer wall or casing of the heater is fabricated from $^3/_{16}$-in. steel plate, reinforced against warping. However, for vertical cylindrical heaters (see Part I), in which the shell itself serves as a load-carrying structural member, the normal plate thickness can be as low as $^1/_4$ in. Floor plates, too, are normally designed for a thickness of $^1/_4$ in. Prevailing design guidelines call for sealing the heater casing plate by welding, in order to prevent air and water infiltration.

The heater's structural-steel framework provides load-carrying members, which permit lateral and vertical expansion of all parts of the fired heater. The framework also supports the tube coil, independently of the refractory. Common design practice calls for fireproofing the main structural columns to a specified height above grade, as well as the main floor beams.

When header boxes are provided to receive return-bend fittings, the minimum plate thickness is normally $^3/_{16}$ in., reinforced against warping.

Where appurtenances such as ladders and platforms are provided or anticipated, the structural design must be adjusted to carry such loads.

Refractories

The casing described above is lined internally with insulating materials. Aside from the basic function of preventing the steel structure from overheating, the insulation also serves to contain the firebox heat at high temperature by reradiating it to the tube coil. In addition, the internal insulation serves to minimize casing heat loss, and also functions as a barrier to prevent flue-gas particle migration to the steel casing. Such migration, in the case of sulfur-bearing fuels, may lead to acid corrosion of the steel plate.

In order to properly select and design a refractory lining for a fired heater, concern must be given to several important factors:

Extreme temperature. Exposure to temperatures beyond the design limitation of a refractory material can cause melting or fusion, and failure under load.

Thermal shock. Extreme or frequent temperature fluctuations can cause disintegration and spalling of refractory linings.

Mechanical stress. Abnormal vibration can contribute to the deterioration of some materials. Stresses due to the expansion and contraction of the structure can

*To meet the author, see Part I, which appeared on p. 98 of the June 19, 1978 issue. Part III will appear in the Aug. 14 issue.

cause the loss of lining integrity unless proper allowance is made in the mechanical design.

Erosion. Extremely fine particles such as flyash or catalyst being carried at high velocity in a flue-gas stream can cause erosion of the refractory material.

Chemical attack. Some fuels contain impurities that can react with various refractory constituents, causing slagging and failure of the refractory lining. Alkalis and acids, depending on the temperature and dewpoint of the flue gases, can attack the components of a refractory lining, causing corrosion and deterioration.

Cost. The economic evaluation of refractory materials and construction types is complicated because materials having the best insulating properties often fall short on mechanical strength. Thus, most refractory choices represent a compromise between insulating value and mechanical serviceability.

Insulation

Insulating systems for modern fired heaters fall into three basic categories:

Insulating firebrick (IFB). This is a porous brick with good insulation characteristics, manufactured by firing mixtures of sawdust, coke and high-alumina fireclays. Design temperature ratings of IFB range from 1,600°F to 2,800°F.

Typical IFB walls are suspended from the heater casing, and supported by horizontal steel angles. With this arrangement, heavy loadings on the lower bricks of a wall are avoided. Several concepts are employed to retain the wall in place. One method anchors individual bricks, at frequent intervals, to the steel casing with steel hooks or rods. Another method utilizes long steel rods placed through holes in the bricks, with the rods then anchored at intervals to the steel casing. Fig. 2 illustrates a typical wall construction.

When IFB walls are designed for vertical cylindrical casings, the "keying" effect of the bricks on the curved wall normally holds the wall in place without the use of tieback hardware such as hooks or rods.

Improved insulating effectiveness of IFB walls can be achieved by incorporating a backup layer of mineral-wool block insulation. This "brick and block" construction serves in countless operating installations.

With the current emphasis on shop preassembly of heater sections in modular form, the problems associated with shipping brick-lined modules have discouraged their use in new installations. By far, the more popular setting employed in conventional heaters today is the castable refractory-wall construction. Nevertheless, brick and block settings remain the standard for high-temperature specialty units such as steam-hydrocarbon reformer heaters and pyrolysis heaters.

Castable refractory. Castable refractory used in heater settings is normally an insulating castable applied by pouring or gunning. For shop-assembled modules, gunning under controlled conditions has been shown to be a very economical method of application. However, pneumatic placement of castable is a skilled craft, and the techniques of the applicator can mean the difference between success or failure of the installation.

Lumnite-Haydite-Vermiculite insulating castable (1:2:4 mix by volume) is an inexpensive material having excellent insulation characteristics. Because of its low cost, LHV is used extensively in heater applications. In addition, it has a very low expansion coefficient and is therefore used on large wall areas, without expansion joints.

A maximum temperature limit of 1,800° to 1,900°F precludes the use of LHV on exposed walls (unprotected by tubes) in close proximity to the flame burst. For such applications, proprietary castable refractory mixes are available.

It is to be noted that as the service temperature and density of the castable increase, the insulating effectiveness decreases, resulting in a need for additional thickness to achieve the same cold-face temperature. In many cases, dual-layer constructions are employed in which the high-temperature, high-density material is exposed to the flame, and the lower-grade, better-insulating material is provided as a backup layer.

Thicknesses are typically 5 in. for convection-section walls and radiant section walls protected by tubes, and 6 to 8 in. for exposed radiant walls, arches and floors. It is now fairly common to provide a facing of first-quality firebrick over castable areas on the floor, particularly when the heater is designed for liquid fuels.

As important as the choice of insulating material is, the key to a properly applied castable refractory wall is the anchoring system. The most popular system employs "V" clips or modifications thereof, welded to the steel casing. These clips are normally $\frac{1}{8}$ to $\frac{3}{16}$ in. dia., with anchor heights usually not less than 70% of the castable thickness. The preferred anchor material is austenitic stainless steel. Typical anchor spacings are a maximum of twice the lining thickness but not exceeding 12 in. on a square pattern for walls, and 9 in. on a square pattern for arches.

Ceramic fiber. Ceramic-fiber construction is the most current development in the field of fired-heater insulation. These linings normally consist of a hot-face layer followed by one or more layers of backup material. For the hot-face layer, ceramic fiber or blanket with a minimum density of 8 lb/ft^3 is recommended, with a minimum thickness of 1 in. Backup layers should also have a minimum thickness of 1 in. and a minimum density of 4 lb/ft^3. Mineral-wool block insulation can also be employed as backup material, provided that the fuel does not contain more than 1% (wt.) sulfur if it is fuel oil, or 1½% (vol.) H_2S if it is gas.

Advantages of ceramic-fiber installations derive primarily from their light weight, which permits a reduction of structural steel, and from their immediate availability for operation without special startup procedures such as curing, dryout or cold-weather precautions.

Because ceramic fiber is more porous than insulating castable, it is desirable to provide an internal protective coating on the casing plate to prevent corrosion.

Ceramic-fiber construction should not be employed in convection sections when soot blowing or steam lancing is contemplated.

The tube coil

The tube coil of a fired heater is the most important component of a heater installation. It is also a major

cost contributor to the overall heater investment.

Normally, the tube coil consists of a number of tubes connected in series by 180-deg return bends. In the event of internal coke deposition in the tubes, all-welded coils can usually be cleaned by a steam-air decoking procedure, or by a relatively new process using an abrasive propelled by a high-velocity gas stream. In many older installations—as well as in current high-temperature designs, wherein heavy oils having the potential for substantial coke deposition are heated—plug-type headers are employed to connect the tubes. In this construction, the tubes are rolled into, or welded to, the headers. The tubes can be cleaned by reaming or turbining. Occasionally, all-welded coils are provided with plug-type headers at certain key locations, simply to permit internal tube inspection.

Tube design

Principal factors affecting the selection of tubing material for elevated temperatures are service life, environment, and cost.

Specified service life varies substantially for individual heating applications, and even for the same application within different operating companies. For example, one company might use Type 304 stainless steel as a coil material with an expected life of, say, 8 to 11 years for a particular service. Another company might select a chromium-molybdenum steel, expecting to make tube replacements after about five years and perhaps replace all the tubes within seven years.

In the case of the Type 304 steel, the initial investment is substantially greater, as is the cost of the replacement of any tubes lost through faulty operation. More-continuous operation, however, with less downtime for tube replacement, may make Type 304 a more-economical long-term choice.

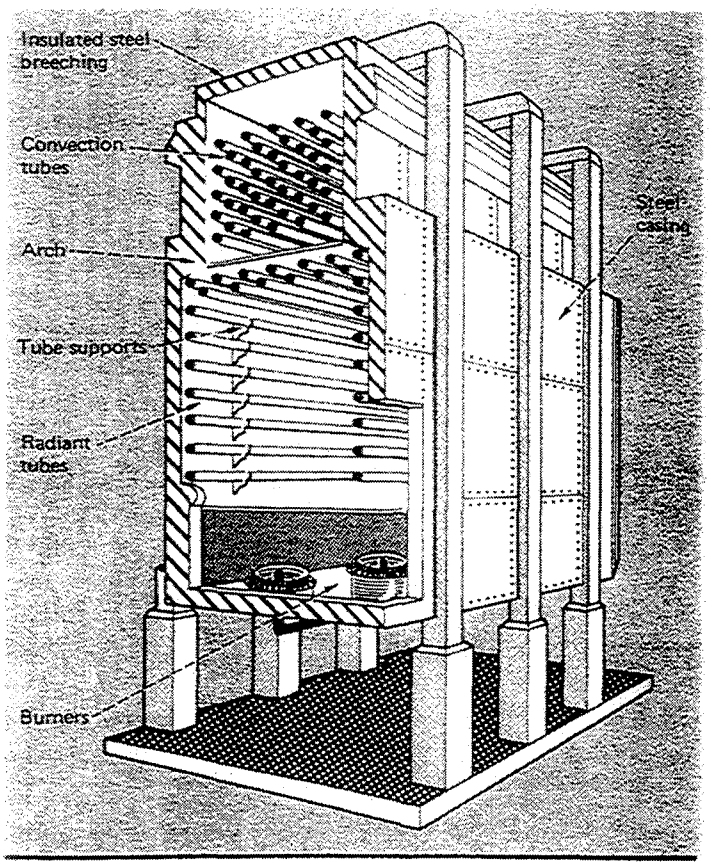
The temperature and stresses to which the tubing is subjected are as crucial as the media to which it is being exposed. In fired heaters, the tube metal temperature is always higher than the bulk fluid temperature at a given location. In addition, the temperature differential between tube wall and bulk fluid may increase as coke or scale is deposited on the inner wall. Therefore, consideration must be given not only to the initial tube-wall temperature but also to the maximum metal temperature that may result at the end of a run.

Also important is whether the stress is constant or cyclic. Thermal stresses that are created when a unit is starting up or shutting down can result in failure of steel components. Under certain conditions, such stresses can be of greater magnitude than the steady-state operating stresses.

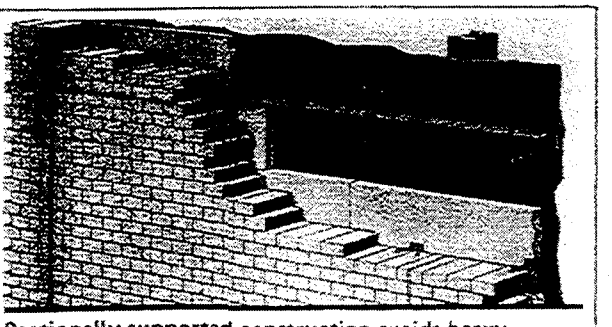
The medium to which the tubing is exposed affects the oxidation or corrosion behavior of the steel. If anticipated oxidation levels are severe, tube material selection must provide a high resistance to scaling. As for corrosion activity, the selection should reflect, where possible, data obtained from actual operating conditions at commercial units or pilot plants.

Probably the most important factor in the selection of tubing material is cost. A steel with excellent elevated-temperature properties will have limited application if its cost is prohibitive. One of the reasons for the relatively wide selection of fired-heater tube-coil materials is that a particular material is often the most economical for a given application. Further, within an individual fired heater, it is not unusual to include two or more different tube materials in the most economical tube-coil design.

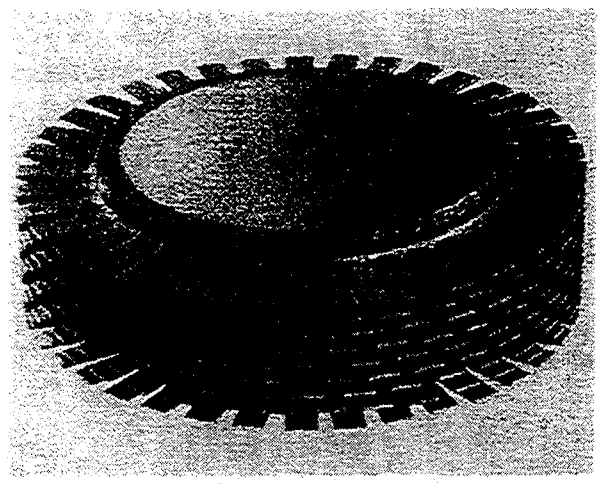
The use of a steel at high temperature levels will result in creep or permanent deformation even at stress levels well below the yield strength of the material. The tube will eventually fail by creep-rupture, even when a corrosion or oxidation mechanism is not active. For steels at lower temperature levels, creep effects are negligible, indicating that under such circumstances the



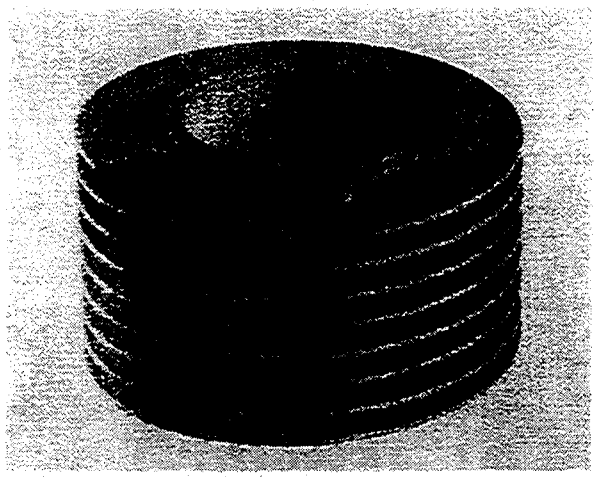
Typical fired-heater arrangement shows tube sections and structural components Fig. 1



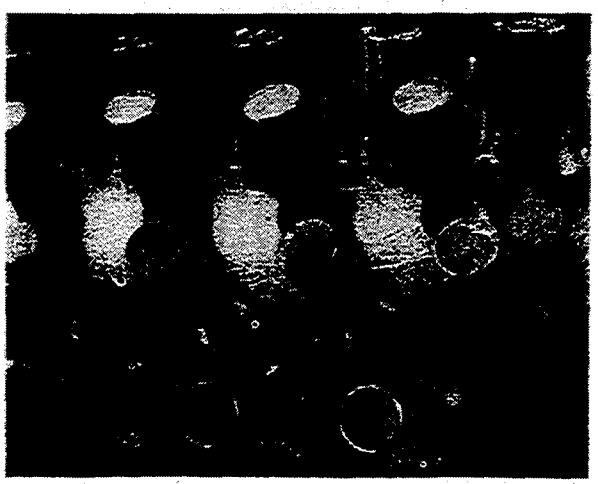
Sectionally supported construction avoids heavy loadings on the lower bricks in fired-heater wall Fig. 2



a. Serrated fins



b. Solid fins



c. Studs

Three types of extended-surface devices used in the convection section **Fig. 3**

tube will last indefinitely unless corrosion or oxidation effects manifest themselves.

There are, therefore, two different design considerations for heater tubes. At lower temperatures, in the "elastic range," the design stress is based on the yield strength. At higher temperatures, in the "creep-rupture range," the design stress is based on the rupture strength. For the temperature range where elastic and rupture stresses cross, the design tube-wall thickness requirement must satisfy both conditions.

Tubing materials

Carbon steel, the most widely used material for heater tubing, is suitable where corrosion or oxidation is relatively mild. The widespread usage of this material reflects its relatively low cost, generally good service performance and good weldability.

The alloy steels (Table I) used for elevated temperature service generally contain either molybdenum, chromium or silicon. The molybdenum is added principally to give higher strength; chromium is added to suppress graphitization and to yield improved oxidation resistance; and silicon is added to provide a further improvement in oxidation resistance.

The austenitic stainless steels are essentially alloys of iron, chromium and nickel and, as a group, are used for handling many corrosive materials or for resisting severe oxidation. Type 304, the most popular of the austenitic stainless steels, has excellent resistance to corrosion and oxidation and has high creep-strength. Types 321 and 347 are similar to Type 304 except that titanium and columbium, respectively, have been added. These additives combine with carbon and minimize intergranular corrosion that may occur in certain media after welding.

Type 316, which contains molybdenum, is used for high-strength service up to about 1,500°F and will resist oxidation up to about 1,650°F. For service above 1,600°F, Types 309 and 310, which contain about 25% chromium and 12 and 20% nickel, respectively, are used. These steels have excellent strength at these temperatures and, because of their chromium content, can be used in applications where extreme corrosion or oxidation is encountered.

Alloy 800 (20% chromium and 32% nickel) has excellent strength at up to 1,800°F and resists oxidation and carburization. It is used for tubes in pyrolysis heaters and steam superheaters, and for outlet pigtails and manifolds on steam hydrocarbon-reformer heaters.

Centrifugally cast materials, such as HK-40 (25% chromium and 20% nickel), are widely used for tubing in steam hydrocarbon-reformer heaters and in pyrolysis heaters.

Return bends. The least expensive method of connecting the tubes is to join them by means of 180-deg return bends. The return bends are welded onto the tube ends in this arrangement, which typifies the majority of modern heaters. As noted already, internal cleaning of all-welded coils is performed by steam-air decoking procedures, or by recently developed high-velocity abrasive techniques. All-welded design permits return bends to be positioned either in the path of the flue gases, where they function as heat absorbing surfaces, or

in header boxes external to the firebox and the flue-gas flow. Return bends can be of wrought or cast material.

Plug-type headers. Many types of plug-type headers using several closure designs have been developed. Compared to 180-deg return bends, these headers are somewhat more expensive and their use in new fired-heater equipment is relatively rare. As noted above, plug-type headers are used where mechanical cleaning of the tubes by turbining is anticipated and, occasionally, where tube internal inspection is planned. Plug-type headers cannot be placed in the firebox or in the path of the flue gases; they must be installed in header boxes external to the firebox and the flue-gas flow. Plug-type headers consist of cast material and, since they are external to the heat-transfer zone, can be designed for a lower temperature than the heat-absorbing tubes.

Extended surface improves convection

The surface area required in the convection section is controlled by film resistance on the flue-gas side. As a means of increasing the convection transfer rate per lineal foot of tubing, extended-surface devices have found almost universal acceptance. In today's designs, bare-tube convection sections are generally reserved for those relatively rare applications where the combustion of extremely poor grades of fuel oil presents the risk of heavy ash deposition on the convection-section tubes.

Popular devices employed in heater convection sections are reviewed here. Note that extended surface is never installed in the radiant section.

Serrated fins. This arrangement uses a V-notched or serrated fin that is helically wrapped around and continuously welded to the tube (Fig. 3a). Fins can be supplied in many combinations of thickness, height and density (number of fins per unit length of tube). Typically, thickness ranges from 0.035 to $\frac{3}{16}$ in., height from $\frac{1}{4}$ to $1\frac{1}{2}$ in., and density from 2 to 7 fins per in.

Solid fins. This type is a noninterrupted fin that is helically wrapped around and continuously welded to the tube (Fig. 3b). These fins are available in the same ranges of thickness, height and density as serrated fins. Solid fins are mechanically stronger than serrated fins, but generally display a slightly lower heat-transfer rate for the same fin configuration and flue-gas mass flow.

Studs. Here, nominally cylindrical studs are flash-welded to the tube circumference (Fig. 3c). A stud diameter of $\frac{1}{2}$ in. is fairly standard for the industry, although $\frac{3}{8}$-in. studs are sometimes specified. Stud height ranges from $\frac{1}{2}$ to 2 in.

Cylindrical studs are the only extended surface that can be effectively employed on tubes positioned normal to, as well as parallel to, the flow of flue gas. Studs ordinarily cost more than finned tubing.

Dimensions of extended-surface devices used in gas-fired heaters ($\frac{1}{2}$-in. dia. studs or 0.05-in. thick [minimum] fins) are somewhat smaller than those used in oil-fired units ($\frac{1}{2}$-in. dia. studs or 0.10-in. thick [minimum] fins). Fin dimensions should preferably be limited to a maximum height of $\frac{3}{4}$ in. and a maximum density of 3 fins per in.

Table II presents calculated maximum tip temperatures for various extended-surface devices.

Choice of heater-tube materials is restricted by limiting design metal temperature — **Table I**

Material	Type or grade	Limiting design metal temperature, °F
Carbon steel	B	1,000
Carbon-½ Mo	T1 or P1	1,100
1¼ Cr-½ Mo	T11 or P11	1,100
2¼ Cr-1 Mo	T22 or P22	1,200
5 Cr-½ Mo	T5 or P5	1,200
7 Cr-½ Mo	T7 or P7	1,300
9 Cr-1 Mo	T9 or P9	1,300
18 Cr-8 Ni	304 or 304H	1,500
16 Cr-12 Ni-2 Mo	316 or 316H	1,500
18 Cr-10 Ni-Ti	321 or 321H	1,500
18 Cr-10 Ni-Cb	347 or 347H	1,500
Ni Fe-Cr	Alloy 800H	1,800
25 Cr-20 Ni	HK-40	1,850

Maximum tip temperatures for some materials used in extended-surface devices — **Table II**

Extended-surface material	Tip temperature, °F
Fins	
Carbon steel	850
5 Cr	1,100
11-13 Cr	1,200
18 Cr-8 Ni	1,500
Studs	
Carbon steel	950
5 Cr	1,100
11-13 Cr	1,200
18 Cr-8 Ni	1,500

Limiting metal temperatures for some materials used to fabricate tube supports — **Table III**

Material	Type or grade	Limiting design metal temperature, °F
Carbon steel	A 283 Gr C	800
5 Cr-½ Mo	Gr C5	1,100
Alloy Cast Iron	A 319 Class III, Type G	1,400
18 Cr-8 Ni	Gr CF8	1,500
25 Cr-12 Ni	Type HT	1,800
50 Cr-50 Ni		1,800
50 Cr-50 Ni-Co	IN 65	1,900
60 Cr-40 Ni		2,000
25 Cr-20 Ni	Gr HK40	2,000

Tube supports and guides

Proper mechanical design of a fired heater requires that the tube coil be adequately supported by tube hangers and tubesheets that are connected to the structural framework of the heater and not to the refractory.

Horizontal tubes. Horizontal coils having internal return bends are supported by intermediate tube hangers in the radiant section, and by tubesheets in the convection section. When the return bends (or plug-type headers) are located external to the heater environment in header boxes, end tubesheets and intermediate tube supports are provided. A common arrangement features header boxes having external return bends for the convection section, and internal return bends for the radiant section.

The design of intermediate tube supports used in a horizontal radiant section should ensure that the supports can be removed without tube removal and with a minimum of refractory replacement. It is also desirable for intermediate radiant tube supports to have restraints, or "keepers," to prevent the tubes from lifting off the supports during operation. Intermediate tubesheet castings in the convection section should be sectionalized to minimize the amount of tube removal needed when a casting is replaced. Typically, the maximum unsupported length of horizontal tubes should not exceed 35 times the tube outside diameter, or 20 feet, whichever is less.

Vertical tubes. Vertical coils can be supported either from the top or from the bottom. Top-supported tubes are provided with bottom guides, bottom-supported tubes with top guides. No intermediate supports are used. If necessary, intermediate guides are provided to restrain the vertical tubes from bowing inward toward the flame, or laterally toward adjacent tubes.

Materials used for tube supports

End tubesheets for tubes with external return bends or plug headers are normally of carbon-steel plate, $\frac{1}{2}$ in. thick. If the tubesheet temperature exceeds 800°F, alloy material should be used. End tubesheets are insulated on the hot side with castable refractory—generally 3 in. minimum in the convection section and 5 in. minimum in the radiant section.

In the radiant section, intermediate supports for horizontal tubes—as well as top supports for vertical tubes—are usually of cast 25% chromium-12% nickel, although many companies prefer the higher-grade 25% chromium-20% nickel. The same materials are used for intermediate guides and bottom guides of vertical tubes, although the use of 18% chromium-8% nickel for bottom guides is fairly common. Bottom supports for vertical tubes normally consist of alloy cast iron and are shielded from flame radiation by the floor refractory.

When the firing of fuel oil containing more than 100 ppm of vanadium is contemplated, radiant-section supports and guides should be made of higher alloys such as 50% chromium-50% nickel, 60% chromium-40% nickel, or IN-657, which is a proprietary 50% chromium-50% nickel alloy stabilized with columbium for additional high-temperature strength. These materials, however, are substantially more expensive than the conventional, lower-grade alloys.

Intermediate tubesheet castings exposed to hot gases in the convection section are normally fabricated from the same materials used for radiant-section supports exposed to flame radiation. In the colder flue-gases, alloy cast iron is commonly employed. Table III lists a variety of tube-support materials, and their usual temperature limits.

Burners

The fundamental criteria for selecting a burner include (1) the ability to handle fuels having a reasonable variation in calorific value, (2) provision for safe ignition and easy maintenance, (3) a reasonable turndown ratio between maximum and minimum firing rates, and (4) predictable flame patterns for all fuels and firing rates.

Gas-fired burners

Burners designed for gaseous fuel only are classified into two basic categories: premix inspirating and raw-gas burning.

Premix inspirating. The premix burner relies on the kinetic energy made available by the expansion of the fuel gas through an orifice to inspirate and mix combustion air prior to ignition at the burner tip. Approximately 50 to 60% of the combustion air is inspirated as primary air into the burner ahead of the ignition point.

Some of the advantages of this type of burner are:

1. Operating flexibility is good over a range of conditions. The amount of air inspirated varies with the fuel-gas pressure, and consequently requires only limited adjustment of secondary (noninspirated) combustion air. Premix burners can operate at low excess-air rates and are not significantly affected by changes in wind velocity and direction.
2. Flame length is short, and flame pattern sharply defined at high heat-release rates.
3. Burner orifices or spuds are fairly large, and, since they are located in a cold zone, are less subject to plugging than the smaller openings on noninspirating gas burners.

Some of the disadvantages of inspirating burners are:

1. Relatively high gas pressures must be available. Below a gas pressure of 10 psig at the burner, the percentage of inspirated air falls rapidly and flexibility is greatly reduced.
2. Flashback of the flame from the burner tip to the mixing orifice may occur at low gas pressures, or when the fraction of gases having high flame-propagation velocities, such as hydrogen, becomes too high.
3. The noise level of premix inspirating burners is higher than that of noninspirating types.

Raw-gas burning. The nozzle-mixing, raw-gas burner receives fuel gas from the gas manifold without any premixing of combustion air. The gas is then burned at a tip equipped with a series of small ports (Fig. 4a).

Some of the advantages of this type of burner are:

1. It has the greatest available turndown ratio for any given combustion condition.
2. It can operate at very low gas pressures on a wide variety of fuels and without flashback.
3. Noise level is reasonably low.

Some of the disadvantages of raw-gas burners are:

1. Flexibility is limited over its wide turndown range. Because no primary air is inspirated, combustion-air adjustments must be made over the full operating range of the burner.

2. The drilling of the burner ports is very sensitive, and any enlargement of the port opening will generally result in unsatisfactory flame conditions.

3. Flames tend to lengthen, and flame conditions become unsatisfactory as the burner is pushed beyond its design level.

4. The gas orifices or burner ports are exposed to the hot zone and are subject to plugging at low velocities and high temperatures.

Oil-fired burners

Special measures must be provided for burning fuel oil, since mixing of fuel and combustion air occurs in the gaseous phase. To accomplish this, all liquid-fuel burners use atomizing devices to break up the liquid mass into micron-size droplets. This increases the surface-to-mass ratio, thereby allowing extremely rapid heating and vaporization of the oil mass.

Oil burners in fired heaters almost always utilize steam as the atomizing medium. Such burners are designed with a double pipe in the feed tube to inject the steam and oil separately into a mixing chamber or atomizer immediately ahead of the burner tip. Steam pressure is slightly higher than oil pressure upon entering the atomizer, where the steam mixes with the oil due to the shearing action. Furthermore, oil in contact with hot water vapor tends to foam or emulsify, thus contributing to the atomization process. The steam and the finely dispersed oil then issue through a series of orifices into the turbulent air stream.

For proper combustion of oil, the following important requirements should be met:

1. The oil must be heated high enough that its viscosity is not greater than 150–200 Ssu.

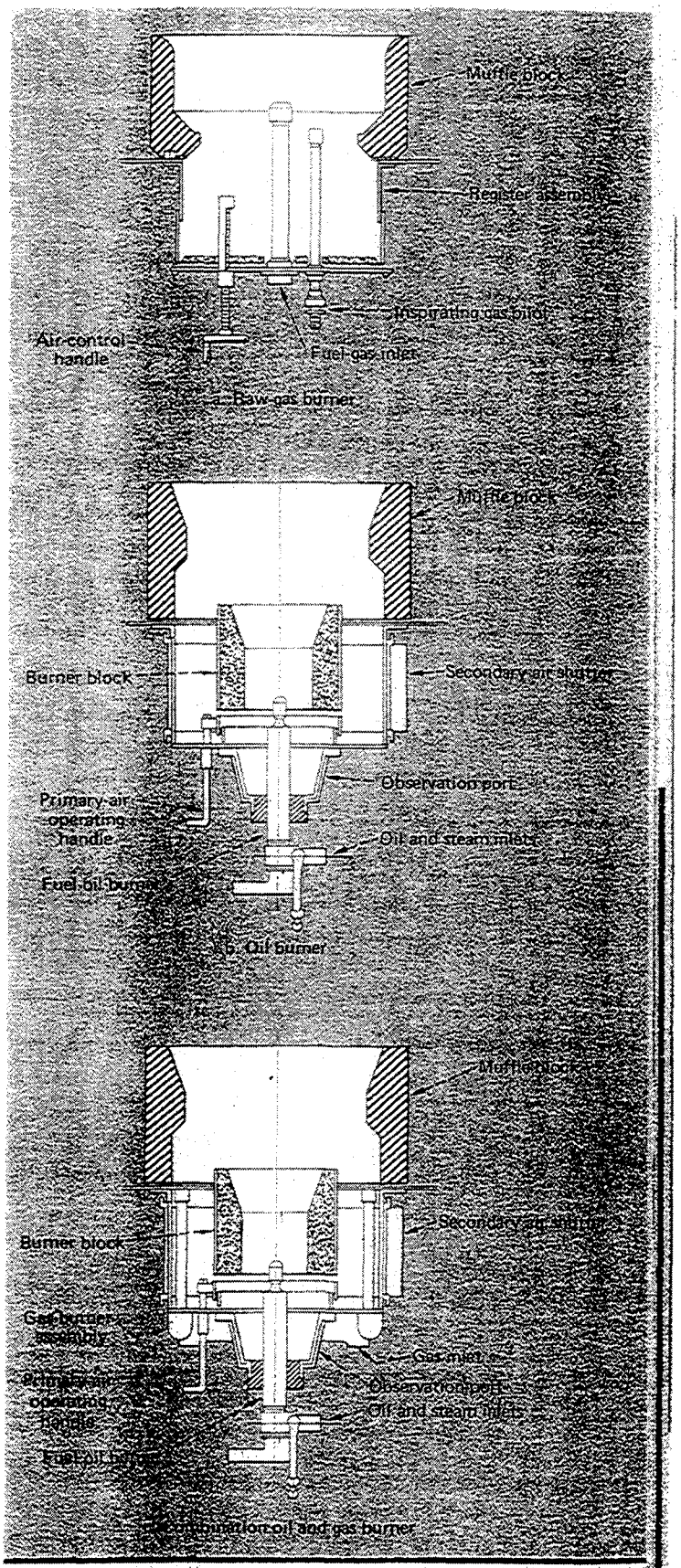
2. Oil pressure must be held constant, typically at about 75 psig.

3. The steam delivered at the burner must be absolutely dry. If available, a moderate superheat of approximately 50°F is preferred. Typical steam pressure is about 100 psig.

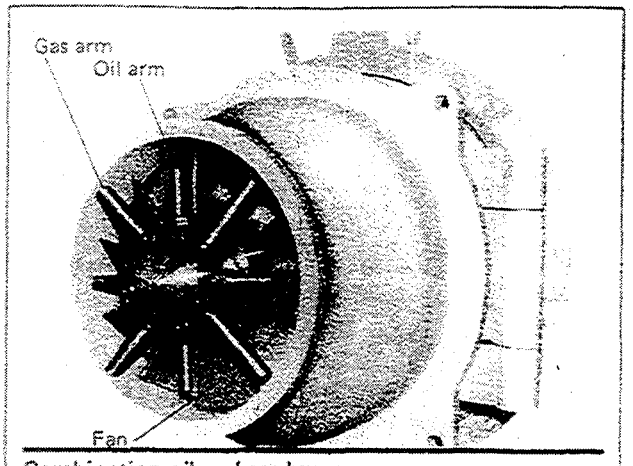
Fig. 4b depicts a typical oil burner. To maintain flame stability during the firing of high-viscosity fuels, natural-draft oil burners should be of the double-block design, as illustrated. Atomizing steam needs fall in the range of about 0.15 to 0.35 lb steam per lb of oil.

When volatile fuels such as naphtha or gasolines are burned, a safety interlock should be provided on each burner. The interlock sequentially shuts off the fuel, purges the oil gun with steam, and shuts off the steam purge before the oil gun can be removed.

For those rare instances when oil must be burned and steam is not available, air atomization or mechanical atomization can be employed. The operating requirements of air-atomized oil burners are similar to those of steam-atomized ones, although a slightly higher oil temperature may be needed, to compensate for the cooling effect of the atomizing air. In addition, air (supplied by either a blower or a compressor) must be



Burner designs must offer safe operation and easy maintenance, and be able to handle variations in fuel quality **Fig. 4**



Combination oil and gas burner uses rotor fan to mix fuel and air Fig. 5

automatically modulated to adjust for changes in the firing rate. Uniformity of air supply is essential.

Mechanically atomized units take advantage of the oil's kinetic energy to atomize the fuel stream in the tip itself. Oil-temperature and viscosity ranges are similar to those needed for steam atomization. Oil must be available at a supply pressure in the area of 300 psig. When wide turndown ratios are required, oil pressures may go as high as 1000 psig.

Combination oil and gas burners

Combination burners (Fig. 4c) are designed to burn all-oil, all-gas, or any combination of oil and gas simultaneously. Typically, these burners feature a double-block design in which a single oil gun is arranged in the center of an array of gas nozzles.

Gas firing in the newer combination burners is almost always accomplished with raw-gas nozzles. Separate register adjustments allow independent control of primary air for oil combustion, and secondary air for fuel-gas combustion.

Radiant wall burners

The use of radiant-wall burners is generally confined to high-temperature specialty units such as steam hydrocarbon-reformer heaters and pyrolysis heaters. Because the capacity of an individual burner is generally low, these heaters require many more burners than a conventional fired heater having the same total heat release. However, this feature affords the advantages of improved heat distribution and pinpoint firing control. Radiant-wall burners generally burn only fuel gas, and are available as either inspirating or as raw-gas types.

The former type is designed to inspire 100% of the combustion air requirement. The flame from inspirating burners is flat and radial, rather than projected into the firebox. Impingement of the flame on a refractory burner block heats the block to incandescence, thus achieving the radiant-wall effect.

The raw-gas design is linear in shape, with the burners installed along two or three firing levels on the heater wall. The flames are thrown upward along the vertical or inclined firebox wall to create a radiant-wall effect. This burner type has been adapted to burn light oils in a radiant-wall mode of firing.

Rotor-fan burners

These burners (Fig. 5) exploit the kinetic energy of fuel under pressure to generate, by the reaction principle, mechanical work sufficient to rotate a built-in fan. The design of the fan assures proper proportioning of air in relation to fuel flow over a wide operating range.

Fuel-air mixing is produced by the discharge of fuel from the fan's rotating driver arms—through multiple orifices—into the air stream created by the fan's action. Thorough and instantaneous mixing of fuel and air is achieved. Relatively low amounts of excess air are needed to obtain complete, short-flame combustion.

The rotor-fan burner is used primarily to debottleneck existing heaters by reducing flame impingement in overloaded fireboxes, or by increasing heat input into draft-limiting heaters.

High-intensity burners

Several high-intensity burners are available for fired-heater applications. In general, they feature a large, cylindrical-shaped, refractory-lined combustion chamber. Combustion is fully established in this chamber, but not completed. By means of the circulation patterns developed in the chamber, flames of controlled shape and size can be produced at relatively low excess air. High-intensity combustion expells the flue gases at high velocity and temperature, producing very uniform firebox-temperature profiles.

High-intensity burners require a forced-draft air supply at relatively high pressures. Depending on the specific design, air pressure ranges from 6 to 20 in. H_2O.

Pilot burners

These units are supplied as an integral part of the main burner. They are most often installed where it is desired to simplify burner lightoff procedures, particularly with oil firing; where an extreme turndown to a fixed, minimum load is required; where intermittent on-off operation is required; and where extreme modulation of firing rate is needed.

The primary disadvantage of pilot burners is that they constitute a potential source of gas leakage into the firebox. The possibility always exists of a pilot being accidentally extinguished, permitting gas to be admitted to the heater during a shutdown. Also, because of their small port drillings, pilot burners clog easily and should be routinely inspected and cleaned.

Pilot burners are almost always gas-fired and are usually fueled from an outside source such as a propane or LPG drum which is not part of the process system. If the pilots are fueled from the main burner supply line, the gas offtake to the pilots must be upstream of the control and block valves.

Burner noise—what to do about it

Burner noise results from the flow of fuel and air through the burner and from the combustion process itself. Although at present little can be done to reduce combustion noise, it is possible to minimize propagation of sound to the outside of the heater.

Noise emitted by premix-inspirating gas burners originates primarily at the premix-venturi inspirator, and secondarily at the outlet-burner orifices. A method of reducing the inspirator noise is to replace the single-orifice inspirator with a multi-orifice inspirator. Further noise suppression can be achieved by fitting a silencer, or mute, on the primary air intake.

In raw-gas burners, the noise is emitted by the fuel gas as it passes through the burner nozzles. Noise from liquid-fuel burners is generated in the same way as in raw-gas burners, except that the noise is lower in frequency. Combustion noise from these burners can be effectively suppressed by acoustically absorbent air-intake plenums. These guidelines for plenum-chamber design will ensure reasonable noise reduction:

1. As much of the plenum interior as possible, including the heater wall, should be lined with acoustic material.
2. Absorbent surfaces should be arranged so that prior to escaping, sound waves from the burner air intake will undergo several reflections. For greatest effectiveness, the acoustic material should have a density of about 6 lb/ft^3 and its thickness should exceed 4 in.
3. Steel plate with a minimum thickness of $\frac{1}{8}$ in. should be used for plenum fabrication. Plenums should enclose the burner registers and be undercoated with sound-deadening material.
4. Provision should be made for inspection and for draining oil leaks. Acoustic lining should be omitted where drips collect.
5. All of the combustion air should be taken from the plenum chamber. There should be no line of sight from the burner or the plenum interior to the outside. Hand holes, and openings for external air register control, oil gun, etc., should be well sealed.

Dampers

The function of the stack damper is to control the heater's draft by maintaining a negative pressure of approximately 0.05 in. H_2O in the region directly below the convection section. The damper adjusts as necessary to maintain this negative pressure.

Stack dampers of single-leaf construction are used for small-diameter stacks, and multi-leaf construction for large-diameter stacks. In rectangular ductwork, which carries the flue gases between a heater and a separate stack, louvre-type dampers are normally used.

Stack dampers are usually manually operated from grade by means of cables. In the case of large dampers, it is now fairly common to maneuver them with pneumatic operators.

Table IV gives maximum flue-gas temperatures for various damper materials.

Flue-gas temperature restricts use of damper materials — Table IV

Material	Gas temperature, °F
Carbon steel	900
Cast iron	900
Alloy cast iron	1,000
11-13 Cr	1,200
18 Cr-8 Ni TP304	1,500
25 Cr-12 Ni TP309	1,800
25 Cr-20 Ni TP310	1,800

Cleaning the convection section

In order to maintain maximum thermal efficiency in a fired heater, it is necessary to keep convection heat-transfer surfaces clean. Although the extended surface of the convection section improves heat transfer, its physical arrangement renders it susceptible to the accumulation of fuel-ash deposits when oil fuels are fired. The major fouling constituents of an oil fuel are sulfur, vanadium, sodium and ash. The latter is a most important component, since a high ash content will significantly increase the deposition rate.

The viscosity range of the fuel also influences ash deposition. The larger ash particles that accompany the burning of high-viscosity fuels carry over into the convection section to increase the fouling rate. Current methods used for onstream cleaning include manual lancing, soot blowing and water washing.

Manual lancing. This method requires a minimum of capital investment. Lance doors provided on the convection section sidewall permit workers to insert a steam or air lance and blow deposits off convection surfaces in the vicinity.

The effectiveness of lancing is poor by comparison with that of soot blowing. In addition, lancing requires the participation of more than one operator to transport the equipment from grade to the convection section, which is normally located at a relatively high elevation. Consequently, the success of a lancing program is dependent on operator initiative.

At the present time, manual lancing facilities are generally confined to retrofit installations on existing heaters—especially where a soot-blower retrofit would involve substantial modifications to convection coils.

Some operators have had reasonably good success in removing convection-section deposits by sandblasting using a nozzle inserted manually through the lance doors. However, the erosive potential of sandblasting on the convection-section refractory and on the heating surfaces must not be overlooked.

Rotating-element soot blowers. Also known as a fixed rotary type soot blower, this device is actually a multi-nozzle steam-lance tube installed in the convection section. A mechanical valve and drive assembly external to the convection section rotates the element, and automatically opens and closes the steam-supply valve.

The effectiveness of this type of soot blower is limited by the physical strength of the ash. Some combinations of ash chemistry and flue-gas temperature produce deposits that cannot be removed with such soot blowers. Further, highly corrosive fuels will materially reduce the life of the element or the lance tube.

Retractable-lance soot blowers. This apparatus differs from the rotating-element type in that the lance tube remains retracted from the convection section when not in use. During soot-blower operation, the lance is extended into the heater by means of an external drive.

The lance is equipped with two cleaning nozzles, of larger diameter than those installed along the length of the multi-nozzle rotating-element design. The retractable-lance blower cleans more effectively than does the fixed-rotary type, because its two nozzles permit a greater concentration of the cleaning-fluid stream. In addition, the retractable lance is exposed to high-temperature flue gases only during the cleaning cycle.

Notes on soot blowers

The effective cleaning range of a soot blower is dependent on its orientation, the flue-gas temperature, the arrangement of heating surface, and the blowing pressure. As a general guide, the vertical blowing capability of fixed-rotary soot blowers will not exceed 3 rows up and 3 rows down, with a maximum horizontal blowing radius of 3.5 feet. Corresponding limits for retractable lances are 4 rows up and 4 rows down, with a maximum horizontal blowing radius of 4 feet.

The rate of steam consumption of a retractable-lance soot blower will vary from 8,000 to 12,000 lb/h, depending upon the pressure at the nozzle. The cycle time required to operate a retractable lance is approximately 2½ min. The fixed-rotary soot blower consumes steam at a rate of about 10,000 to 14,000 lb/h with a blowing time of approximately 40 s. Soot blowers are normally operated once per 8-h shift.

Steam for soot blowing is normally taken from supply lines at the relatively low pressures of 150 to 200 psig. Below 150 psig, soot blowing is virtually ineffective. More effective cleaning can be achieved with pressures higher than 200 psig. When available, pressures of 400 to 600 psig should be considered.

A completely automatic sequencing control system is recommended for multi-blower installations. Upon activation by one pushbutton, each of the soot blowers will operate in a predetermined sequence until the entire blowing cycle is completed.

Water washing

Onstream water washing of convection sections represents a very effective method of removing deposits from the tubes. Permanently installed networks of high-alloy headers and nozzles are positioned at selected elevations in the convection bank. Copious quantities of condensate or fresh water flow across the tube surfaces. Contact of the water with the hot tubes instantly vaporizes the water, thereby dislodging soot deposits.

Frequency of water washing depends on the nature and extent of the deposits. Typically, water washing is peformed about once per week. Understandably, care must be exercised to avoid getting water on the refractory or on intermediate tube-support castings. Water washing of stainless-steel tube coils is not recommended in view of the danger of chloride stress corrosion.

Performance monitoring

Getting the optimum performance from a heater requires the close monitoring of key variables on both the process and combustion sides. Data can be taken that will reveal excess air, thermal efficiency, and heat absorption. In addition, certain data provide an indication as to how well the heater is being fired.

Process stream flowrate. In most applications, where provision is made for individual-pass flow control, the flowrate to each parallel pass should be monitored. This is recommended for multi-pass heaters processing liquid hydrocarbons, where low flowrate in an individual pass can lead to excessive vaporization, increased pressure drop and further flow reduction—culminating in overheating and possible rupture of the tube.

Fuel firing rate. The rate of fuel input is normally controlled by the process-fluid outlet temperature. Measuring the fuel firing rate permits direct determination of firebox heat release from the fuel's heating value.

Process-stream temperatures. If individual-pass flow control is planned, it is recommended that indicators be installed to show the fluid outlet temperature after each parallel pass. These temperatures can be used as a guide for adjusting the flowrates of each pass, as well as for determining the process-side heat absorption. Measuring the outlet fluid temperatures after each parallel pass in the radiant and convection sections enables one to determine the duty split on the process side between the radiant and convection sections.

Flue-gas temperatures. Measurement of the flue-gas temperature leaving the radiant section serves as an index to the firing balance in the firebox, and also as an indicator of overfiring conditions. This temperature measurement should be made at 50-ft intervals along the length of the firebox. Such measurements are often useful for establishing the maximum firing rate.

Flue-gas temperatures should also be monitored at the inlet to each convection coil, and at the outlet from the convection section. These temperatures provide an indication of heater efficiency and of convection tube fouling.

Flue-gas draft profile. Draft measurements should be taken at the firebox near the burner level, at the inlet to the convection section, at the outlet from the convection section, and at a point downstream of the stack damper. Draft readings provide information on the pressure drop of combustion air and flue gas. The information is helpful for adjusting the burner registers and the stack-damper. Draft readings also indicate how close the heater is to its limiting operating conditions.

Flue-gas sampling. Provisions for flue-gas sampling are recommended at the exit of the radiant section and at the outlet of the convection section. Sampling to measure the oxygen in the flue gas at the first site provides an indication of the operator's firing technique. Measurement of combustibles is also recommended for this location.

A determination of the oxygen content in the flue gas leaving the convection section is also needed for calculating the heater's combustion efficiency. If the oxygen content of the gases leaving the radiant section is known, the extent of air leakage into the convection section can be estimated.

Tubeskin temperatures. Tubeskin thermocouples are recommended, as a minimum, for the outlet tube of each pass and for one shock tube of each pass. Skin temperatures help set maximum firing rates. They also serve as indicators of local overheating.

FIRED HEATERS—III

How combustion conditions influence design and operation

This third of four articles highlights some of the major process considerations that affect the design of fired heaters. Basic combustion principles are reviewed, as well as methods for determining coil arrangements, surface area, fluid pressure drops, and stack height.

*Herbert L. Berman, Caltex Petroleum Corp.**

☐ The process design for a fired heater must reconcile complex relationships between a number of variables. Among these are the physical properties and phase behavior of the process fluid, the heating value and combustion behavior of the fuel(s), the economic proportioning of heating duty between the convection and radiant sections, the pressure drops of the process fluid and the flue gas, and the economic stack height. The relative importance of these variables will be examined here.

Combustion basics

For those fuels containing hydrogen, two sets of heating values are reported. The gross or higher heating value, HHV, is determined by assuming that all of the water vapor produced in the combustion process is condensed and cooled to 60°F. The net or lower heating value, LHV, assumes that the water vapor formed by combustion remains in the vapor phase and is numerically equal to the gross heating value less the latent heat of vaporization of the water.

In concept, the gross heating value can be considered as the *actual* heat release, whereas the net heating value can be considered as the *useful* portion of the actual heat release. For those fuels that do not contain hydrogen, e.g., CO, only one heating value is reported.

Unlike the boiler industry, which works on the basis of the gross heating value, the fired-heater industry almost always uses a net value. In this discussion, any reference to heat of combustion, heat input, or thermal efficiency will imply a net basis.

*To meet the author, see Part I, which appeared on p. 98 of the June 19, 1978 issue. Part II appeared in the July 31 issue. Part IV will conclude the series on Sept. 11.

Although the chemistry of the combustion process is exceedingly complex, it can be readily simplified in terms of final reaction products. As with all chemical reactions, in order for the combustion process to go to completion within a reasonable time, an excess of one of the reactants must be present. Under no circumstances can an excess of fuel be tolerated. Combustion in a fired heater normally uses air as the source of oxygen. Consequently, there must be an excess of air above stoichiometric requirements to ensure complete combustion of the fuel.

Reasonable excess-air requirements for natural-draft fired heaters are 20% for gas firing and 25% for oil firing. For forced-draft heaters, in which a greater degree of air control is possible, reasonable design excess-air values are 15% and 20% for gas and oil firing, respectively.

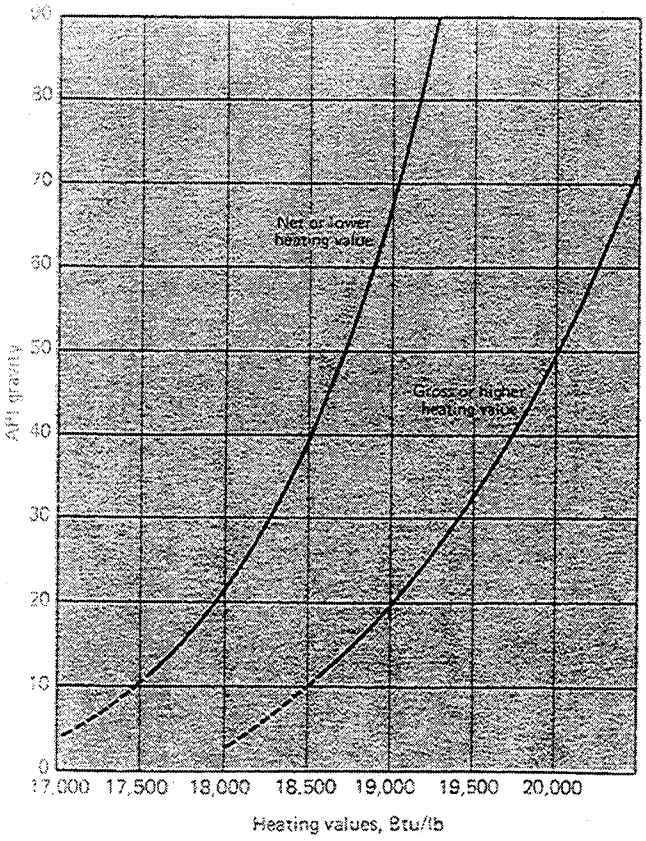
Fuel gas

Table I presents basic combustion constants for components of most industrial gaseous fuels. The following example illustrates the determination of heating value and stoichiometric air requirement for a multicomponent gaseous fuel. (Basis: 100 lb-moles of fuel.)

Component	Molecular weight	Mole %	Lb
CH_4	16.042	86.0	1,379.6
C_2H_6	30.068	8.6	258.6
C_3H_8	44.094	1.3	57.3
H_2	2.016	1.5	3.0
CO_2	44.010	2.6	114.4
			1,812.9

Properties used to predict heating values and air requirements for gaseous fuels — Table 1

Gas	Formula	Molecular weight	Heat of combustion, Btu per lb		Combustion air,
			Gross	Net	Lb per lb of combustible
Carbon monoxide	CO	28.01	4,347	4,347	2.462
Hydrogen	H_2	2.016	61,095	51,623	34.267
Methane	CH_4	16.042	23,875	21,495	17.195
Ethane	C_2H_6	30.068	22,323	20,418	15.899
Propane	C_3H_8	44.094	21,669	19,937	15.246
n-Butane	C_4H_{10}	58.12	21,321	19,678	14.984
n-Pentane	C_5H_{12}	72.146	21,095	19,507	15.323
n-Hexane	C_6H_{14}	86.172	20,966	19,415	15.238
Ethylene	C_2H_4	28.052	21,636	20,275	14.807
Propylene	C_3H_6	42.078	21,048	19,687	14.807
Butylene	C_4H_8	56.104	20,854	19,493	14.807
Benzene	C_6H_6	78.108	18,184	17,451	13.297
Toluene	C_7H_8	92.134	18,501	17,672	13.502
p-Xylene	C_8H_{10}	106.16	18,633	17,734	13.663
Acetylene	C_2H_2	26.036	21,502	20,769	13.297
Naphthalene	$C_{10}H_8$	128.164	17,303	16,708	12.932
Ammonia	NH_3	17.032	9,667	7,985	5.998
Hydrogen sulfide	H_2S	34.076	7,097	6,537	6.005



Heating values of liquid fuels have been correlated as a function of API gravity — Fig. 1

Component (contd.)	LHV Btu/lb	Btu	Lb air per lb fuel	Lb air
CH_4	21,500	29,654,500	17.195	23,720
C_2H_6	20,420	5,280,100	15.899	4,110
C_3H_8	19,940	1,142,400	15.246	874
H_2	51,620	154,900	34.267	103
CO_2	—	—	—	—
		36,231,900		28,807

$$\text{Average molecular weight} = \frac{1,812.9}{100} = 18.13$$

$$\text{Net heating value} = \frac{36,231,900}{1,812.9} = 19,990 \text{ Btu/lb}$$

$$\text{Air requirement} = \frac{28,807}{1,812.9} = 15.9 \text{ lb air/lb fuel}$$

Fuel oil

Unlike gaseous fuels, the heating value of a fuel oil is normally not developed from a component analysis. Instead, the heating value of a liquid fuel can be expressed as a function of the specific gravity of the oil, with an accuracy sufficient for most engineering computations. Fig. 1 depicts the relationship between the gross and net heating values of fuel oil as a function of API gravity. The API gravity, in turn, is related to the specific gravity, as follows:

$$°\text{API} = \frac{141.5}{\text{Sp. gr.}} - 131.5 \qquad (1)$$

If a fuel oil analysis is available, the stoichiometric air requirement can be approximated from the weight

Nomenclature

A_c	Convection surface area, ft²
A_f	Fin surface of extended-surface tube, ft²/ft
A_i	Internal tube surface, ft²/ft
A_o	External tube surface, ft²/ft
A_R	Radiant surface area, ft²
A_s	Total surface of extended-surface tube, ft²/ft
c_p	Specific heat, Btu/lb/°F
d_i	Tube inside dia., in.
d_o	Tube outside dia., in.
D	Tube outside dia., ft
D'	Stack dia., ft
e	Thermal efficiency
E	Fin efficiency
f	Fanning friction factor
F_i	In-tube fouling resistance, [Btu/h/(ft²)(°F)]⁻¹
F_o	External fouling resistance, [Btu/h/(ft²)(°F)]⁻¹
g	Flue-gas mass velocity, lb/s/ft²
g'	Fluid mass velocity, lb/s/ft²
G	Flue-gas mass velocity, lb/h/ft²
h_c	Convection film coefficient, Btu/h/(ft²)(°F)
h_i	In-tube fluid film coefficient, Btu/h/(ft²)(°F)
h_o	Total convection heat-transfer coefficient, Btu/h/(ft²)(°F)
h_{rg}	Gas radiation coefficient, Btu/h/(ft²)(°F)
h_w	Tube-wall coefficient, Btu/h/(ft²)(°F)
H_A	Heat available in flue gas, Btu/lb of fuel
H_D	Net heating value of fuel, Btu/lb
k	Thermal conductivity, Btu/h/(ft²)(°F)/ft
K_c	Thermal conductivity of coke/scale layer, Btu/h/(ft²)(°F)/in.
K_m	Thermal conductivity of tube wall, Btu/h/(ft²)(°F)/in.
L_e	Hydraulic length, ft
LMTD	Log-mean temperature difference, °F
L_s	Stack height, ft
Δp	Pressure drop, psi
p'	Ambient pressure, psia
P	Design pressure, psig
q	Transfer rate, Btu/h/ft² of outside tube surface
q_i	Transfer rate, Btu/h/ft² of inside tube surface
Q	Heat absorbed, Btu/h
Q_c	Convection-section heat absorption, Btu/h
Q_F	Heat fired, Btu/h
Q_R	Radiant-section heat absorption, Btu/h
R_i	In-tube film resistance [Btu/h/(ft²)(°F)]⁻¹
R_o	External film resistance [Btu/h/(ft²)(°F)]⁻¹
R_w	Tube-wall resistance [Btu/h/(ft²)(°F)]⁻¹
R_t	Total resistance [Btu/h/(ft²)(°F)]⁻¹
S	Design stress, psi
t_c	Thickness of coke/scale layer, in.
t_m	Thickness of tube wall, in.
T_a	Ambient temperature, °R
T_B	Bulk fluid temperature, °F
T_g	Flue-gas temperature, °F
T_{ga}	Flue-gas temperature, °R
T_m	Tube metal temperature, °F
u	Bulk viscosity, lb/ft/h
u_w	Viscosity at wall temperature, lb/ft/h
U	Overall coefficient, Btu/(h)(ft²)/°F
V	Specific volume, ft³/lb
W	Flowrate, lb/h

percentages of carbon, hydrogen, oxygen and sulfur, according to the following expression:

$$\text{lb air/lb fuel} = (0.1159)\%C + (0.3475)\%H + (0.0435)\%S - (0.0435)\%O$$

For a fuel oil consisting by weight of carbon 84.6%, hydrogen 10.9%, sulfur 1.6%, and oxygen 2.9%, the stoichiometric air requirement is:

$$(0.1159)(84.6) + (0.3475)(10.9) + (0.0435)(1.6) - (0.0435)(2.9) = 13.54 \text{ lb air/lb fuel}$$

Combustion products

For the two types of fuel, the determination of flue-gas quantity per unit quantity of fuel is performed simply by assuming 20% excess air for the fuel gas and 25% excess air for the fuel oil.

For the gas:

Total air = (1.20)(15.9) = 19.07 lb air/lb fuel

Flue gas = 19.07 + 1 = 20.07 lb flue gas/lb fuel

For the oil:

Total air = (1.25)(13.54) = 16.93 lb air/lb fuel

Flue gas = 16.93 + 1 = 17.93 lb flue gas/lb fuel

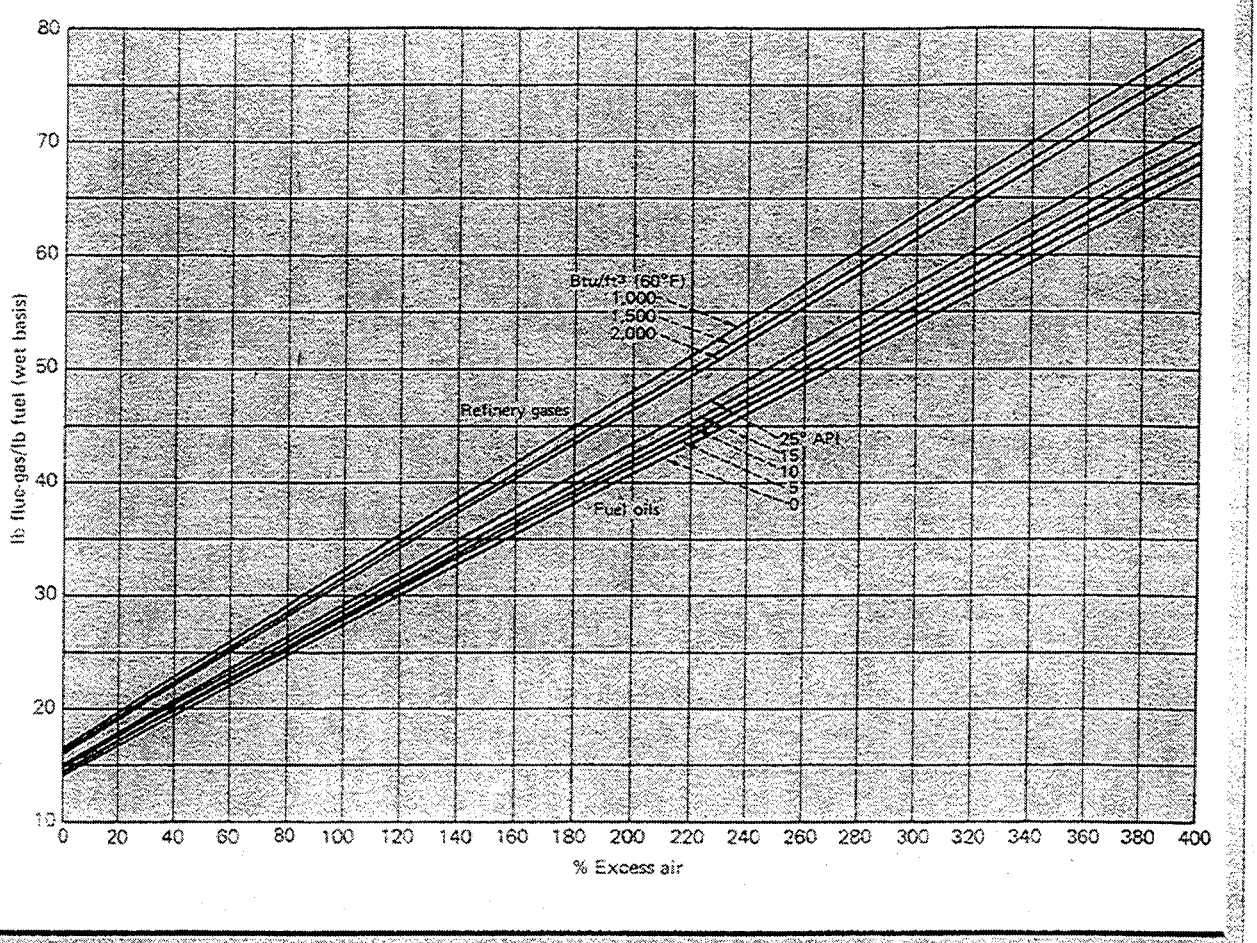
A generalized relationship that gives the weight ratio of flue gas to fuel is presented in Fig. 2 for several representative fuel gases and fuel oils.

Heating-coil arrangement

The normal flow of process fluid through a fired heater starts at the inlet to the convection section. The stream moves through the convection section countercurrent to the flow of flue gases, then into the shock bank. After leaving the shock bank, it passes into the radiant section, where the major portion of the heat is absorbed.

As described in Part II, adjacent tubes are connected by means of 180-deg return bends or plug-type headers. Each bank of consecutive tubes in which the fluid travels from entry until exit is known as a "pass," or parallel stream. (This definition differs from that applied to unfired heat-transfer equipment.) In a two-pass heater, the fluid is distributed into two streams at the inlet, with each stream flowing separately through its respective tube coil and then recombining after exiting from the heater. A single-pass or "series-flow" heater with 40 tubes, for example, would be equivalent in area to a two-pass heater having two coils of 20 tubes each.

The primary constraint affecting the selection of the number of passes and tube size is the allowable pressure



Generalized curves give quantity of flue gases released by liquid and gaseous fuels Fig. 2

drop. The fewer the number of passes, the less the potential for flow maldistribution.

Each heater application must be evaluated individually with regard to the number of passes and tube size. Such evaluation must examine not only the cost of the heater but also the cost of external distribution and collection manifolds, pass control valves, and other ancillaries. For normal heating applications, a tube coil consisting of 4-in. IPS (iron pipe size) tubes often represents the lowest heater investment (excluding externals). As tube diameter increases or decreases from this size, the heater cost tends to become progressively more expensive.

Thermal efficiency

For each fuel, profiles can be developed at constant values of excess air to express the heat extracted from the products of combustion as a function of flue-gas temperature. The percentage of the heat extracted from the flue gas will range from 0% at the flame temperature, to 100% at the datum flue-gas temperature of 60°F. This heat includes the heat absorbed by the charge stock plus the heat lost from the heater casing. At any flue-gas temperature, the differential between the percent heat extracted and 100% constitutes the percent heat loss up the stack.

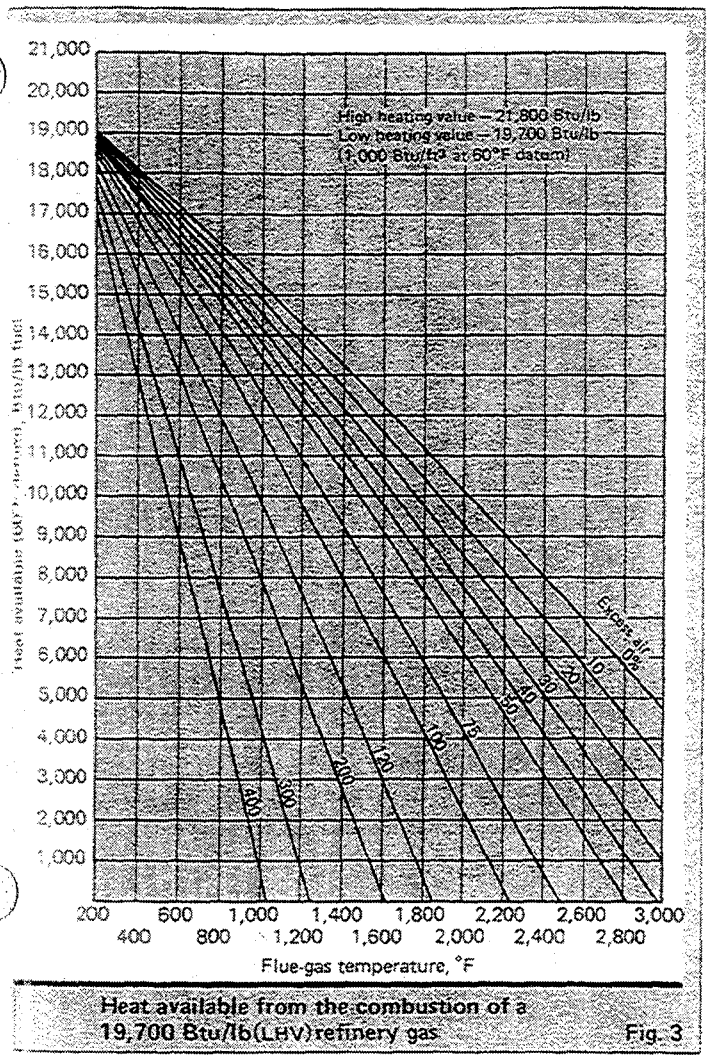
Fig. 3 and 4 illustrate for two typical fuels (a 19,700 Btu/lb refinery gas and a 15° API fuel oil), profiles of heat available in the flue gas as a function of temperature and excess air. Data for other fuels, identified by their net heating values (Btu/lb), are given in Ref. [9].

Assuming that the fuel and combustion air are supplied at the datum temperature of 60°F, for a given combination of flue-gas temperature and excess air, the percent heat extracted from the flue gas is:

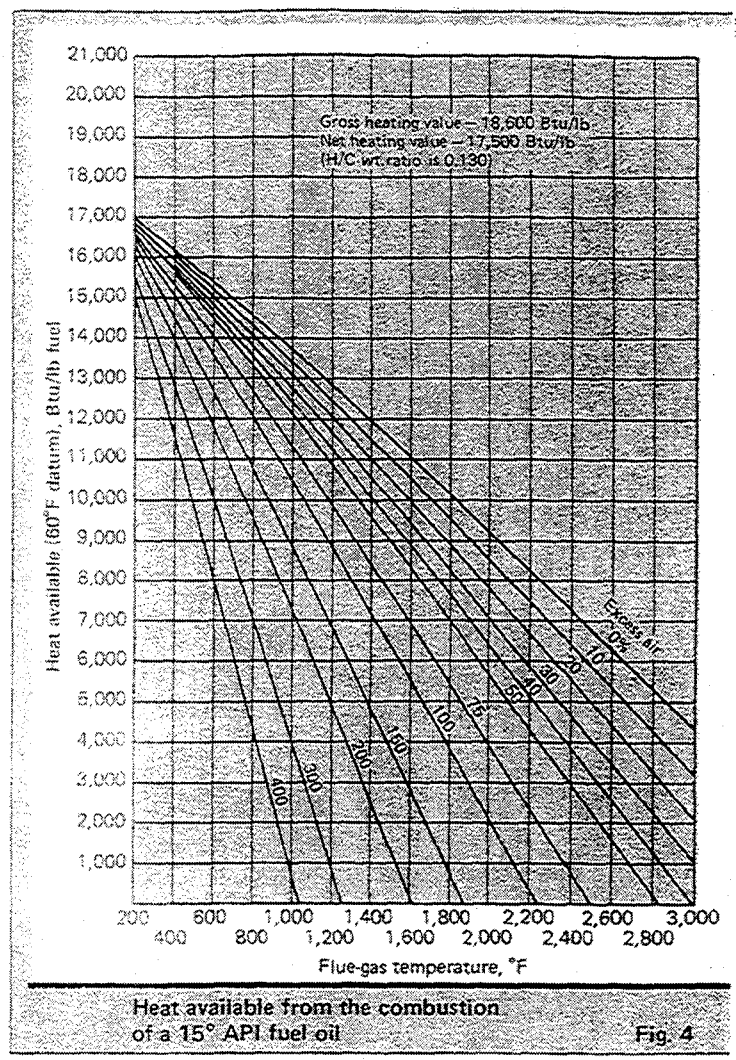
$$\% \text{ Heat extracted} = \frac{(100)(\text{Heat available, Btu/lb fuel at flue gas temp.})}{(\text{Net heating value of fuel, Btu/lb})}$$

$$= (100)H_A/H_F$$

The heat transmitted from the heater casing to the atmosphere via radiation and convection (the so-called "radiation loss") is generally assessed at 1½ to 2% of the heat fired for conventional installations, and at 2 to 2½% for heaters with extensive hot-duct runs and/or air preheaters.



Heat available from the combustion of a 19,700 Btu/lb (LHV) refinery gas — Fig. 3



Heat available from the combustion of a 15° API fuel oil — Fig. 4

The actual or calculated thermal efficiency of the heater is computed as the percent heat extracted less the percent radiation loss. The heat fired is then determined from the relationship:

$$\text{Thermal efficiency} = e = \frac{\text{Heat absorbed, Btu/h}}{\text{Heat fired, Btu/h}}(100) = \frac{Q}{Q_F}(100)$$

The quantity of fuel consumed is obtained as follows:

$$\text{Fuel consumed, lb/h} = \frac{\text{Heat fired, Btu/h}}{\text{Fuel heating value, Btu/lb}} = \frac{Q_F}{H_F}$$

The flue-gas mass flowrate is approximated by multiplying the fuel consumption rate by the appropriate weight ratio of flue gas to fuel, obtained from the curves in Fig. 2.

The foregoing relationships define the interdependence of excess air, flue-gas temperature and thermal efficiency. In the design of heaters having convection sections, thermal efficiency is determined by selecting the flue-gas temperature. Specification of this temperature is normally contingent upon that of the process fluid at the inlet. A typical design basis would assume a temperature approach of about 150°F between flue-gas temperature and inlet fluid temperature.

A temperature differential of this magnitude represents a reasonable balance between thermal efficiency and capital investment. However, many criteria affect the temperature approach for a particular application. These include fuel value, project payback time, minimum stack-height requirement, type and material of extended surface, auxiliaries such as sootblowers, etc. In the case of all-radiant heaters without convection sections, the designer does not have the degree of freedom to select the design flue-gas temperature. For these heaters, the temperature is equal to the residual radiant-section gas temperature.

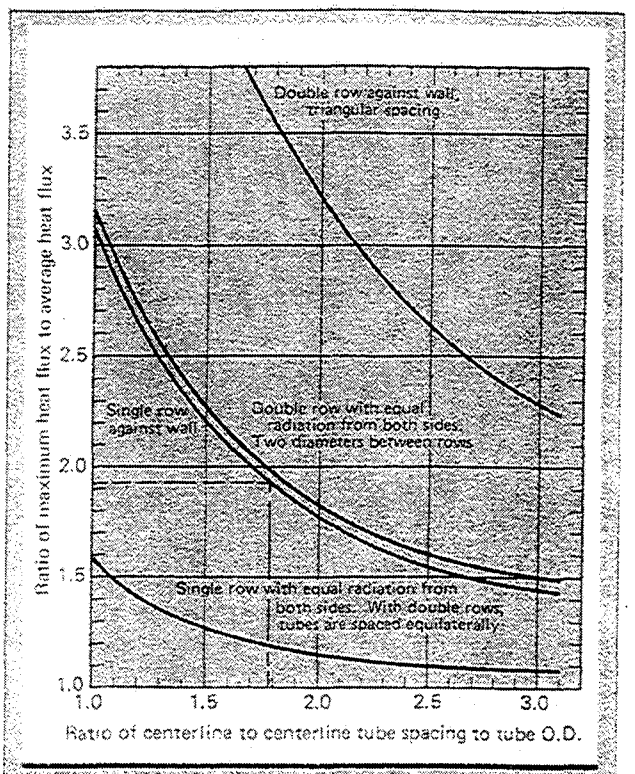
Radiant section

In order to evaluate the split in heat absorption between radiant and convection sections, it is necessary to determine the radiant efficiency—the fraction of heat liberated that is absorbed by the heat-transfer surface in the combustion chamber. For a given fuel at a given value of excess air, the radiant efficiency can be shown to be a function of the residual radiant-section gas

Typical heat-transfer rates across the radiant sections of heaters in various services — Table II

Service	Average radiant rate* Btu/h/ft^2 (based on O.D.)
Atmospheric crude heaters	10,000-14,000
Reduced-crude vacuum heaters	8,000-10,000
Reboilers	10,000-12,000
Circulating-oil heaters	8,000-11,000
Catalytic-reformer charge and reheat heaters	7,500-12,000
Delayed coker heaters	10,000-11,000
Visbreaker heaters—heating section	9,000-10,000
Visbreaker heaters—soaking section	6,000-7,000
Propane deasphalting heaters	8,000-9,000
Lube vacuum heaters	7,500-8,500
Hydrotreater and hydrocracker charge heaters	10,000
Catalytic-cracker feed heaters	10,000-11,000
Steam superheaters	9,000-13,000
Natural-gasoline plant heaters	10,000-12,000

*For tubes spaced at twice the nominal diameter, fired on one side and backed by refractory.



Distribution of radiant heat-transfer rate around tube depends upon coil arrangement and firing mode — Fig. 5

temperature [3]. In turn, for a particular heater configuration, this temperature can be expressed graphically as a function of excess air, tube metal temperature, and radiant transfer rate [4].

The selection of the average radiant transfer rate is, essentially, the first step in the process design of a fired heater. By definition, the average radiant rate represents the heat transferred to the charge stock in the radiant section, divided by the total radiant-section heat-transfer surface, based on the tube O.D. The higher the design radiant rate, the less the amount of heat-transfer surface, the smaller the heater, and the lower the investment cost.

Unduly high radiant rates, however, result in higher maintenance costs. Because the refractories and tube supports are exposed to higher temperatures, they have shorter service lives. Furthermore, high tube-wall temperatures reduce tube life and raise the potential for coke deposition and product degradation. Actual radiant transfer rates will reflect the experience of both the user and the designer. A tabulation of typical rates for various commercial heating services is shown in Table II.

Distribution of the radiant heat-transfer rate around a tube is not uniform. In fact, the rate varies substantially around the tube circumference, depending upon the ratio of tube spacing to tube diameter, and upon the firing mode—i.e., whether the tubes are fired from one side or from both sides. If the maximum radiant rate is assumed to occur at the front 60 deg of the fired tube face, the ratio of maximum radiant rate to average radiant rate can be obtained from Fig. 5 for various tube coil configurations and firing modes [5].

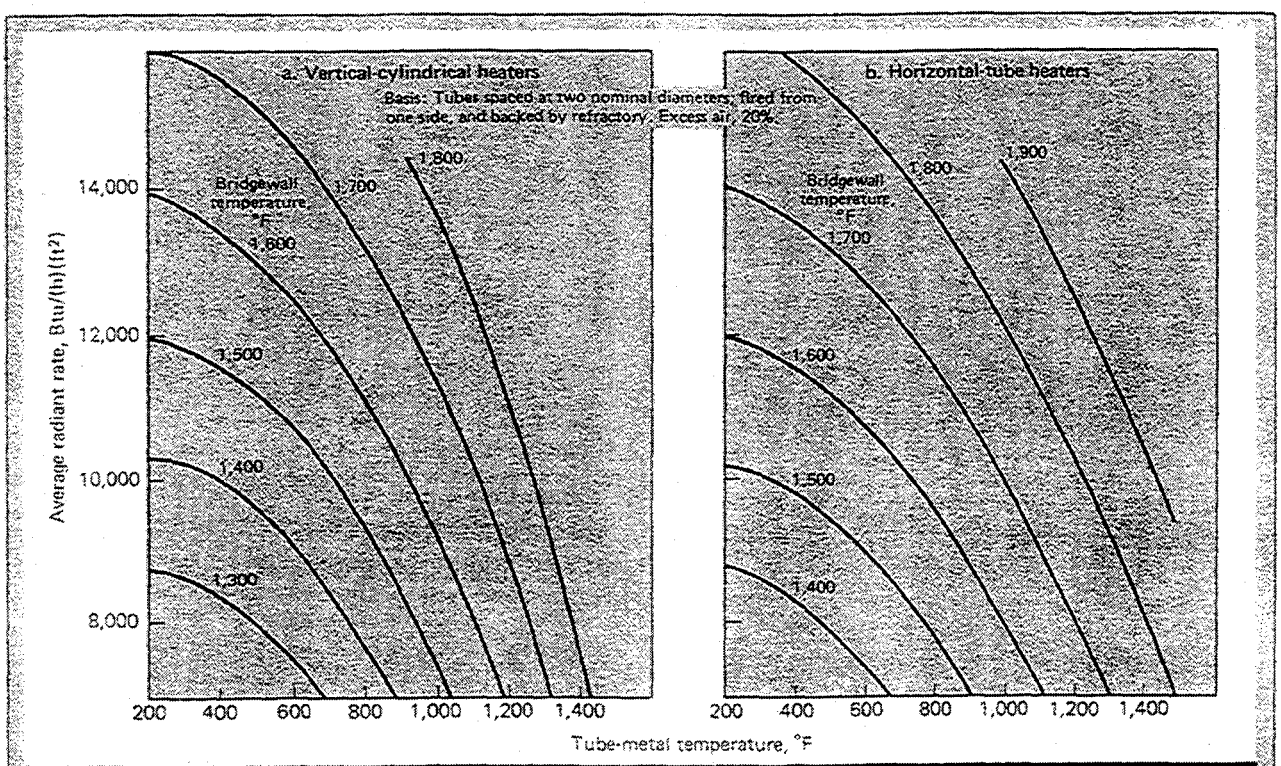
As noted in Part II, the tube metal temperature is always hotter than the bulk fluid temperature at any given location. The magnitude of the metal temperature depends on the fluid temperature, the radiant transfer rate, the tubeside film coefficient, the thermal conductivity of the tube material, and the thermal resistance of coke or scale deposits. The metal temperature can be expressed as the sum of the bulk fluid temperature plus the temperature differentials across the fluid film, the coke/scale layer, and the tube wall:

$$T_m = T_B + \frac{q_t}{h_i} + \frac{q_t l_c}{K_c} + \frac{q_t l_m}{K_m} \qquad (2)$$

For vertical-cylindrical heaters and horizontal-tube heaters, Figs. 6a and b estimate the residual radiant-section gas temperature—the so-called bridgewall temperature (SWT)—as a function of the radiant transfer rate and the average metal temperature. The charts assume that the tube coil is fired from one side, and that the tubes are spaced at twice their nominal diameter. As an expedient, the metal temperature can be reasonably approximated for most conventional heating applications by adding 75°F to the average bulk-fluid temperature in the radiant section.

Once the bridgewall temperature and the flue-gas temperature (FOT) are established, the split in duty between radiant and convection sections can be obtained as a function of the heat available in the products of combustion at these temperatures. The split is governed by the relationship that the ratio of the heat available at the bridgewall temperature to that at the flue-gas temperature is equal to the ratio of the radiant duty to the total duty:

$$\frac{H_{A,SWT}}{H_{A,FOT}} = \frac{Q_R}{Q} \qquad (3)$$



Bridgewall temperature enters into calculation of duty-split between radiant and convection sections Fig. 6

The radiant heat-transfer surface area is obtained as the quotient of the radiant duty divided by the average radiant rate:

$$A_R = \frac{Q_R}{q} \tag{4}$$

Convection section

All combustion should be completed before the flue gas reaches the convection section. Although convective heat transfer is the primary mode of heat transmission in this section, radiative effects also contribute.

The first two or three rows of tubes adjacent to the point at which the flue gas enters the convection section are known as the shield bank. Because these tubes are generally oriented at a staggered triangular pitch normal to the flue-gas flow, they will usually screen and absorb most of the residual radiative component. When the shield bank can "see" the firebox, as in most modern installations, industry practice is to consider the surface area of one row of shield tubes to be radiant transfer area. Consequently, this surface is usually included in the radiant-section surface requirement.

To estimate a film coefficient based on pure convection for flue gas flowing normal to a bank of bare tubes, a method was developed by Monrad [6] and was subsequently revised to the following form [7]:

$$h_c = \frac{2.14\, g^{0.6} T_{ga}^{\ 0.28}}{d_o^{\ 0.4}} \tag{5}$$

where g = flue-gas mass velocity, lb/s/ft² at minimum cross-section; T_{ga} = average flue-gas temperature, °R; and d_o = tube outside diameter, in.

This equation does not take into account radiation from the hot gases flowing across the tubes, or re-radiation from the walls of the convection section. As an approximation, the radiation coefficient of the hot gas may be obtained from the following equation [8]:

$$h_{rg} = 0.0025 T_g - 0.5 \tag{6}$$

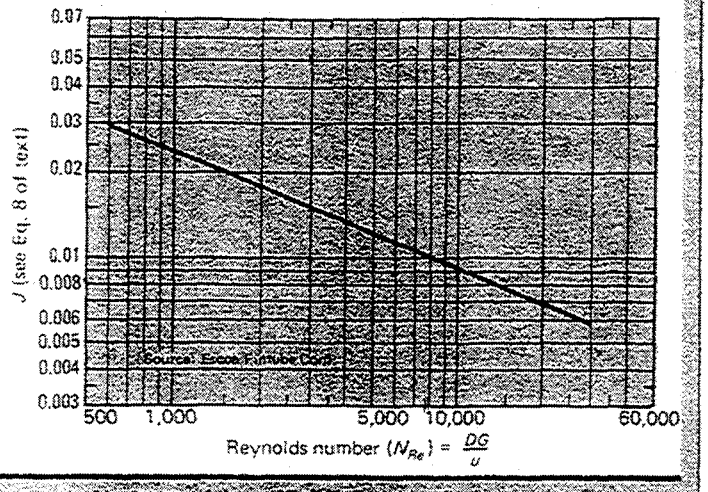
where T_g = average flue-gas temperature, °F.

Re-radiation from the walls of the convection section usually ranges from 6 to 15% of the sum of the pure-convection and the hot-gas-radiation coefficients. A value of 10% represents a typical average. Based on this value, the total heat-transfer coefficient for the bare-tube convection section can be computed as:

$$h_c = (1.1)(h_c + h_{rg}) \tag{7}$$

Convection-section surface-area requirements are controlled by the film resistance on the flue-gas side. In order to increase transfer rates per unit length of tubing, convection sections in modern heater designs are almost always equipped with extended surface. Various types of extended surface employed in heater convection sections have been already described. With all of these extended-surface types, radiant transfer to the convection tubes is so small that it can be neglected. For example, the heat-transfer coefficient of a serrated-fin surface, h_o, can be obtained from Fig. 7 as a function of the Reynolds number:

$$h_o = Jc_pG \Big/ \left(\frac{uc_p}{k}\right)^{2/3} \tag{8}$$



Heat-transfer coefficients on the flue-gas side of serrated fins — Fig. 7

Fluid mass velocities govern choice of tube size and number of passes — Table III

Service	Mass velocity, lb/s/ft^2
Atmospheric crude heaters	175-250
Reduced-crude vacuum heaters (outlet tube)	60-100
Reboilers	150-250
Circulating-oil heaters	350-450
Catalytic-reformer charge and reheat heaters	45-70
Delayed coker heaters	350-450
Hydrotreater and hydrocracker charge heaters	150-200
Steam superheaters	30-75
Steam generators (forced circulation)	100-150
Catalytic-cracker feed heaters	300-400

where G = flue-gas mass velocity, lb/h/ft^2; c_p = flue-gas specific heat, Btu/lb/°F; μ = flue-gas viscosity, lb/h/ft; and k = flue-gas thermal conductivity, Btu/h/(ft^2)(°F)/ft.

The effective outside heat-transfer coefficient is calculated from:

$$\text{Effective } h_o = h_o(EA_f + A_o)/A_t \qquad (9)$$

The fin efficiency, E, takes into account the variability of fin effectiveness as a function of fin configuration, the thermal conductivity of the extended surface, and the convection film coefficient. For serrated fins, values of E may be approximated from Fig. 8.

The overall coefficient, U, is obtained from several relationships:

$$R_o = \frac{1}{\text{Effective } h_o} + F_o \qquad (10)$$

$$R_w = \frac{A_t}{h_w A_i} \qquad (11)$$

$$R_i = \frac{A_t}{A_i}\left(\frac{1}{h_i} + F_i\right) \qquad (12)$$

$$R_t = R_o + R_w + R_i \qquad (13)$$

$$U = \frac{1}{R_t} \qquad (14)$$

The convection surface requirement is calculated as:

$$A_c = \frac{Q_c}{\text{LMTD}(U)} \qquad (15)$$

The highest convection-transfer rate normally occurs at the lowest extended-surface row. In many designs, the density of the extended surface is reduced for the lowest row or two in order to keep the convection rate from becoming excessive. A reasonable estimated value for the maximum convection heat-transfer rate (expressed on an equivalent bare-tube basis) for conventional heating applications is twice the average radiant-transfer rate.

The maximum tube-metal temperature in the convection section is estimated in the same manner as for the radiant section, but with bulk fluid temperature, in-tube film coefficient, and transfer rate appropriate for the convection zone.

Fluid pressure drop

The optimum combination of tube size and number of passes is contingent on an accurate evaluation of the fluid pressure drop. Historically, for heaters processing heavy oils, an important criterion for choosing the tube size and the passing arrangement has been the cold-oil velocity. However, with the advent of numerous all-vapor applications, a more meaningful basis has proved to be the fluid mass velocity. Table III presents a listing of typical fluid mass velocities for the more frequently encountered heating applications.

The pressure drop for a single-phase fluid, either all vapor or all liquid, can be predicted with reasonable confidence using well-established hydraulic principles. The following relation allows pressure-drop computations to be made with an accuracy sufficient for most engineering purposes:

$$\Delta p = \frac{(0.00517) f (g')^2 (V)(L_e)}{d_i} \qquad (16)$$

where Δp = pressure drop, lb/in^2; f = Fanning friction factor; g' = fluid mass velocity, lb/s/ft^2; V = average specific volume, ft^3/lb; d_i = tube inside diameter, in.; and L_e = hydraulic length, ft.

The total hydraulic length, L_e, is based on the sum of the actual tube lengths plus the equivalent lengths of return fittings and elbows. These equivalent lengths can be approximated for various fittings as a multiple of the tube inside diameter. Typical equivalent lengths are 50 diameters for 180-deg return bends, 30 diameters for 90-deg elbows, and 100 diameters for plug-type headers.

The pressure drop for a mixed-phase flow regime can be approximated using the foregoing relation—albeit with a lesser degree of confidence than for single-phase computations. In the case of mixed-phase flow, the

specific volume is very sensitive to the quantity of vapor present at a given location. Consequently, a more accurate estimate of overall pressure drop can be obtained by raising the number of incremental pressure-drop determinations over the length of the coil.

In most vaporizing hydrocarbon services, vapor formation is not linear along the tube length; instead, it increases near the heater outlet. If an arithmetic average of the specific volumes at the inlet and outlet of a mixed-phase zone is used, the resultant pressure-drop value will be overly conservative. To compensate for this, it is suggested that the log mean average of the inlet and outlet specific volumes be used for each mixed-phase zone under consideration.

In view of the vast quantity of literature, both theoretical and empirical, concerning mixed-phase pressure drop, it would seem simplistic to apply a single friction factor to a mixed-phase flow system. Yet for many years, a single friction factor has been used with great success to design countless mixed-phase fired heaters. For example, in conventional hydrocarbon vapor-liquid systems, a factor of 0.0045 has proven to be of sufficient engineering accuracy for pressure-drop determinations.

Because it is unusual for a charge stock to enter a fired heater at its bubble point, vaporization normally begins part way through the tube coil. Under this condition, individual pressure-drop determinations should be made for an all-liquid, single-phase regime, and for a separate mixed-phase regime. The point of initial vaporization is arrived at via trial-and-error, mixed-phase pressure-drop calculations.

In heaters having high percentages of vaporization, it is not unusual for a temperature peaking condition to occur. As the mixed-phase fluid flows through the coil, it undergoes a substantial drop in pressure per unit length of flow. This causes the rate of vaporization to outstrip the rate at which latent and sensible heat can be supplied by the products of combustion. Thus the bulk fluid temperature drops even though the enthalpy increases. It may well be that at some intermediate location along the coil, the bulk fluid temperature will rise above the outlet temperature.

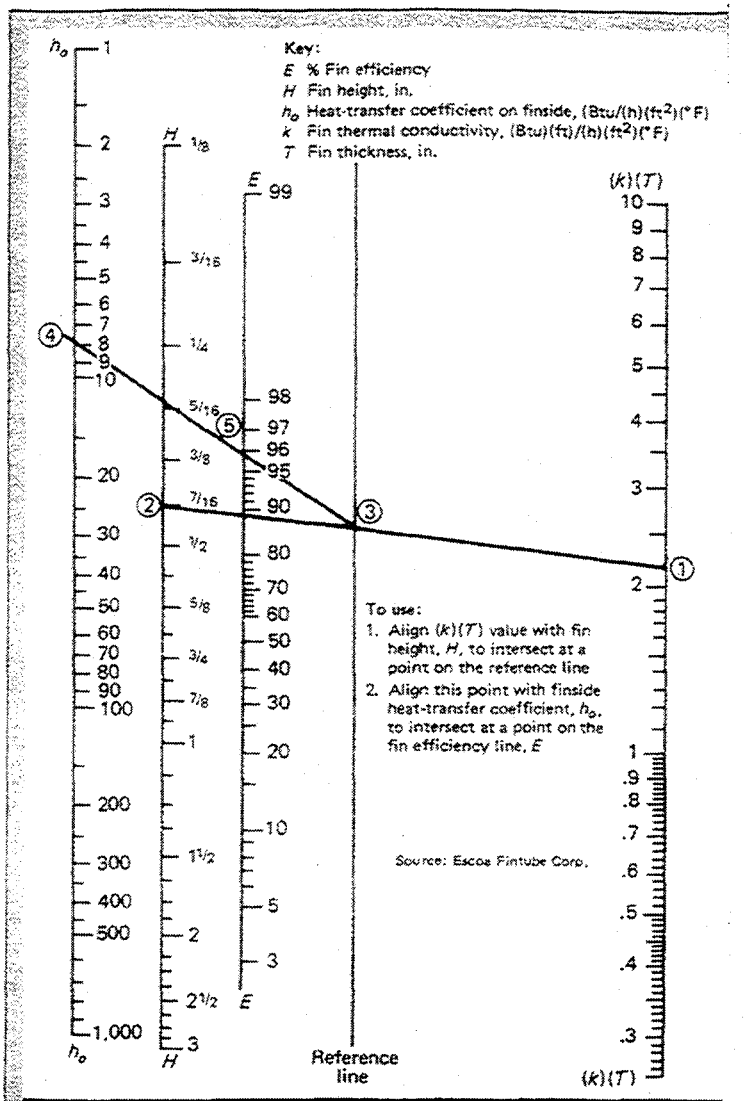
Although this temperature peaking effect is a perfectly normal condition, it is incumbent upon the designer to properly assess its magnitude and the consequences, particularly in high-temperature services.

Tube-wall thickness

As mentioned previously, there are two design criteria that have been applied to heater tubes. At lower temperatures in the elastic range, the design stress is based on the yield strength. At higher temperatures in the creep-rupture range, the design stress is based on the rupture strength. For both conditions, the recommended tube-wall thickness is determined from the mean diameter formula:

$$t_m = \frac{Pd_o}{2S + P} \qquad (17)$$

In the elastic range, the corrosion allowance is added to the wall thickness as determined above. The design pressure, P, corresponds to the maximum pressure excursion during an upset condition. The design stress, S, is based on two thirds of the yield strength for ferritic steels or 90% of the yield strength for austenitic steels.



Fin efficiency varies with design and thermal conductivity of fin, and with convection film coefficient. Fig. 8

In the creep-rupture range, a fraction of the corrosion allowance is added to the wall thickness, as determined by Eq. (17). The design pressure, P, corresponds to the operating pressure, and the design stress, S, is based on 100% of the minimum rupture strength for a specified service life. Typical design service lives range from 60,000 to 100,000 h.

A more detailed presentation of the foregoing, as well as design stress values and a procedure for evaluating the corrosion fraction, is to be found in the 2nd ed. of the API's Recommended Practice 530, "Calculation of Heater Tube Thickness in Petroleum Refineries" [*10*].

When the fluid being heated is water or steam, the requirements of the ASME Boiler and Pressure Vessel Code normally prevail. The tube wall thickness should be calculated in accordance with Section I, using Code formulas and stresses.

Design stresses are very sensitive to temperature, par-

Sample calculation

The following sequence of calculations illustrates the principal steps in the process design of a fired heater.

Fluid	Dowtherm A
Heat absorbed, Btu/h	27,390,000
Flowrate, lb/h	550,000
Inlet temperature, °F	490
Outlet temperature, °F	580
Inlet vaporization	Nil
Outlet vaporization	Nil
Outlet pressure, psig	150
Allowable pressure drop, psi	25
Design pressure, psig	250
Fuel gas, Btu/lb (LHV)	19,700

Design basis

Vertical-cylindrical heater with horizontal-tube convection section
Excess air, 20%
Average radiant rate, 10,000 Btu/h/ft²

Efficiency

Take flue-gas temp. (FGT) = 490 + 150 = 640°F
Heat available at FGT = 16,600 Btu/lb (Fig. 3)

$$\% \text{ Heat extracted} = \frac{(100)(16,600)}{(19,700)} = 84.3\%$$

Calculated efficiency = 84.3 − 2.0 (radiative loss) = 82.3% LHV

$$\text{Heat fired} = \frac{27,390,000}{0.823} = 33,280,000 \text{ Btu/h}$$

$$\text{Fuel consumed} = \frac{33,280,000}{19,700} = 1,689.3 \text{ lb/h}$$

Flue-gas flow = (1,689.3)(19.9) = 33,620 lb/h (Fig. 2)

Radiant/convection duty split

Select tube coil having 4 passes of 4-in. IPS, Sched. 40 (4.5-in. O.D. × 0.237-in. avg. wall)

$$\text{Fluid mass vel.} = \frac{550,000}{(3,600)(4)(0.0884)} = 432 \text{ lb/s/ft}^2$$

Assume fluid temp. at radiant inlet = 520°F

$$\text{Radiant-section ave. fluid temp.} = \frac{520 + 580}{2} = 550°\text{F}$$

Take radiant-section average tube-metal temperature = 550 + 75 = 625°F
Bridgewall temp. (BWT) = 1,470°F (Fig. 6a)

$$\frac{H_{A,BWT}}{H_{A,FGT}} = \frac{11,700}{16,600} = \frac{Q_R}{27,390,000}$$

Q_R = radiant duty = 19,310,000 Btu/h
Q_c = convection duty = 8,080,000 Btu/h

$$A_R = \text{radiant surface} = \frac{19,310,000}{10,000} = 1,931 \text{ ft}^2$$

Assume 52 radiant tubes on 8-in. centers

$$\text{Tube-circle dia. (TCD)} = \frac{(52)(8)}{(12)(\pi)} = 11.03 \text{ ft}$$

Take effective tube length (ETL) of shield bank = 10 ft
With 8 tubes per row on 8-in. equilateral centers:

$$\text{Shield-bank free area} = \frac{(8)(10.0)(8.0 - 4.5)}{(12)} = 23.33 \text{ ft}^2$$

$$g = \text{flue-gas mass velocity} = \frac{33,520}{(3,600)(23.33)} = 0.400 \text{ lb/s/ft}^2$$

Consider surface equivalent to one shield row as radiant surface

$$\text{Surface of one shield row} = (8)(10.0)(1.178) = 94 \text{ ft}^2$$

$$\text{Vertical-tube radiant surface} = 1,931 - 94 = 1,837 \text{ ft}^2$$

$$\text{Vertical-tube ETL} = \frac{1,837}{(52)(1.178)} = 30 \text{ ft}$$

$$\text{Ratio: } \frac{\text{ETL}}{\text{TCD}} = \frac{30.0}{11.03} = 2.72$$

Shield bank

Shield-bank heat absorption is approximated via a trial-and-error process
Use three-row shield bank
Take average fluid temp. in shield bank = 515°F
Fluid temperature increase in shield bank is small and can be neglected
Assume flue-gas temp. drop across shield = 210°F

LMTD calculation:

1,470	1,260
515	515
955	745

$$\text{LMTD} = 846°\text{F}$$

$$h_c = \frac{(2.14)(0.400)^{0.6}(1,825)^{0.28}}{(4.5)^{0.4}} = 5.54$$

$$h_{rg} = (0.0025)(1,325) - 0.5 = 2.91$$
$$h_o = (1.1)(5.54 + 2.91) = 9.29$$

Assume clean tube design and no internal or external fouling resistances

$$h_i = (0.027)\frac{k}{D}\left[\frac{(DG)}{(u)}\right]^{0.8}\left[\frac{(c_p u)}{(k)}\right]^{0.333}\left[\frac{(u)}{(u_w)}\right]^{0.14}$$

In view of the small variation in viscosity between bulk and wall temperatures, the value of $(u/u_w)^{0.14}$ may be taken to be 1.0.

$$h_i = (0.027)\frac{(0.064)}{(0.375)}\left[\frac{(0.375)(1,555,200)}{(0.605)}\right]^{0.8} \times \left[\frac{(0.543)(0.605)}{(0.064)}\right]^{0.333}(1.0) = 487$$

$$R_i = \frac{A_o}{h_i A_i} = \frac{1.178}{(487)(1.054)} = 0.002295$$

$$h_w = \frac{K_m}{t_m} = \frac{324}{0.237} = 1,367$$

$$R_w = \frac{A_o}{h_w A_i} = \frac{1.178}{(1,367)(1.054)} = 0.000818$$

$$R_o = \frac{1}{h_o} = \frac{1}{9.29} = 0.10764$$

$R_c = 0.11076$; $U = 9.03$
Shield-bank transfer rate = (U)(LMTD) = (9.03)(846) = 7,639 Btu/h/ft²
Three-row shield-bank surface = (3)(8)(10.0)(1.178) = 283 ft²
Shield-bank heat absorption = (283)(7,639) = 2,162,000 Btu/h

Check of flue-gas temperature above shield bank:

$$\frac{H_{A,BWT}}{H_{A,shield}} = \frac{11,700}{H_{A,shield}} = \frac{19,310,000}{21,472,000}$$

$H_{A,shield}$ = 13,010 Btu/lb of fuel

Flue-gas temperature above shield bank = 1,260°F (as assumed)

Convection bank

Fin-bank heat absorption = 8.08 MM − 2.162 MM = 5,918,000 Btu/h

Fluid temperature leaving fin bank = 510°F

LMTD calculation:

1,260	640
510	490
750	150

LMTD = 373°F

Try 3 fins/in., with each fin ¾ in. high × 0.05 in. thick (A_t = 7.33 ft²/ft)

Fin-bank free area = $(8)(10.0) \times \left[\frac{(8.0 - 4.5)}{(12)} - \frac{(2)(0.05)(0.75)(3)}{(12)}\right]$ = 21.83 ft²

$$G = \frac{33,620}{21.83} = 1,540 \text{ lb/h/ft}^2$$

$$\text{Reynolds number} = \frac{(4.5)(1,540)}{(12)(0.084)} = 6,875$$

J = 0.011 (from Fig. 7)

$$h_o = \frac{(0.011)(0.28)(1,540)}{\left[\frac{(0.084)(0.28)}{(0.030)}\right]^{2/3}} = 5.58$$

Fin efficiency, E, = 84% (Fig. 8)

$$\text{Effective } h_o = \frac{(5.58)}{(7.33)}[(0.84)(6.152) + 1.178] = 4.83$$

$$R_t = \frac{(7.33)}{(487)(1.054)} = 0.01428$$

$$R_w = \frac{(7.33)}{(1,367)(1.054)} = 0.00509$$

$$R_o = \frac{1}{4.83} = 0.20701$$

R_i = 0.22638

U = 4.42

A_c = convection surface $= \frac{5,918,000}{(373)(4.42)} = 3,590$ ft²

Surface area per convection row = (8)(10.0)(7.33) = 586 ft²

Number of finned rows $= \frac{3,590}{586} = 6.1$ (use 6)

Convection surface = (6)(586) = 3,516 ft²

Pressure drop

$$\text{Reynolds number} = \frac{(4.026)(1,555,200)}{(12)(0.581)} = 898,000$$

Fanning friction factor = 0.0038

$$L_{e,\text{convection + shield}} = (2)(13.0) + (16)(11.5) + \frac{(17)(50)(4.026)}{(12)} = 495 \text{ ft}$$

$$L_{e,\text{crossover}} = 20 + \frac{(3)(30)(4.026)}{(12)} = 50 \text{ ft}$$

$$L_{e,\text{radiant}} = (2)(33.0) + (11)(29.0) + \frac{(12)(50)(4.026)}{(12)} = 586 \text{ ft}$$

Total L_e = 1,131 ft

Average specific volume = 0.0194 ft³/lb

$$\Delta p = \frac{(0.00517)(0.0038)(432)^2(0.0194)(1,131)}{(4.026)} = 20.0 \text{ psi}$$

(vs. 25 psi allowed)

Tube-wall thickness

Average radiant rate = 10,000 Btu/h/ft²

Factor (Fig. 5) max. to ave. flux ratio = 1.93

Factor for local flux variation = 1.25 (assumed)

Factor for conductive/convective effects = 0.85 (assumed)

Maximum local radiant rate = (10,000)(1.93)(1.25)(0.85) = 20,500 Btu/h/ft²

At outlet fluid temperature, h_i = 522

$$T_m = 580 + \frac{(20,500)(4.5)}{(522)(4.026)} + \frac{(20,500)(4.5)(0.237)}{(315)(4.263)} = 640°\text{F}$$

To allow for fouling effects and to provide safety margin, use design temperature = 750°F

With carbon steel tubes, assume 1/16-in. corrosion allowance

$$t_m = \frac{(250)(4.5)}{(2)(15,500) + 250} + 0.063 = 0.099\text{-in min. wall}$$

As the least wall thickness, use Schedule 40 (0.237 in. average wall)

Stack design

Size stack for mass velocity of 0.8 lb/s/ft² at 125% of design gas flow

$$\text{Cross-sectional area} = \frac{(1.25)(33,620)}{(3,600)(0.8)} = 14.59 \text{ ft}^2$$

Stack diameter, D' = 4 ft 4 in.

Assume average gas temperature in stack = 640 − 75 = 565°F.

Assume stack-exit gas temperature = 490°F

Draft under arch = 0.050 in.

Shield-bank loss = (3)(0.2)(0.0030)(1.25 × 0.400)²(46.3) = 0.021 in.

Fin-bank loss = (6)(1.0)(0.0030)(1.25 × 0.428)²(35.6) = 0.183 in.

Stack-entrance loss = (0.5)(0.0030)(0.8)²(27.8) = 0.027 in.

Damper loss = (1.5)(0.0030)(0.8)²(27.8) = 0.080 in.

Stack-exit loss = (1.0)(0.0030)(0.8)²(24.0) = 0.046 in.

Subtotal = 0.407 in.

Convection-section draft gain = $(0.52)(8.5)(14.69)\left[\frac{1}{540} - \frac{1}{1,515}\right]$ = 0.077

Required stack draft = 0.330 in.

Stack draft gain/ft = $(0.52)(1.0)(14.69)\left[\frac{1}{540} - \frac{1}{1,025}\right]$ = 0.00669 in.

Stack frictional loss/ft = $\frac{(0.8)^2(1,025)}{(211,000)(4.33)}$ = 0.00072 in.

Net stack effect/ft = 0.00597 in.

Stack height required $= \frac{0.330}{0.00597} = 55$ ft 3 in.

ticularly in the higher regions. Consequently, the thickness calculation requires an accurate determination of the tube-wall temperature. It should be noted that the curves in Fig. 5, which show the ratio of the maximum radiant-transfer rate to the average rate do *not* consider convection heat transfer to the radiant tubes, circumferential heat transfer by conduction along the tube wall, or variations in the transfer rate at different zones of the combustion chamber.

The highest transfer rate in horizontal-tube cabin heaters generally occurs at the wall, 5 to 10 ft above the floor, or at the shield bank, where convection effects are most pronounced. The transfer rate in the lower third of cylindrical heaters may approach 1.1 to 1.5 times the average radiant-transfer rate, depending on the ratio of effective tube length to tube-circle diameter. The magnitude of this local variation is compensated somewhat by the combined conductive/convective effect, which tends to reduce the disparity between maximum and average transfer rates.

Stack design

The main functions of a stack are to induce the flow of combustion air into the heater, and to produce draft sufficient to overcome all obstructions to the flow of flue gas while maintaining a negative pressure system throughout. Never should a heater be operated with greater than atmospheric pressure at any point within the structure. Positive pressure within a heater setting creates a driving force for the outward movement of hot gases, which can lead to serious overheating and corrosion of the steel structure.

The draft produced by a column of hot flue gas depends on the density difference between the hot gas and the ambient air. The draft, in inches of water, can be expressed as:

$$\text{Draft} = (0.52)(L_s)(p')\left(\frac{1}{T_a} - \frac{1}{T_{ga}}\right) \qquad (18)$$

where L_s = stack height, ft; p' = atmospheric pressure, psia; T_a = ambient temperature, °R; and T_{ga} = flue-gas temperature, °R.

Because of heat loss through the stack casing, the flue-gas temperature at the top of the stack is substantially lower than at the inlet at the stack base. The magnitude of the differential depends on several factors, including the stack dimensions and the effectiveness of the stack insulation.

For most applications, the average flue-gas temperature in the stack may be conservatively estimated at 75°F less than the inlet gas temperature.

The relationship between stack diameter and stack height is also affected by cost optimization. Assuming that there are no requirements dictating a minimum stack height, a reasonable basis for selecting the diameter would result in a flue-gas mass velocity in the range of 0.75 to 1.0 lb/s/ft².

As noted above, a major objective in stack design is to ensure negative pressure throughout the heater. The most critical location in this respect occurs where the flue gas enters the convection section. Positive pressure manifests itself first in this zone.

It is recommended that the stack design be based on a negative pressure of 0.05 in. H_2O at this entry point. The use of a design pressure less negative is not realistic considering the confidence level of the design computations. On the other hand, the use of still lower pressure (i.e., more negative) would tend to create too great a driving force for the leakage of air through various apertures and seams in the structures.

The frictional loss of the flue gas flowing through the stack, in in. H_2O is:

$$\text{Loss per foot of stack height} = \frac{(g)^2(T_{ga})}{(211{,}000)(D')} \qquad (19)$$

where g = mass velocity in stack, lb/s/ft²; and D' = stack diameter, ft.

The remainder of the flue-gas losses can be expressed in terms of the velocity head, based on the flue-gas mass velocity at the location under consideration:

$$\text{Velocity head (in. } H_2O) = (0.0030)(g^2)(V_g) \qquad (20)$$

The losses can be estimated as the product of the velocity head at each location and a factor. For bare convection tubes, this factor is 0.2 (per row); for finned tubes it is 1.0 (per row); for the stack entrance it is 0.5; for the damper it is 1.5; and for the stack exit it is 1.0.

It should be noted that the convection section exerts a stack effect due to its physical height and serves to reduce the overall draft requirement.

In order to assure good flue-gas distribution throughout the convection section, it is usual for a flue-gas withdrawal opening to be provided at every 40 ft of convection section length.

At best, the determination of flue-gas flow is but an approximation. Operation at excess air levels greater than design produces increased quantities of flue gas. Furthermore, efficiency falloff due to solids deposition on heat-transfer surfaces also results in greater flue-gas quantities. Consequently, a stack design based only on the design flue-gas flow affords virtually no overcapacity in terms of the firing rate, and it severely restricts increases in throughput. To ensure flexibility for nominal capacity increments, the designer will often add an overcapacity of 25% to the flue-gas design flow when estimating the various flue-gas frictional losses.

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FIRED HEATERS—IV

How to reduce your fuel bill

This final article in a four-part series takes a look at how energy conservation has affected the design and operation of fired heaters. Items of concern include excess-air reduction, enhanced heat recovery, combustion-air preheating, and the conversion of gas-fired heaters to liquid firing.

*Herbert L. Berman, Caltex Petroleum Corp.**

☐ Demands on the CPI for greater energy conservation have had direct impact on the design and operation of fired heaters. Fuel savings have been realized with a variety of measures, ranging from the inexpensive—such as the fine-tuning of operating procedures and the upgrading of maintenance techniques—to the capital-intensive—such as the installation of complex heat-recovery facilities at substantial initial outlay. The following discussion will focus on methods that have proved successful in raising the thermal efficiency of fired heaters.

Reduction of excess air

In existing installations, excess air is the most important combustion variable affecting the thermal efficiency of a fired heater. Although heater operation is easier to control at high excess-air levels, it is very costly. The higher the excess air, the greater the fuel consumption for a given heat absorption. The extra fuel is consumed in heating the excess air volume from ambient temperature to the temperature of the exiting flue gases.

In order to exercise greater control over excess air, the oxygen content of the flue gas should be monitored above the combustion zone. Often, excess air in the combustion section may run as low as 10 to 15%, but stack-gas analysis reveals an oxygen content equivalent to as much as 100% excess air. This differential results from air leakage into the heater that occurs between the combustion zone and the stack. Such leakage cannot be corrected by burner adjustments.

Air leakage into a heater can occur at many locations. One route of entry is through the seams of the steel casing, between adjacent plates and stiffening members. Air can also enter through distorted or poorly gasketed header boxes. The terminal tubes in the tube-coil, where they enter and leave the heater casing, can likewise be a source of air leakage. Only through a rigorous, continuing maintenance effort can air leakage into heater settings be controlled effectively.

There are various schemes for controlling excess air via the monitoring of flue-gas oxygen content. Four such monitoring schemes are listed here, in order of increasing sophistication and cost [1]:

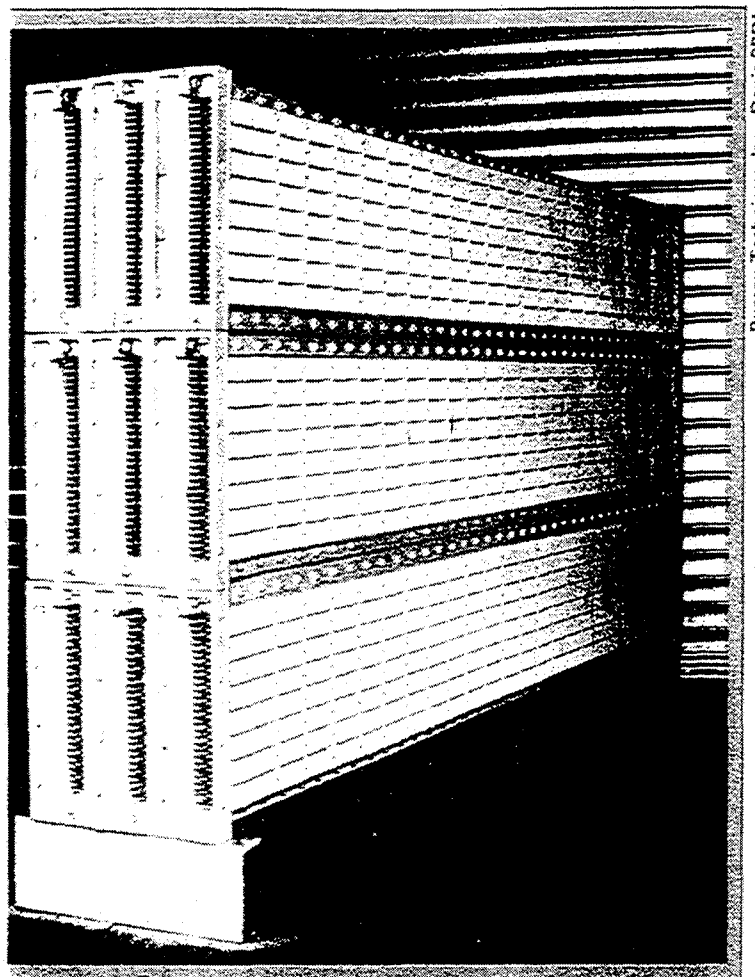
1. The least costly scheme requires only periodic checking of oxygen content using a portable oxygen analyzer and a portable draft gage. With these readings as a guide, the operator can make the adjustments necessary to operate at minimum excess-air levels.

2. One of the most common monitoring systems employs a continuous oxygen analyzer equipped with a local readout device and a permanently mounted draft gage. Continuous readings enable the operator to adjust heater operation whenever necessary.

3. Bringing the oxygen and draft readouts into the control room and adding a remotely controlled, pneumatic damper positioner is the next step toward improving excess-air control.

4. Although generally justifiable only for large heaters, further sophistication in excess-air control can be achieved with automatic stack-damper control. In this control scheme, the damper is positioned automatically according to a draft signal received from a probe located below the convection section. The controller changes the damper position so as to hold a set-point

*To meet the author, see Part I, which appeared on p. 98 of the June 19, 1978 issue. Part II appeared July 31; Part III, Aug. 14. See box at the end of this article for information on reprints.



Recuperative preheater warms combustion air. Unit is equipped with finned, cast-iron tubes. Fig. 1

draft corresponding to the targeted excess air. The draft set point is normally changed only to achieve variations in heater duty.

Further recovery of convection heat

The potential for additional heat recovery in the convection section exists when the heater operates at a relatively high flue-gas temperature. First consideration should be given to augmenting the convection-section surface area by adding several rows of convection tubes in the same heating service. Many current installations have anticipated such expanded service, and have been designed to accommodate the future installation of two rows of convection tubes.

The installation of supplementary heat-exchange units to reclaim additional convection heat often provides a quick economic return. Falling into this category are a number of units that recover process heat. Examples are feed preheaters for pyrolysis and steam hydrocarbon-reformer heaters, reboilers operated in conjunction with catalytic reformers, and superheaters providing process steam for petroleum refinery distillation units.

Very often the supplementary unit recovers convection heat for use in a steam-generating facility rather than in a process-stream heating application. Steam-generating units driven by convection heat are routinely coupled with catalytic-reformer heaters and steam hydrocarbon-reformer heaters. Fired heaters in such installations have convection sections equipped with several independent coils so that the same convection section furnishes heat for boiler-feedwater heating, steam generation, and steam superheating. Coils installed in the convection sections of steam-generating fired heaters are almost always of the forced-circulation type.

Occasionally, flue gases from several fired heaters are routed to a central waste-heat recovery facility. Typically, the recovered heat is utilized for steam generation. Waste-heat boilers of this type are normally designed with a flue-gas bypass around the boiler, which allows the heaters to operate even when the boiler is taken out of service. Where several heaters are involved, provisions for positive isolation at the flue-gas ducts from each heater will enable the user to take an individual heater out of service while the remaining heaters and the waste-heat boiler stay on line.

Before any add-on heat recovery device is retrofitted to a heater, the structural integrity of the existing steelwork and foundation must be examined to assess the effects of the increased loads.

Furthermore, it should be noted that the additional convection-section heat recovery will result in a lower stack-gas temperature, thereby reducing the stack draft. In addition, the installation of more surface area will increase the flue-gas pressure drop. Therefore, it is imperative that the effect of any alterations on the stack draft be analyzed beforehand in order to determine whether additional stack height will be required.

Replace bare tubes with extended surface

Many older generation fired heaters are equipped with bare-tube convection sections and operate with high flue-gas temperatures at 65 to 70% thermal efficiency. By replacing the bare tubes with extended surface tubes, efficiency improvement in the neighborhood of 10% may be realized in some installations. This corresponds to a stack-gas temperature reduction of about 300°F.

The most economical conversion from bare to extended-surface tubes can be made when the reduction in tube size is such that the tip-to-tip diameter across the extended surface tube is the same as the outside diameter of the original bare tube. On this basis, it is very likely that the existing convection section tube sheets can be retained. However, the reduction in tube I.D. will result in higher fluid pressure drops unless an increase in the number of convection-section parallel passes can be tolerated. Conversely, if the same tube I.D. is maintained, the conversion from bare to extended surface will necessitate the replacement of the convection-section tube sheets—with accompanying down-time and expense.

When conversion is contemplated for a heater firing liquid fuel, it is recommended that soot blowers be installed, in view of the greater fouling tendency of extended-surface devices. The cavities required to accommmodate the soot-blower lances can usually be

created simply by omitting the installation of a row or two of tubes at selected locations in the convection section.

Again, it is mandatory that the stack draft be assessed to ascertain whether the existing stack is adequate.

Preheat the combustion air

Fuel consumption in a fired heater can be reduced markedly by preheating the combustion air. In the preheater, heat is transferred from the flue gas to the combustion air, reducing the exit temperature of the flue gas and raising the thermal efficiency. With air-preheat systems, exit flue-gas temperatures often range in the 300 to 350°F range and efficiency levels commonly reach 90 to 92% (LHV). With such systems, the attainable thermal efficiency is no longer controlled by the approach between the flue-gas and inlet-fluid temperatures.

The salient features of the more popular air-preheaters are noted here:

■ *Recuperative-type air preheaters.* These devices, typically of tubular construction, transfer heat by convection from flue gas to combustion air. Customarily, the air flows inside the tubes, whereas the flue gas flows across the tube bundle. The preheater can be installed in the fired heater above the process convection section or, as is more usually the case, at grade alongside the heater.

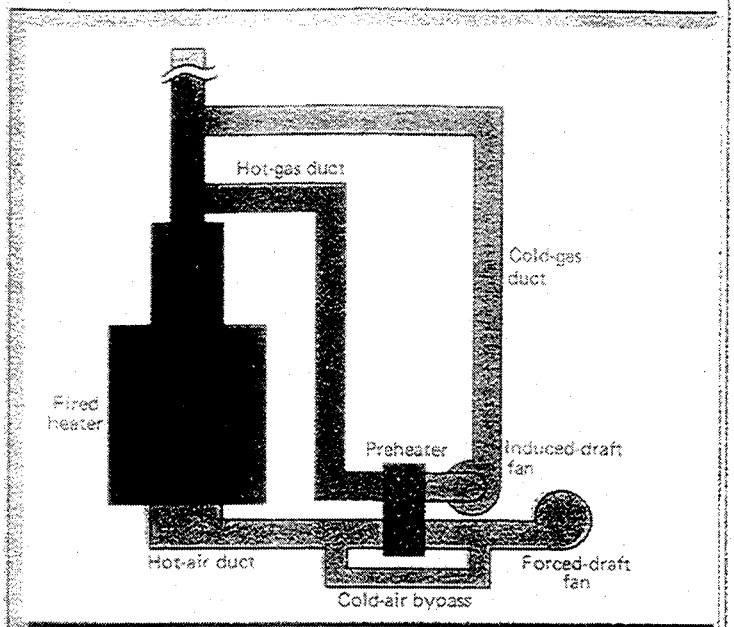
Materials selected for heat-transfer surfaces vary from designer to designer. If the flue gas is well above the acid dewpoint, a manufacturer may select cast iron tubes having internal and external fins. In the zone approaching the dewpoint, fins are provided on the gas side only, in order to keep tube-metal temperatures as high as possible. Below the dewpoint, plain tubes of borosilicate glass are used in order to minimize acid corrosion.

Tubular-type air preheaters are essentially tight between air and gas, with no leakage of air into the flue-gas side. Fig. 1 shows a bank of tubular cast-iron heating elements equipped with internal and external fins.

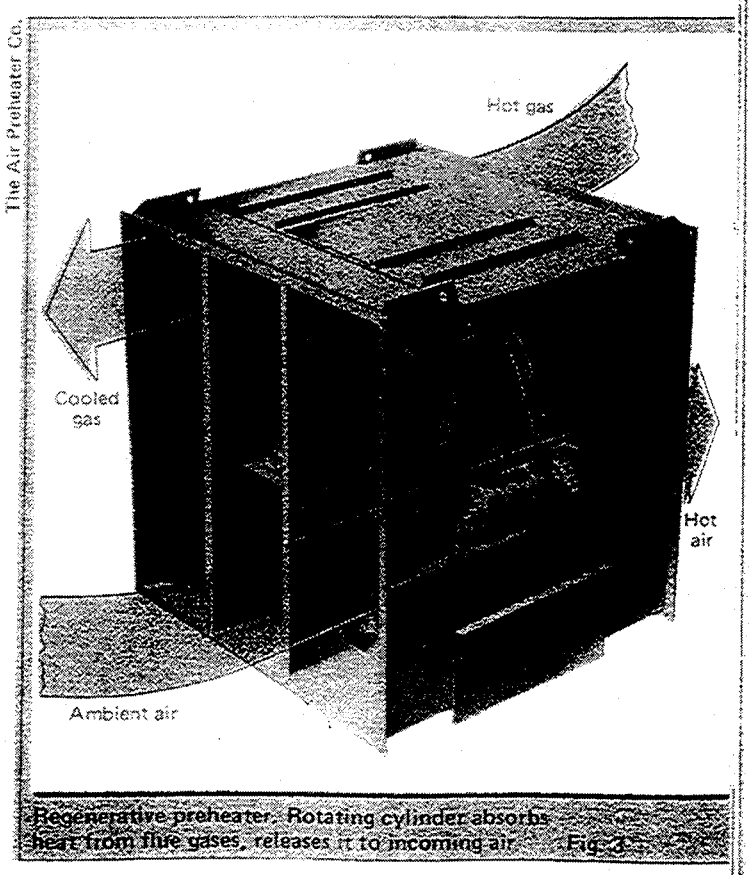
A schematic arrangement for a typical recuperative air preheater installation is shown in Fig. 2. (The arrangement shown is equally applicable to regenerative preheater systems, described below.) The system employs a forced-draft fan to supply combustion air, and an induced-draft fan to maintain a negative pressure and draw the flue gas through the system to the stack. A cold-air bypass enables the operator to route a portion of the incoming combustion air around the preheater when ambient temperature is very low. The preheater surface area can thus be maintained above the acid dewpoint, at a nominal sacrifice in thermal efficiency.

■ *Regenerative-type air preheaters.* This apparatus consists of heat-transfer elements housed in a subdivided cylinder, which rotates inside a casing. Hot flue gases pass through one side of the cylinder, cold air through the other side. As the cylinder slowly rotates, the elements continuously absorb heat from the flue gas and release it to the incoming air stream.

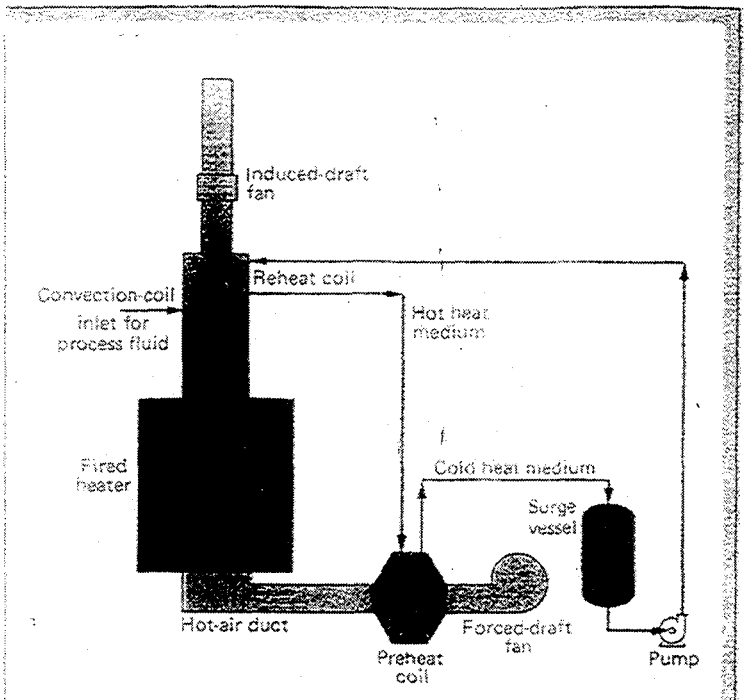
The cylinder is subdivided by baffles which, like the seals between the cylinder and the casing, help to minimize leakage of air into the flue-gas stream. This leakage, which results from the pressure differential that exists between the air side and the flue-gas side, is normally 10 to 15% of the total air flow. The preheater system, particularly the forced-draft and induced-draft



Typical process scheme used for both recuperative and regenerative air preheaters Fig. 2



Regenerative preheater. Rotating cylinder absorbs heat from flue gases, releases it to incoming air Fig. 3



Heat-medium preheater transfers heat from flue gases to air via intermediate, circulating fluid Fig. 4

fans, must be designed to accommodate this leakage.

The heating elements of the regenerative air preheater are installed in two or three layers. Corrosion of the elements is usually confined to the final portion of the cold-end layer, where cold air enters and cooled flue gas leaves. If such corrosion does occur, the elements of the cold-end layer can be removed and reversed to extend their service life. For those applications where the elements are exposed to very corrosive atmospheres, porcelain-enameled heating surfaces are available. Fig. 3 illustrates the basic construction of a regenerative air preheater. The main components of the overall process arrangement are shown in Fig. 2.

■ *Heat-medium air preheaters.* Instead of direct heat exchange between air and flue gas, these units employ an intermediate fluid to transfer heat from the flue gas to the incoming combustion air. The heat medium is contained in a closed loop that includes a reheat coil located in the flue-gas flow downstream of the process convection coil, and a preheat coil positioned in the air stream. The circulating fluid extracts heat from the flue gas, lowering the gas temperature and raising the fluid temperature. In turn, the hot fluid releases its heat to the incoming air.

The process scheme for a heat-medium preheat system (Fig. 4) includes, in addition to the preheat and reheat coils, a fluid-surge vessel and a circulating pump. Depending on the design of the heater and the available draft, the system may be provided with forced draft/induced draft, forced draft only, or induced draft only.

■ *Process-fluid air preheaters.* These systems [2] take off a portion of the process stream entering the convection section and send it to a preheat coil mounted in the air stream, which warms the air. The subcooled process fluid is then returned to a reheat coil located in the flue-gas flow downstream of the main-process convection coil, which reheats the fluid to its original temperature level and returns it to the main process stream ahead of the convection coil.

Only a portion of the process fluid stream is drawn off for preheat service. The mass flows and specific heats of the combustion air and process fluid must be carefully balanced in order to achieve the desired temperature changes in each stream.

A schematic arrangement for a process-fluid air preheater is shown in Fig. 5. As with heat-medium preheaters, these systems can be supplied with forced draft/induced draft, forced draft only, or induced draft only.

Process effects of air preheating

Fired-heater operation with preheated combustion air results in higher adiabatic flame temperatures than with ambient air. From the combustion kinetics, then, one might expect heaters using preheated air to generate higher amounts of NO_x than those using ambient air.

However, nitrogen fixation from the atmosphere is usually a relatively minor factor in NO_x production. The nitrogen which most readily converts to NO_x is that which is chemically bound up in the fuel. The quantity of oxides of nitrogen formed from this source is not appreciably affected by elevated combustion temperatures. In fact, with an air preheater, the reduction in the amount of fuel consumed corresponds directly to a reduction in the amount of NO_x formed from fuel-bound nitrogen, and is quite likely to offset the additional atmospheric fixation due to the higher flame temperature.

Another process effect that manifests itself when an air-preheat system is retrofitted to an existing heater is the shift in duty split between the radiant and convection sections. The higher combustion temperature due to preheating, coupled with the lower convection transfer rate at reduced flue-gas flowrates, causes a greater proportion of the total heat absorption to take place in the radiant section.

Consequently, operation with preheated combustion air results in higher radiant duties than with ambient air. The magnitude of the shift in duty split is, of course, contingent upon the degree of preheat applied to the combustion air.

Gas-turbine exhaust used as combustion air

The exhaust gas from gas turbines, widely used as drivers for compressors, pumps, and electrical generators, usually ranges from 800 to 900°F and contains from 17 to 18% oxygen. These gases are well-suited as a source of preheated combustion air for fired heaters, and can be used to cut heater fuel consumption.

In most systems utilizing gas-turbine exhausts, an auxiliary source of combustion air permits independent operation of the heater and the gas turbine. Typically, the overall system is designed so that the gas-turbine exhaust can be vented to atmosphere whenever the heater is operated on ambient air.

Conversion from gas to liquid firing

In the face of curtailments of natural gas deliveries, many industries are seeking alternative fuel sources, primarily liquid fuels. The following areas of concern should be examined closely before any conversion from gas to liquid firing is undertaken.

Impact on radiant section. The major factor affecting the combustion chamber is the size of the oil flame compared to the size of the gas flame. At the same level of heat release, an oil flame is generally longer and wider than a gas flame. For this reason, the distance from burner to tube should be examined to assess the potential for flame impingement on the tube coil. Similarly, vertical clearance dimensions should be reviewed, since the longer oil flame may impinge on the shield tubes, the reradiating cone, or the baffle provided at the top of some older vertical-cylindrical heaters.

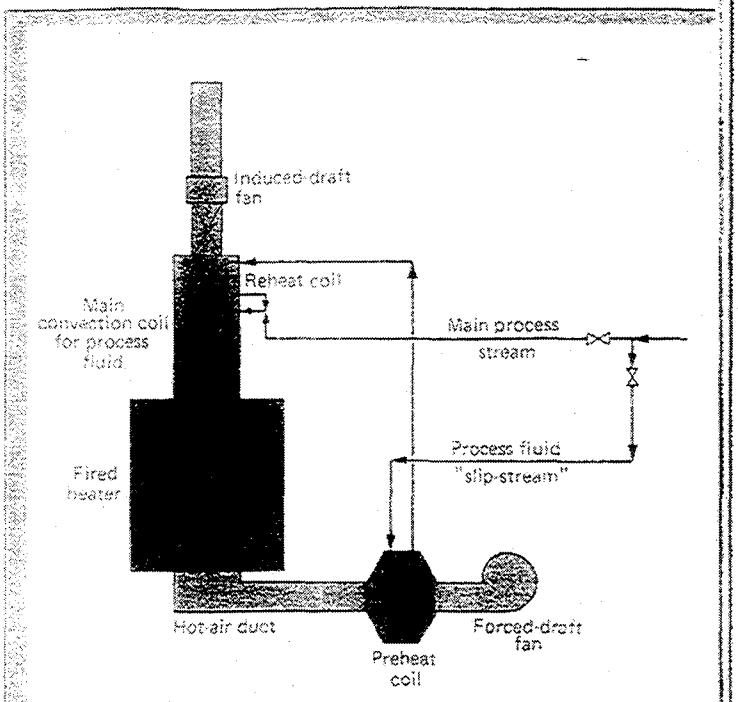
If the contemplated liquid fuel contains substantial vanadium and sodium concentrations, consideration should be given to protecting exposed tube supports from the corrosive combustion environment. Also, if the heater contains a reradiating cone or baffle, the heater manufacturer should be consulted regarding the possibility of removing such vulnerable equipment.

Impact on convection section. Convection-section extended surface in gas-fired heaters is very often of the high-density, finned-tube type. Under heavy liquid-fuel firing, such tubing is difficult to keep clean using conventional onstream cleaning techniques. Before conversion, therefore, the designer should consider replacing high-density finned tubes with either heavy low-density tubes, or studded tubes. Since replacement will reduce the total effective surface area, additional rows of convection tubes will be required to maintain thermal efficiency.

Facilities for onstream convection cleaning, such as soot blowers, should be installed as part of the conversion. The addition of such equipment, as well as the necessary ladder and platform access, requires that the structural integrity of the steelwork and foundation be assessed.

Adjustments to draft. Many heaters designed for gas firing operate virtually at their draft limit, usually because they are pushed well beyond their original design capacity. Because oil firing requires higher excess-air levels to achieve acceptable combustion, draft-limited gas-fired heaters will suffer a capacity decrease when converted to liquid firing, unless additional stack height is provided.

Alterations to burners. Burner replacement in a gas-to-liquid fuel conversion can, in most instances, be made on a one-for-one basis. However, for those conventional gas-fired heaters that operate with relatively small burner-heat releases in the range of 2 to 3 million Btu/h, substitution of oil burners on a one-for-one basis may result in unstable combustion. In these cases, a totally new firing arrangement should be specified as part of the conversion. The final burner arrangement should take into account such parameters as the combustion-chamber configuration, burner turndown requirements and fuel-oil combustion behavior.



Typical arrangement for air preheater that uses process fluid as heat-exchange medium Fig. 5

From a practical standpoint, an important consideration that must not be overlooked in any conversion study is the heater's mechanical condition and its remaining useful life. Investment costs for gas-to-oil conversions are expensive, particularly when convection sections must be revamped. Furthermore, the downtime necessary to effect the conversion is costly, due to lost revenues. Consequently, if conversion is contemplated for a heater that is in poor mechanical condition, an evaluation of maintenance costs, the conversion investment and revenue losses may well show replacement to be economically more attractive.

A good part of conversion cost lies in the fuel handling and distribution system. Conversion of only a few large units, particularly if they are located in the same general plot area, will reduce this cost.

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A SPECIAL REPORT

Rating Fired Heaters

The method of rating or predicting the performance of direct fired heaters presented in this special report simply involves a heat balance calculation between the heat releasing streams or hot furnace gases and the heat absorbing process streams plus a rate relationship.

There are several limitations on furnace design. One is efficiency which depends on fuel costs, process fluid temperature, tube material and the user's policy on investment payout. Excess air rates also have a significant effect on furnace efficiency, and, in general, the lower the rates the smaller the heater.

Another limitation on design is the maximum allowable absorption rate in the radiant section. The higher rates result in smaller, less expensive heaters but also generally require more maintenance. The user must determine the optimum rate according to his particular philosophy, at the time, on investment versus maintenance costs.

Finally, the user usually sets the maximum allowable pressure drop of the process fluid in the heater. A process designer who specifies a low pressure drop can increase heater costs considerably by requiring the heater designer to use higher tube sizes and reduce the number of parallel passes.

Here at last is a method of predicting fired heater performance that will provide the user with an optimum sized heater and can serve as a common basis for calculation between the user and manufacturer.

Frank P. Evans

MECHANICAL ENGINEERING EDITOR

Here is a method used to predict the performance of direct-fired heaters in typical refinery and petrochemical process applications

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DIRECT-FIRED HEATERS have been used to supply process heat ever since the beginning of the refining and petrochemical industries. Relatively little information, however, has been published covering methods for predicting the performance of these heaters. The heat transfer situation existing in a refinery heater is so complicated that direct application of the basic radiation and convection relationships is extremely involved and laborious. Therefore, the industry has depended to a large degree on strictly empirical correlations.

This article describes a method for computing the performance of both the radiant and convection sections of fired heaters. The method is based on fundamental correlations for heat transfer by radiation and convection. It is applicable to the usual types of refinery heater, where combustion takes place in a firebox without flame impingement on tubes or refractory walls. However, it cannot be used directly for applications involving radiant-type burners or specially-fired refractory walls.

The rating procedures were developed from methods presented by Lobo and Evans[2] for the radiant section, and Monrad[5] for the convection section. No attempt will be made here to repeat the detailed basis and development of their correlations. Reference should be made to the original papers if more extensive background information is required.

Rating a fired heater simply involves a heat balance between

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

Figure 1 shows a cross-section of a typical fired heater as used in refineries today. It consists of a firebox or radiant section, a convection section, a hood for collecting the cooled flue gas, and a stack to dispose of the gas and provide draft.

The radiant section provides space for the fuel to be completely mixed with air and burned. It also contains heat-absorbing tubes, which remove a large portion of the heat from the flue gas before it passes out to the convection section. The tubes are generally arranged around the outside of the firebox just in front of refractory walls, as shown in the sketch. For some applications, however, tubes may be located in the center of the furnace, with burners arranged to fire on both sides. At the temperature levels existing in the firebox, most of the heat is transferred by radiation. Therefore, the tubes must be arranged for uniform and efficient radiant absorption.

Many heaters in the past have been designed so that the convection section was completely shielded from the firebox by a refractory bridge wall. This was to protect the front rows of convection tubes from excessive radiant heat. It has been found, however, that with proper design of the convection section, a bridge wall is not required. The tubes in the convection section that can see the fire are called shield-tubes. Although they may be included mechanically as part of the convection section, for process design the shield tubes must be considered as part of the radiant section.

The convection section recovers additional heat from the flue gas, at a lower temperature level than can be economically obtained in the radiant section. Here the tubes are arranged to give high mass velocities and turbulence in the gas, so as to obtain good convective transfer.

The hood and stack must simply collect and dispose of the flue gas without excessive frictional loss, and provide enough draft to draw the gases through the firebox and convection section.

Since the physical arrangement and governing heat transfer mechanism are different in the radiant and convection sections, different methods are used to rate the two sections.

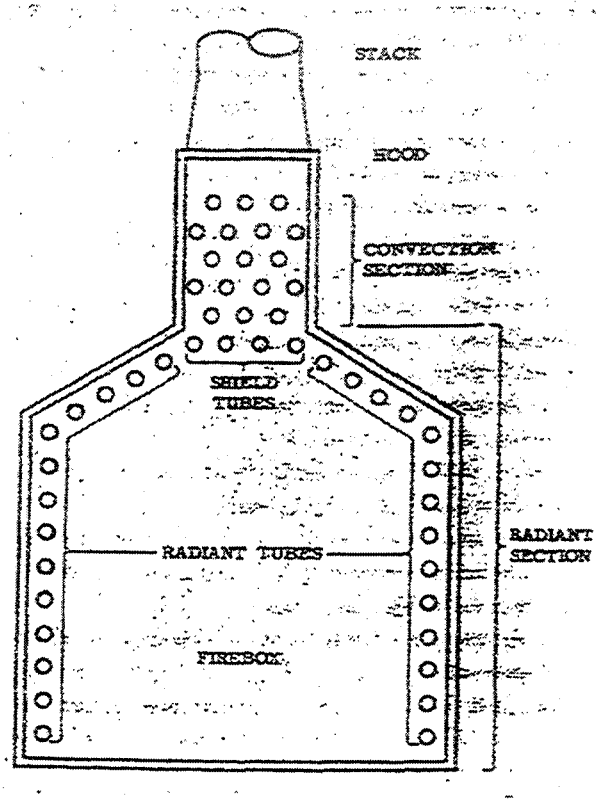


FIGURE 1—Cross-section of a typical fired heater.

HEAT TRANSFER: RADIANT SECTION

Extensive investigations have been made of the radiant heat transfer between solid surfaces in various arrangements, and between hot gases and solids.[1-6,7] Lobo and Evans[2] applied the basic radiation concepts to furnace design, and developed a rating method that was generally applicable without excessive complication. The method described here follows their basic pattern. It has been further simplified, however, by elimination of some minor variables and inclusion of additional general correlations.

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

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

The Stefan-Boltzman constant σ has a value of 0.173×10^{-8} Btu/sq ft., hr, °R⁴.

For radiant heat transfer between two real surfaces at temperatures T_a and T_b, the relation becomes

$$q_r = \sigma A F (T_a^4 - T_b^4) \qquad (2)$$

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

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

The usual heat-absorbing surface consists of a number of parallel cylindrical tubes in front of a refractory wall.

Part of the radiation from the hot gas strikes the tubes directly and is absorbed. The remainder passes through to the refractory, and is reradiated back into the furnace. Again, part of the reradiated energy is absorbed by the tubes, and the remainder passes through.

This complicated situation is handled in the calculations by replacing the tube bank with an equivalent plane surface, A_{cp}. It equals the number of tubes, times their exposed length, times the center-to-center spacing. The tube bank, however, will not absorb all the energy radiated to the cold plane area. So the total cold plane area must be corrected by an absorption efficiency factor α. Hottel[2,7] has published values for the factor α as a function of tube arrangement and spacing. His curves for one or two rows of tubes in front of a refractory wall are reproduced in Figure 2.

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

In the type of heater shown in Figure 1, the shield tubes require special handling. These tubes are not backed by a reradiating refractory wall. Instead, any energy that is not directly absorbed by the shield tubes passes on to the tubes in the convection section, and is absorbed there. Therefore, from the standpoint of the radiating gas, the factor α for the shield tubes is unity. So in calculating the total equivalent cold-plane area in a furnace, the applicable value of α is applied only to the refractory-backed tubes. The cold-plane area of the shield tubes is taken at full value. It might be argued that the shield tubes do not pick up all this heat, so the computed heat-absorption in the radiant section will be too high. However, actual experience shows that because of the high gas velocities past the shield tubes, convective transfer to them is higher than to the normal radiant tubes, and more than compensates for the loss in radiant transfer. The assumption that α equals unity for these tubes is a reasonable allowance for the added convective effects, and simplifies the calculations.

Exchange-Factor. The remaining term to be evaluated in Equation 2 is the exchange-factor, F. First, we must account for the fact that the gas in the firebox is far from an ideal black body. The only constituents in normal flue gas that contribute significantly to the radiant emission are carbon dioxide and water. The emissivity of a body of gas depends on concentration of each of these two components, the furnace dimensions, and the temperatures of the gas and of the absorbing surface. Lobo and Evans have shown that the composition and dimensional effects can be accounted for by a single term, the partial pressure of carbon dioxide plus water multiplied by the mean beam length. Figure 3 shows the partial pressure P of the radiating components as a function of excess air for the usual hydrocarbon fuels. Table 1 gives the mean beam length L for various furnace shapes. It has been found that the tube wall temperature has only a minor effect. Therefore, the emissivity can be correlated as a function of the PL product and gas temperature, as shown in Figure 4. Variations in tube wall temperature between 600 and 1200° F cause less than one percent deviation from these curves.

The exchange-factor is also dependent on the amount

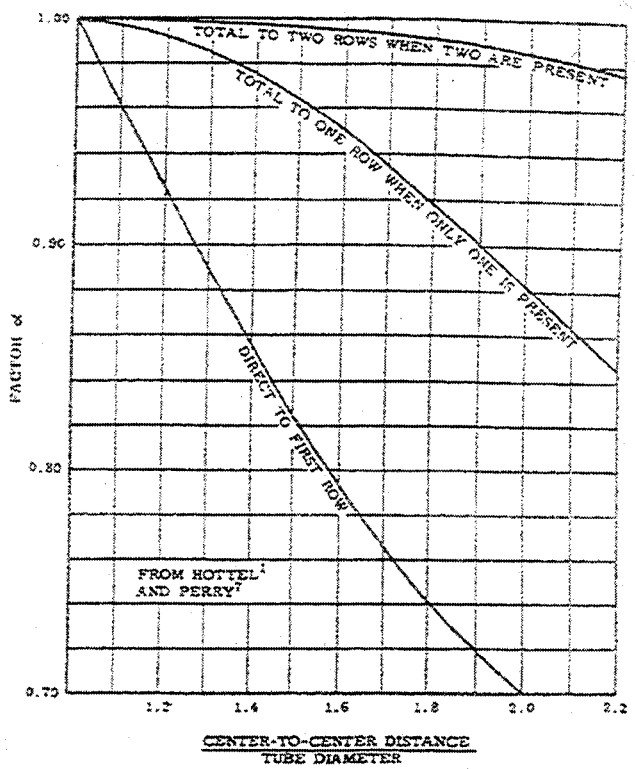


FIGURE 2—Absorption efficiency of the tube-banks.

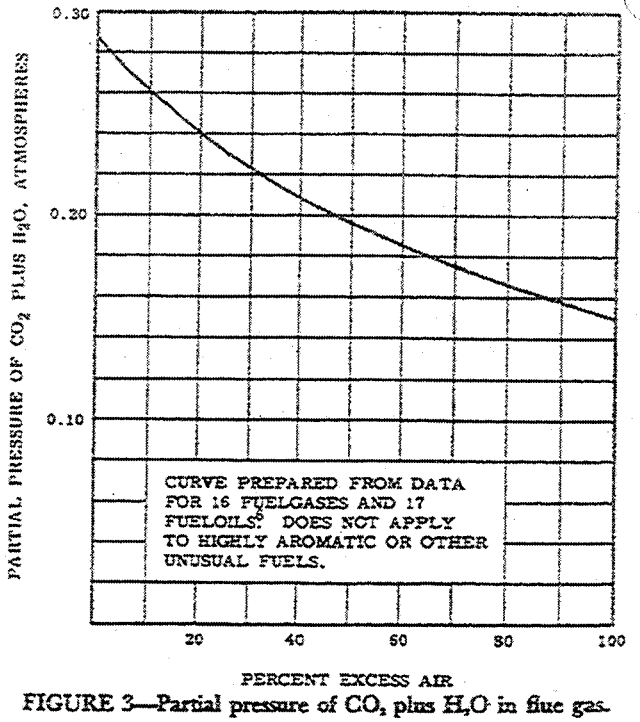


FIGURE 3—Partial pressure of CO_2 plus H_2O in flue gas.

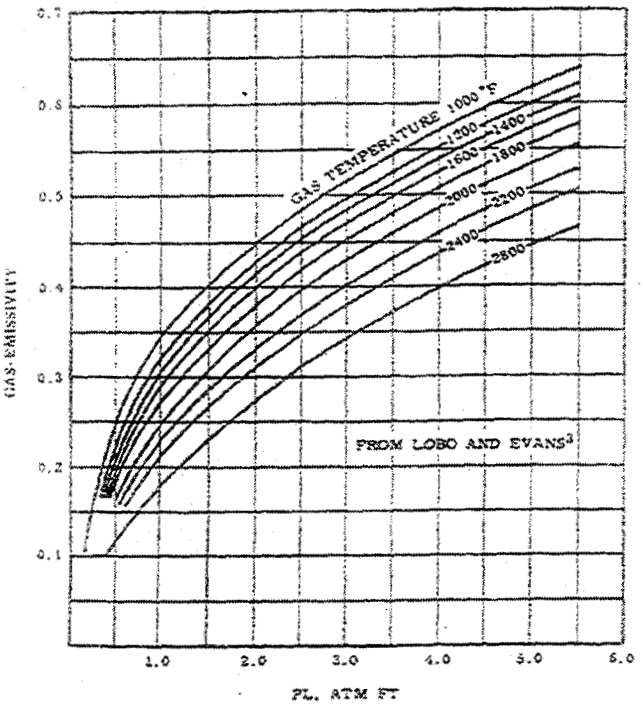


FIGURE 4—Gas-emissivity.

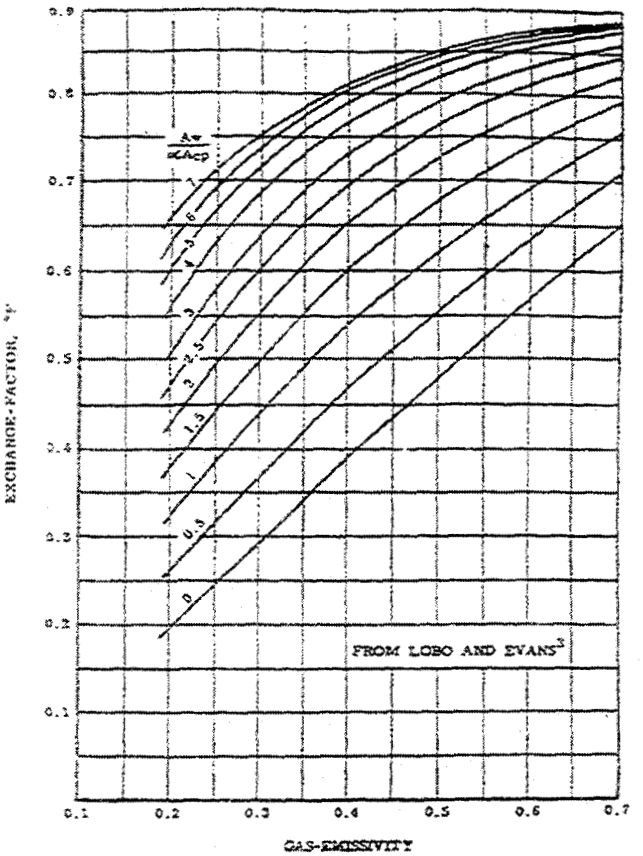


FIGURE 5—Over-all radiant exchange-factor.

of reradiation from exposed refractory. Energy striking such refractory is, in effect, reflected back toward the tubes. Thus a furnace with a large amount of exposed refractory will transfer more heat per unit of tube surface than one with most of the walls covered by tubes. This effect can be correlated on the basis of the ratio of exposed refractory area to equivalent cold-plane area, as shown in Figure 5. The exposed refractory area is defined as the area which would be exposed if the tube bank were replaced by the equivalent cold plane. That is, it is the total envelope area of the firebox, less the equivalent cold-plane area of all the tubes.

Figure 5 also allows for the fact that the tubes do not completely absorb all the radiant energy that strikes them. The curves are based on a tube-surface absorptivity of 0.9, which is a commonly accepted value for oxidized metal surfaces.

TABLE 1—Beam Lengths for Gas Radiation

DIMENSION RATIO	MEAN BEAM LENGTH, L
Rectangular Furnaces, Length-Width-Height, In Any Order	
1-1-1 to 1-1-3 1-2-1 to 1-2-4	3.6 (Furnace Volume)$^{1/3}$
1-1-4 to 1-1-∞	1 x Smallest Dimension
1-2-5 to 1-2-∞	1.3 x Smallest Dimension
1-3-3 to 1-∞-∞	1.8 x Smallest Dimension
Cylindrical Furnaces, Diameter-Height	
1-1	2/3 x Diameter
1-2 to 1-∞	1 x Diameter

The working equation for calculating radiant heat transfer in the firebox thus becomes

$$q_{Rr} = \sigma \alpha A_{cp} F (T_g^4 - T_t^4) \quad (3)$$

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, depending on the temperature level. The relation for convective heat transfer is

$$q_{Rc} = h_{Rc} A_{Rt} (T_g - T_t) \quad (4)$$

Since convective heat transfer is not the major contributor, we may make some simplifying approximations. For the usual furnace, h_{Rc} is about 2.0 Btu/hr sq ft F, A_{Rt} is about two times αA_{cp}, and the exchange factor F is about 0.57. Making these substitutions into Equation 4, we can put it into a form similar to Equation 3.

$$q_{Rc} = (2.0)\ (2\,\alpha A_{cp})\ (F/0.57)\ (T_g - T_t) = 7.0\,\alpha A_{cp} F (T_g - T_t) \quad (5)$$

Total Heat-Transfer Rate. The total heat absorbed in the radiant section is the sum of the radiant and convective transfer

$$q_R = q_{Rr} + q_{Rc} = \sigma \alpha A_{cp} F (T_g^4 - T_t^4) + 7.0\,\alpha A_{cp} F (T_g - T_t) \quad (6)$$

or, dividing through by $\alpha A_{cp} F$,

$$\frac{q_R}{\alpha A_{cp} F} = \sigma (T_g^4 - T_t^4) + 7.0 (T_g - T_t) \quad (7)$$

Thus the ratio $q_R/\alpha A_{cp} F$, is a function of gas and tube wall temperatures only. The relation is shown in Figure 6.

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

$$q_n + q_a + q_f = q_R + q_L + q_{g2} \qquad (8a)$$

Solving this equation for the heat absorbed.

$$q_R = q_n + q_a + q_f - q_L - q_{g2} \qquad (8b)$$

Finally, Equation 8b can be modified to a form similar to Equation 7.

$$\frac{q_R}{\alpha A_{cp} F} = \left[1 + \frac{q_a}{q_n} + \frac{q_f}{q_n} - \frac{q_{g2}}{q_n}\right] \frac{q_n}{\alpha A_{cp} F} \qquad (9)$$

The net heat release q_n is established as soon as the total heater duty and efficiency are established. The values of q_a and q_f are fixed by the temperature of the fuel and combustion air. The loss q_L is usually from one to three percent of the net heat release, and can generally be estimated from past experience. This leaves q_{g2} as the only variable on the right-hand side of Equation 9. It turns out that the fraction of net heat-release retained in the flue gas is primarily a function of temperature and excess air, for all common liquid and gaseous fuels. This relation is shown in Figure 7. Data for sixteen fuel gases and seventeen fuel oils agree with these curves within two percent.

Temperatures. Figure 6 shows the rate of heat transfer between a mass of gas at one uniform temperature, and a tube surface at another uniform temperature. In most actual furnaces, however, neither of these temperatures is constant. Average effective temperatures must be selected in order to use Figure 6.

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

The average effective gas temperature depends on the general shape of the furnace and the firing conditions. Experience has shown that for heaters of the type shown in Figure 1, with an approximately square cross-section and no large areas of refractory with direct flame impingment, the average gas temperature is very close to that of the exit gas. For some other types, however, such as a cylindrical furnace with a high length/diameter ratio, the average firebox gas is appreciably hotter than the exit gas. The magnitude of this difference must be determined empirically from experien with similar designs.

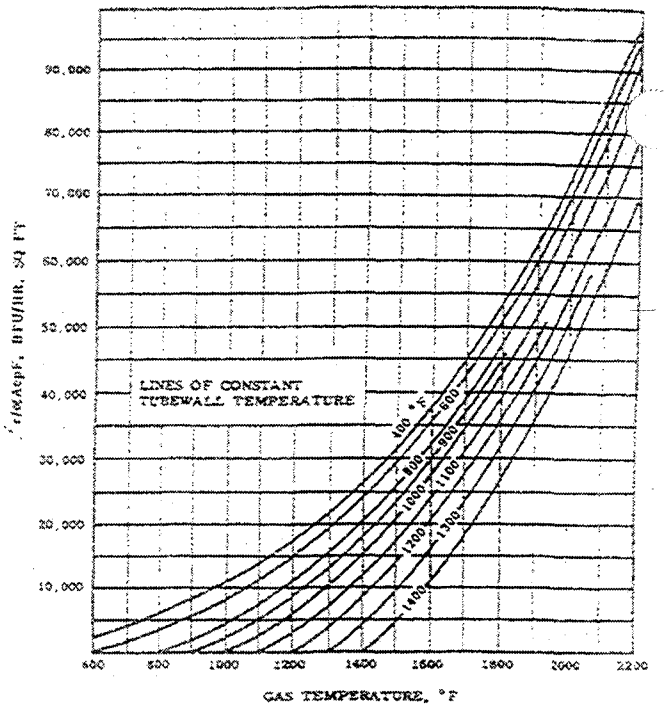


FIGURE 6—Total heat-absorption in the radiant section.

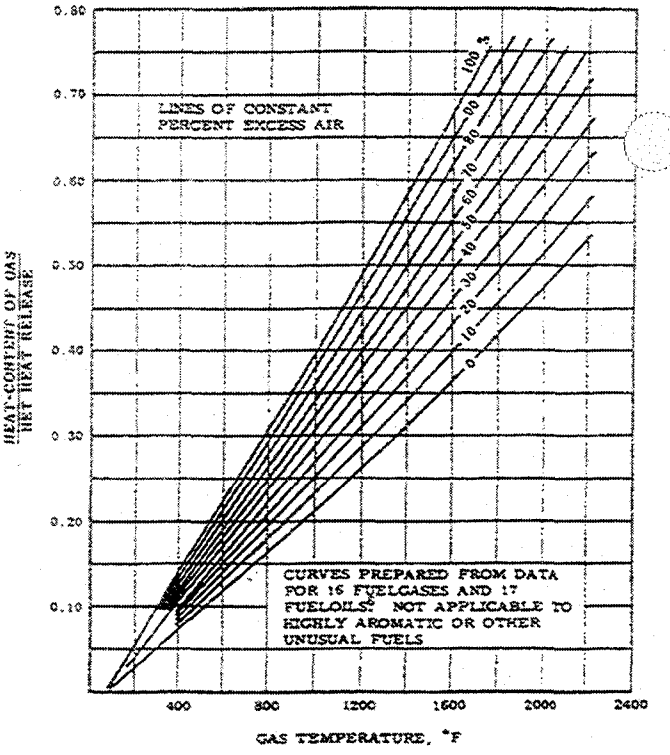


FIGURE 7—Heat-content of flue gas.

Radiant Section Heat Absorption. Equation 7 gives

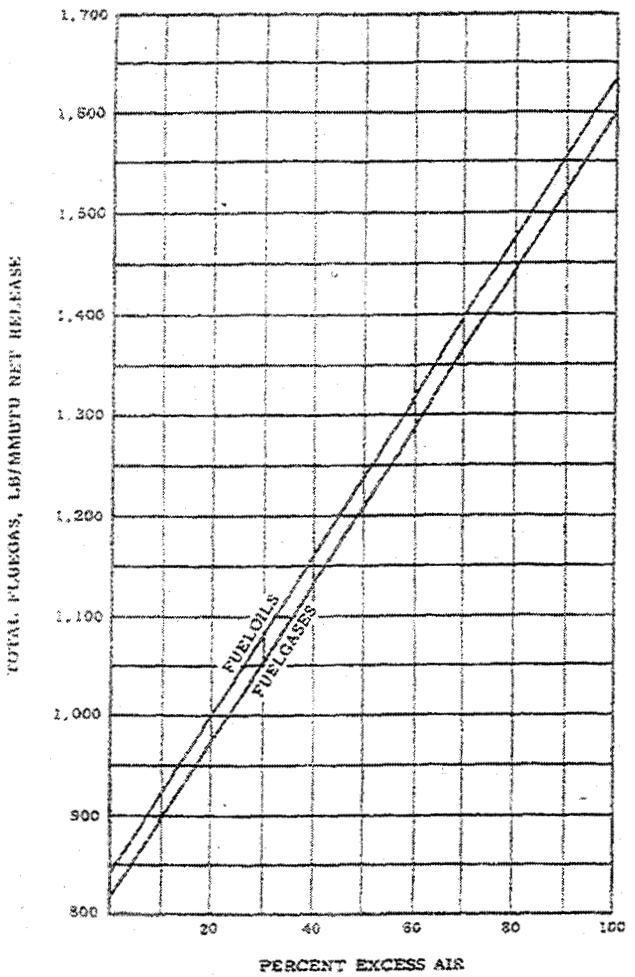


FIGURE 8—Fluegas rates.

the heat transferred to the tubes in the radiant section as a function of average gas temperature. Equation 9 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 9 on Figure 6. The value of t_g is read where the plot of Equation 9 crosses the applicable tube wall temperature. Then, from a heat balance, q_R is readily determined. Note that it is not necessary to plot Equation 9 over the full temperature range. Only that part which crosses the curves in Figure 6 is required. Furthermore, since Equation 9 is nearly linear with respect to gas temperature, two calculated points close to the intersection, joined by a straight line, are usually sufficient.

HEAT TRANSFER: CONVECTION SECTION

As in the radiant section, heat is transferred in the convection section by both radiation and convection. A firm basis for calculating convection-section heat transfer was first outlined by Monrad.[5] He accounted for direct convection, radiation from the gas, and radiation from the refractory walls. The method outlined here follows Monrad's theory, but with revised correlations to include later experimental results.[2,5,8]

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

$$h_{cc} = \frac{2.14\,(T_f)^{0.28}\,(G_{max})^{0.6}}{d^{0.4}} \qquad (10)$$

Figure 9 shows the convection coefficient for several common tube sizes. The average gas film temperature is defined as the average intube fluid temperature plus one-half the log mean temperature difference from flue gas to fluid. The flue gas rate is at the minimum cross-section in the convection section. It has been found that flue gas rates for all ordinary gaseous and liquid fuels can be correlated on the basis of net heat release and percent excess air. The relation is shown in Figure 8.

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.[4] His basic relation was solved using typical tube spacing of about two tube diameters, a tube surface absorptivity of 0.9, and a typical partial pressure of carbon dioxide plus water. The results are shown in Figure 10. The average

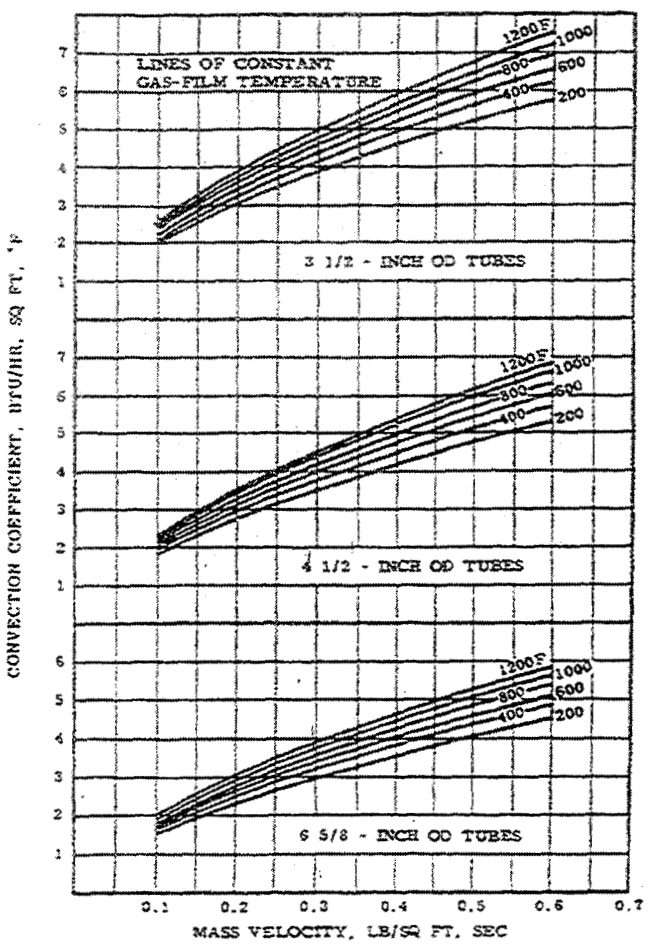


FIGURE 9—Flue gas convection-coefficients for flow across staggered banks of bare tubes.

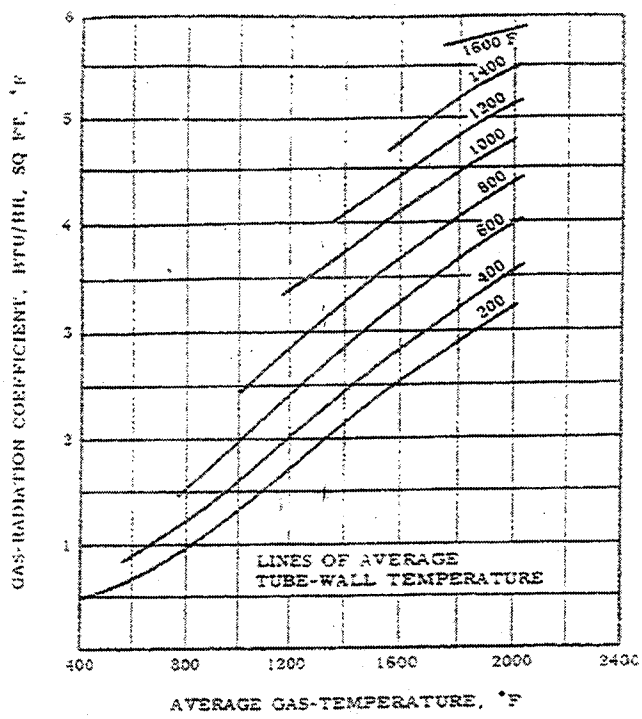


FIGURE 10—Gas-radiation coefficients.

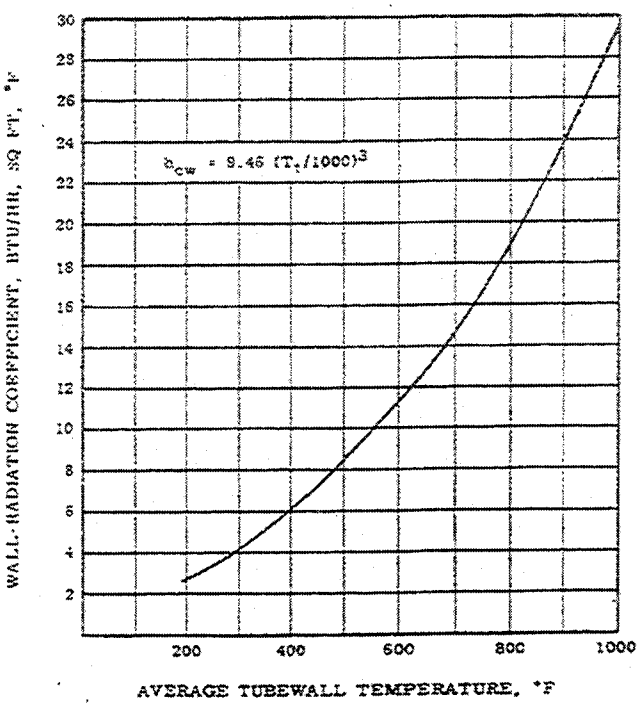


FIGURE 11—Coefficient for radiation from convection-section walls.

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.

Radiation From Refractory Walls. Following Monrad's pattern, we express the effective coefficient for

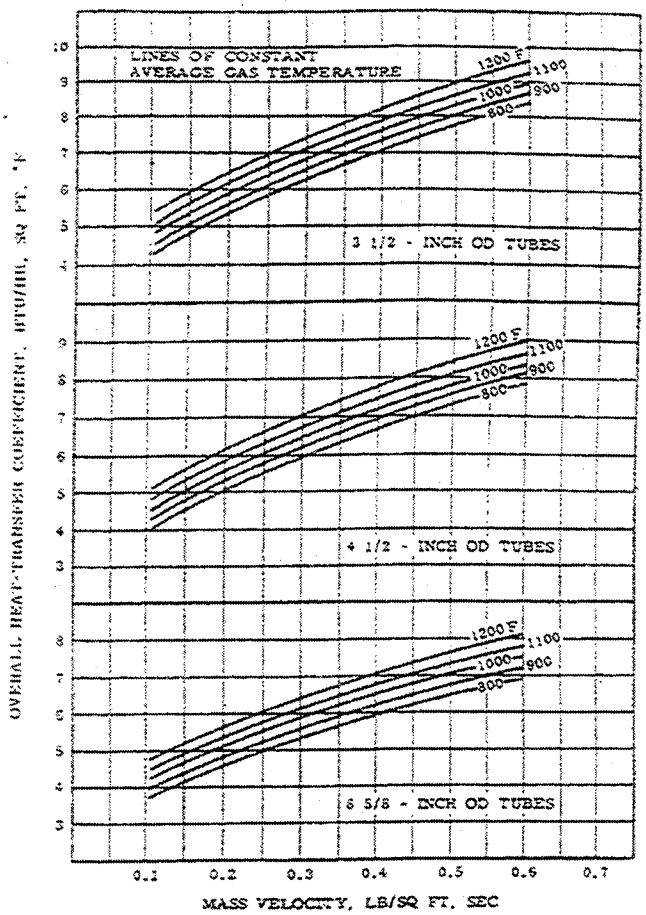


FIGURE 12—Over-all heat-transfer coefficients in the convection section.

wall-to-tube transfer as a function of tube wall temperature, as shown in Figure 11. This coefficient, together with the relative wall and tube areas in the convection section, determines a correction factor f.

$$f = \frac{h_{cw}}{h_{cc} + h_{cr} + h_{cw}} \left(\frac{A_{cw}}{A_{ct}} \right) \qquad (11)$$

Over-all Transfer Coefficient. The total apparent gas film coefficient is determined from the individual coefficients by the following equation

$$h_c = (1 + f)\ (h_{cc} + h_{cr}) \qquad (12)$$

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

$$V_c = \frac{h_c\ (h_{ct})}{h_c + h_{ct}} \qquad (13)$$

In many rating problems, particularly during preliminary work, the detailed calculations outlined above are not justified. For these cases, the curves in Figure 12 can be used. These represent overall convection section coefficients, assuming usual tube arrangements, fluid temperatures, and intube film coefficients.

Finned Tubes. Extended surface on the tubes in the convection section is often used as a means of obtaining

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

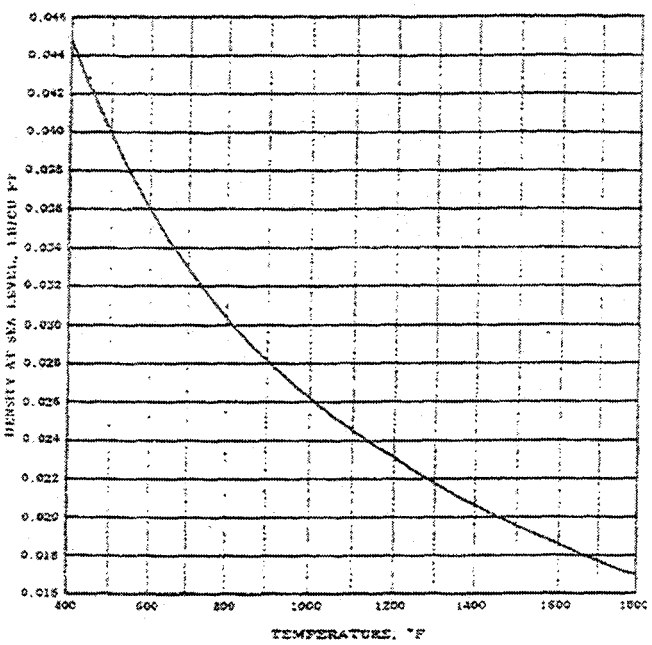


FIGURE 13—Density of flue gas.

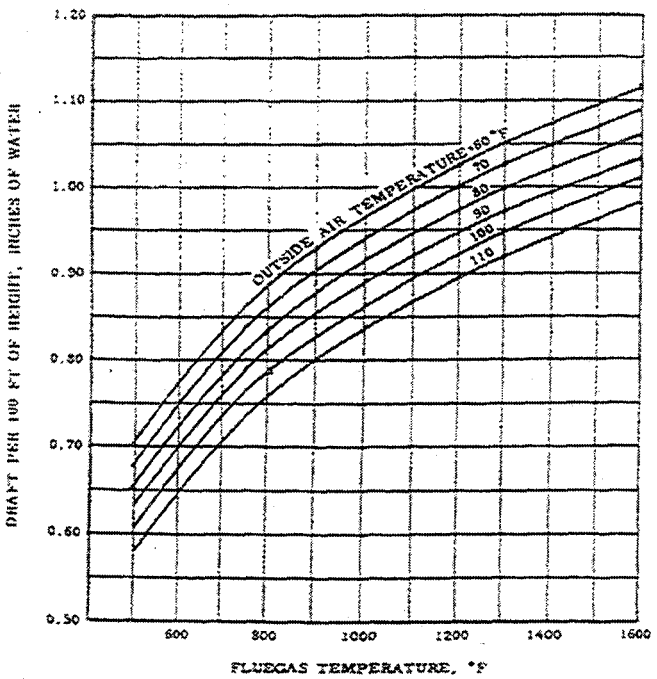


FIGURE 14—Stack draft.

STACK DESIGN

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

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

Convection Section. The pressure drop across 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 equation

$$p_v = \left(\frac{V^2}{2g}\right)\left(\frac{\rho_g}{\rho_w}\right)(12) = 0.0030\, V^2 \rho_g = 0.0030\, \frac{G^2}{\rho_g} \qquad (14)$$

Flue gas densities are shown in Figure 13. In the convection section, the gas density is evaluated at the same average gas temperature that was used to calculate the convection section transfer coefficient.

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
Stack friction	H/50D
Stack exit	1.0
Total	3.0 plus H/50D

Equation 14 and Figure 13 are used to calculate the velocity-head, just as for the convection section.

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

$$\Delta p = \frac{H\,(\rho_a - \rho_g)}{\rho_w}\,(12) = 0.187\, H\,(\rho_a - \rho_g) \qquad (15)$$

Available draft per 100 feet of height is shown in Figure 14. To allow for heat loss from the stack, its draft should be computed on the basis of a temperature 100° F lower than that of the flue gas leaving the convection section.

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

Stack Size. In many installations, minimum stack heights are set by fume dispersal requirements, particularly if there are tall structures nearby. If such is the case, the stack is sized so that the total friction drop is no greater than the available draft. If there are no such outside limitations, the most economical combination of diameter and height can be selected. Usual designs have flue gas velocities in the stack of about 30 feet per second.

FURNACE RATING PROCEDURE

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

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

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

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

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

Maximum heat release per unit firebox volume may be specified. Unit heat release is a rough measure of the probability of flame impingement on the tubes. However, current practice is to judge the possibility of flame impingement from the furnace shape and the type of burner, rather than to set an arbitrary heat release limit.

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

Preliminary Design Steps. Before starting the detailed rating, certain aspects of the design can be established from the process requirements and the other limitations described above. First, calculate the net heat release, using the specified absorption duty and efficiency. Then select an excess air percentage and determine the flue gas rate from Figure 8.

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

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

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

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

For the selected firebox dimensions, calculate the equivalent cold-plane surface, the effective refractory surface, and the mean beam length. Read the partial pressure of carbon dioxide plus water from Figure 3 and calculate the PL product.

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

Assume a value for the average temperature of the flue gas in the radiant section, t_g. At this temperature, read the gas emissivity from Figure 4, and then the exchange factor from Figure 5. Calculate the ratio $q_n/\alpha A_{cp} F$.

If the furnace design is one for which experience has shown a significant difference between average effective gas temperature and exit temperature, apply the appropriate correction to determine the exit temperature t_{ge}. Read the corresponding value of q_{ge}/q_n from Figure 7. Then, using Evuation 9, calculate $q_R/\alpha A_{cp} F$ and plot it against average furnace temperature on Figure 6.

If the assumed value of t_g was correct, the computed point will be on the absorption curve for the average tube wall temperature. Usually, of course, there will be a discrepancy. Then select another value of t_g on the other side of the absorption curve, and repeat the above procedure. When two points on opposite sides of the curve are obtained, join them by a straight line. The point of intersection of this line with the absorption curve is the correct firebox temperature. For this temperature, calculate t_{ge}, read q_{ge}/q_n from Figure 7, and calculate q_R from Equation 8b. This is the heat actually absorbed in the radiant section.

It is now necessary to check the computed radiant absorption to be sure it meets the design limitations. First, divide the heat absorption by the total exposed tube area to get the average heat flux. If it is higher

than the allowable maximum, a new furnace with more tube surface must be selected, and the rating repeated. If the actual heat flux is considerably below the allowable value, a smaller furnace should be considered. Then, from enthalpy data on the process fluid, calculate the crossover temperature from convection to radiant section. If the computed temperature is significantly different from that assumed in starting the rating, it may be necessary to go back and repeat with a new value of average tube wall temperature. Tube wall temperature, however, has such a small effect on the radiant absorption rate that such refinement is usually unnecessary.

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

Convection Section Rating. The convection section must pick up the difference between the heat absorbed in the radiant section and the total heater duty. The temperatures of the flue gas entering the convection section and the process fluid leaving it have already been calculated as part of the radiant section rating. The incoming fluid temperature is set by the original design conditions. The flue gas temperature going to the stack is determined by an over-all heat balance and Figure 7.

$$\frac{q_R}{q_a} = 1 + \left(\frac{q_a}{q_a}\right) + \left(\frac{q_f}{q_a}\right) - \left(\frac{q_R + q_C}{q_a}\right) \qquad (16)$$

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

Select a convection tube arrangement that will give a maximum flue gas mass-velocity of about 0.3 to 0.4 pounds per square foot per second. Calculate the mass-velocity G. For precise ratings, calculate the gas-film coefficient using Figures 9, 10 and 11, and Equations 11 and 12. Calculate the intube coefficient using any applicable method. Then calculate the over-all transfer coefficient from Equation 13. For less critical ratings, read the over-all design coefficient from Figure 12. In either case, finally determine the convection area from the usual heat-transfer equation

$$A_C = \frac{q_C}{U_C\,(LMTD)} \qquad (17)$$

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

Finally, calculate the flue gas pressure drop through the convection section, using Equation 14. Evaluate the flue gas density at the same average temperature that was used in calculating the transfer coefficients.

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

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

TABLE 2—Altitude Correction Factors for Gas-Density and Draft

Altitude Above Sea Level	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

Read the stack draft per 100 feet from Figure 14. Calculate the draft that the stack must produce, and the required height. If additional height is required for fume dispersal, the stack diameter may be reduced.

SAMPLE CALCULATION

The following example illustrates the use of the rating method in designing a typical refinery heater. Assume we are to design a box-type heater as shown in Figure 1 to heat 27,270 barrels per day of 26 API oil from 380° F to 675° F. The fuel is refinery fuelgas at 60° F, with a lower heating value of 1350 Btu/scf. Further assume that the user has specified an efficiency of 77 percent, no air preheat, 25 percent excess air, 12,000 Btu/hr, sq ft maximum radiant heat-flux, and 40 psi maximum pressure drop.

Density = 7.48 lb/gal at 60° F, 5.93 lb/gal at 530° F
Flow = 27,270 x 42 x 7.48/24 = 357,000 lb/hr
Enthalpy = 273 Btu/lb at 380° F, 483 Btu/lb at 675° F
Duty = 357,000 (483-273) = 75.0 MM Btu/hr
Net Heat release = 75.0/0.77 = 97.5 MM Btu/hr
Flue gas at 25 percent excess air
= 97.5 x 1010 = 98,500 lb/hr
= 27.4 lb/sec (Figure 8)
Approximate radiant duty = 0.7 x 75.0 = 52.5 MM Btu/hr
Crossover temperature = about 675 — 0.7 (675-380) = 468° F
Average fluid = (468 + 675)/2 = 572° F
Average tube wall = 572 + 100 = 672° F
Radiant surface = 52.5 x 10^6/12,000 = 4370 sq. ft.
Total surface = about 8700 sq. ft.
Select 4.5-inch OD tubes, 4 parallel passes
Velocity = 6.4 ft/sec
Pressure drop = 29 psi
Surface = 1.178 sq ft/ft
Total exposed radiant length = 4370/1.178 = 3710 ft.

Try a heater 20 feet wide by 25 feet high, with 96 tubes 40 feet long overall, 38.5 feet exposed, 8-inch centers. Estimate 6 shield tubes.

Surface per tube = 38.5 x 1.178 = 45.4 sq ft
Total surface = 96 x 45.5 = 4360 sq ft
Spacing ratio = 8.0/4.5 = 1.78, $\alpha = 0.92$ (Figure 2)
Shield tubes $A_{cp} = 38.5 \times 6 \times 8/12 = 154$ sq ft
Wall tubes $A_{cp} = 38.5 \times 90 \times 8/12 = 2310$ sq ft
$\alpha A_{cp} = 154 + 0.92 \times 2310 = 2280$ sq ft
Envelope area = $2(20 \times 25) + 2 \times 38.5\ (20 + 25)$
= 4470 sq ft
$A_R = 4470 - 2280 = 2190$ sq ft, $A_R/\alpha A_{cp} = 0.96$
Furnace volume = 20 x 25 x 38.5 = 19,250 CF
Mean beam length = $(2/3)(19{,}250)^{1/3} = 17.9$ ft
Partial pressure, $CO_2 + H_2O$, = 0.23 atm (Figure 3)
PL = 0.23 x 17.9 = 4.1 atm-ft

Since there is no preheated air or fuel, $q_a = q_f = 0$. Assume $q_L/q_n = 0.02$. Estimate that the average firebox temperature equals 1500° F.

Emissivity = 0.54 (Figure 4)
Exchange factor = 0.645 (Figure 5)
$\alpha A_{cp} F = 2280 \times 0.645 = 1470$ sq ft
$q_n/\alpha A_{cp} F = 97.5 \times 10^6/1470 = 66{,}400$ Btu/hr, sq ft
For this type of furnace, assume that the exit gas temperature is the same as the average firebox temperature
$t_{ge} = t_g = 1500°$ F
$q_{ge}/q_n = 0.415$ (Figure 7)
$q_R/\alpha A_{cp} F = (1 + 0 + 0 - 0.02 - 0.415)\ 66{,}400$
$= (0.565)\ 66{,}400 = 37{,}500$ Btu/hr, sq ft
(Equation 9)

Figure 15 shows a small portion of Figure 6 with the absorption curve for 672° F tubewall. Plot the calculated $q_R/\alpha A_{cp} F$ of 37,500 at the assumed firebox temperature of 1500° F. This lies to the left of the absorption curve. So try another calculation at 1700° F.

Emissivity = 0.525
$F = 0.635$
$\alpha A_{cp} F = 1450$
$q_n/\alpha A_{cp} F = 67{,}300$
$q_{ge}/q_n = 0.48$
$q_R/\alpha A_{cp} F = (0.98 - 0.48)\ 67{,}300 = 33{,}600$

Plot this point on Figure 15, and join the two calculated points by a straight line. It crosses the absorption curve at 1610° F. This is the firebox temperature.

$q_{ge}/q_n = 0.45$
$q_R = (0.98 - 0.45)\ 97.5 = 51.7$ MM Btu/hr
$q_R/A_{Rt} = 51.7 \times 10^6/4360 = 11{,}850$ Btu/hr, sq ft

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

Enthalpy change = $51.7 \times 10^6/357{,}000 = 145$ Btu/lb
Crossover enthalpy = 483 — 145 = 338 Btu/lb
Temperature = 483° F
Average radiant fluid = $(483 + 675)/2 = 579°$ F
Average tube wall = 579 + 100 = 679° F

The calculated tube wall is so close to the assumed value of 672° F that no further correction is necessary.

For the convection-section design, calculate the various temperatures.

Convection section duty = 75.0 — 51.6 = 23.4 MM Btu/hr
Stack heat content/release = 1 — 0.02 — 0.77 = 0.21 (Equation 16)
Stack Temperature = 820° F (Figure 7)

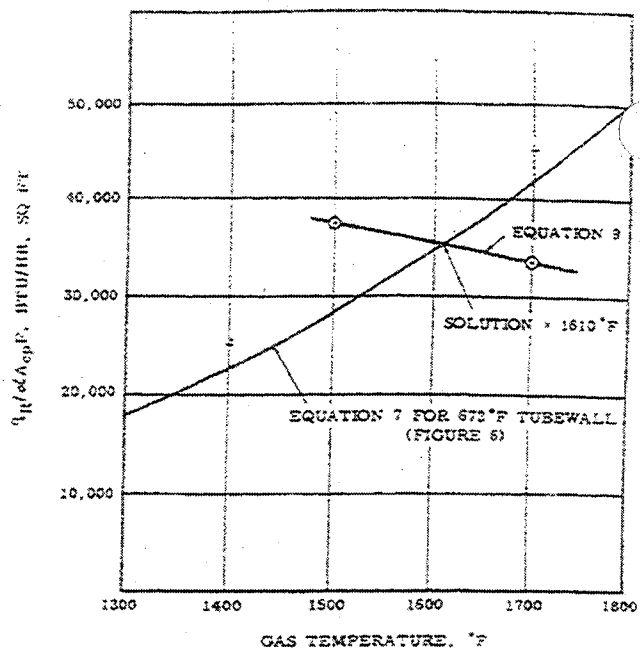


FIGURE 15—Example of graphical solution for flue gas temperature in the radiant section.

Temperature differences, hot = 1610 — 483 = 1127° F
cold = 820 — 380 = 440° F
log mean = 731° F

Average temperatures

Oil	$(380 + 483)/2 = 432°$ F
Tube wall	432 + 100 = 532° F
Gas	432 + 731 = 1163° F
Gas film	432 + 731/2 = 798° F

Select a convection section with six 4.5-inch OD tubes per row, 8-inch spacing, staggered.

Gross width = 6.5 x 8 = 52 inches
Free width = 52 — 6 x 4.5 = 25 inches = 2.08 ft
Area = 2.08 x 38.5 = 80 sq ft
G = 27.4/80 = 0.342 lb/sq ft, sec

Using the detailed film transfer correlations

$h_{ce} = 4.5$	(Figure 9)
$H_{cr} = 2.2$	(Figure 10)
$h_{cv} = 9.3$	(Figure 11)
Total 16.0	

Row-to-row tube spacing = 0.866 x 8/12 = 0.58 ft
Wall area per row = 2 x 0.58 x 38.5 = 44.6 sq ft
Tube area per row = 6 x 45.4 = 272 sq ft

$$f_t = \frac{9.3}{16.0} \times \frac{44.6}{272} = 0.095 \qquad \text{(Equation 11)}$$

$$h_c = 1.095\ (4.5 + 2.2) = 7.3 \qquad \text{(Equation 12)}$$

For this example, assume the calculated intube film coefficient is 100

$$U_c = \frac{100 \times 7.3}{100 + 7.3} = 6.8 \qquad \text{(Equation 13)}$$

$$A_c = \frac{q_c}{U_c\ (LMTD)} = \frac{23.4 \times 10^6}{6.8 \times 731} = 4{,}700 \text{ sq ft} \qquad \text{(Equation 17)}$$

Tubes = 4,700/45.3 = 104 tubes
Use 18 rows = 108 tubes = 4,900 sq ft

For comparison, the approximate over-all coefficient

from Figure 12 is 7.2. It is also worth noting that the most laborious part of computing the total gas-film coefficient is determining the relatively small wall-effect correction. For many problems, an estimated value of about 10 percent is accurate enough.

In order to get an integral number of tube rows, and yet divide the four streams equally, it was necessaary to include 4 percent additional convection surface. This means that the heater should be slightly more efficient than the 77 percent specified. Repeating the rating procedure, assuming 78 percent efficiency, gives a convection section requirement of 111 tubes. By interpolation, the efficiency with 108 tubes is 77.6 percent, and the stack temperature is 800° F.

The total number of tubes is 96 radiant and shield, plus 108 convection, or 51 tubes per pass. With this number of tubes, the pressure drop is 32 psi, still within the specified limit of 40 psi.

Finally the stack size must be determined. First, determine the draft effect of the firebox. Assume sea level elevation, 100° F outside air

Flue gas temperature = 1610° F
Draft = 1.01 inches/100 ft (Figure 14)
Estimate shield tubes are 18 ft above burners
Draft = 1.01 x 18/100 = 0.18 inch

Assuming the burners require 0.25 inch, they are limiting. At the shield tubes, the draft must be 0.07 inch

Convection section G = 0.342
Density = 0.0236 lb/CF (Figure 13)
Velocity head = 0.0030 x (0.342)²/0.0236 = 0.0149 inch (Equation 14)
Friction = 18 x 0.0149/2 = 0.13 inch
Stack temperature = 800 — 100 = 700° F
Density = 0.0330 lb/CF (Figure 13)
Gas volume = 27.4/0.0330 = 830 CF/sec

Try a 6-foot ID stack, which gives a velocity of 29.4 ft/sec.

Velocity head = 0.0030 x (29.4)² x 0.0330 = 0.086 inch (Equation 14)
Say height = 100 ft

L/50 D = 100/(50 x 6) = 0.3
Loss = (3.0 + 0.3) x .086 = 0.28 inch
Total draft required = 0.07 + 0.13 + 0.28 = 0.48 inch
Draft from stack = 0.74 inch/100 ft (Figure 14)
Height = (0.48/0.74) x 100 = 65 ft above shield tubes

In practice, the stack height would probably be increased above the calculated value to allow for unforeseer variations in operating conditions.

About the Author

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Wimpress

NOMENCLATURE

A	Area, sq ft
A_{cp}	Cold plane area, sq ft
A_w	Effective refractory area, sq ft
D	Diameter, ft
d	Diameter, inches
F	Radiant exchange factor
f	Factor for wall-radiation in convection section
G	Mass velocity at minimum cross section, lb/sq ft, sec
g	Gravitational constant, 32.17 ft/sec²
H	Stack height, ft
h	Film heat transfer coefficient, Btu/hr, sq ft, °F
L	Mean length of radiant beam, ft
LMTD	Log-mean temperature difference, °F
P	Partial pressure of carbon dioxide plus water in the flue gas, atmospheres
p_v	Velocity-head, inches of water
Δp	Pressure drop, inches of water
q	Heat rate, Btu/hr
T	Temperature, °R
t	Temperature, °F
U	Over-all heat transfer coefficient, Btu/hr, sq ft, °F
V	Velocity, ft/sec
W_B	Radiant emission from a black body, Btu/hr, sq ft
α	Factor of comparison between a tube-bank and a plane
ρ	Density, lb/cu ft
ρ_a	Density of outside air, lb/cu ft
ρ_g	Density of flue gas, lb/cu ft
ρ_w	Density of water, lb/cu ft
σ	Stefan-Boltzman constant, 0.173 x 10⁻⁸ Btu/hr, sq ft, °R⁴

SUBSCRIPTS

a	Combustion air
C	Convection section
c	Convective heat-transfer
f	Fuel or film
g	Flue gas
i	Intube fluid
n	Net heat of combustion
R	Radiant section
r	Radiant heat-transfer
t	Tube or tube wall
w	Refractory wall
1	Inlet conditions
2	Outlet conditions

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Generalized method predicts fired-heater performance

Direct-fired process heaters make up a large part of many plants' initial cost. Their fuel also represents a major operating expense. Both builders and operators of process plants, therefore, need a reliable rating method.

Norman Wimpress, C. F. Braun & Co.

☐ Purchasers and users frequently must make their own ratings of fired heaters in order to:

- Estimate heater sizes, fuel consumption, and waste heat recovery, during the conceptual design-stage.
- Evaluate vendors' offerings with regard to reasonableness of designs and consistency between competing proposals.
- Predict the effects of changes in feedrates, feed properties and similar operating variables.
- Anticipate the effects of proposed modifications to an existing heater.

Final design is usually done by the heater manufacturer, following the purchaser's duty specification. In general, each manufacturer has its own rating method for its particular design. Such a method can be expected to be more accurate for its range of application than any general procedure.

Suggested method validated by experience

The method proposed for computing the performance of both the radiant and convection sections of fired heaters is based on fundamental correlations for heat transfer by radiation and convection, adjusted on the basis of operating experience with industrial heaters. Experience has shown the method to be well suited for the type of calculations outlined.

Fig. 1 shows a cross-section of a typical process-plant fired heater. It consists of a firebox or radiant section, a convection section, and a stack system to dispose of the combustion products and provide draft.

The radiant section, where the fuel is burned, contains heat-absorbing tubes, which remove a portion of the heat from the combustion products before they flow to the convection section. In this type of furnace, the tubes are located around the outside of the firebox (in front of refractory walls), combustion taking place in the open central space.

There are also designs in which the tubes are suspended in the center of the furnace, with combustion space on both sides. In these furnaces, the burners may be arranged to fire against refractory walls so as to enhance radiant heat transfer to the tubes. In any case, the tubes must be arranged for uniform and efficient radiant heat absorption, and there must be adequate volume for complete combustion without local overheating or flame impingement on the tubes.

The convection section recovers additional heat from the flue gas at a lower temperature than in the radiant section. Here, since the primary heat-transfer mechanism is convection, the tubes are arranged to create high mass velocities and turbulence in the gas. Tubes having fins and other types of extended surface are frequently installed to improve convective heat transfer.

The first rows of tubes in the convection section, which are exposed to radiation from the hot gas and refractory in the radiant section, are generally called shield or shock tubes. They are usually a structural part of the convection section and are rated with it. However, special allowance must be made for the radiant heat added to these tubes.

The stack system collects and disposes of the flue gas. In natural-draft furnaces, the stack height must provide adequate draft to draw the gas through the firebox and convection section. In induced-draft furnaces, stack height is usually set by gas-dispersion requirements.

Because the governing heat-transfer mechanism is different in the radiant and convection sections, the two sections are rated by different methods.

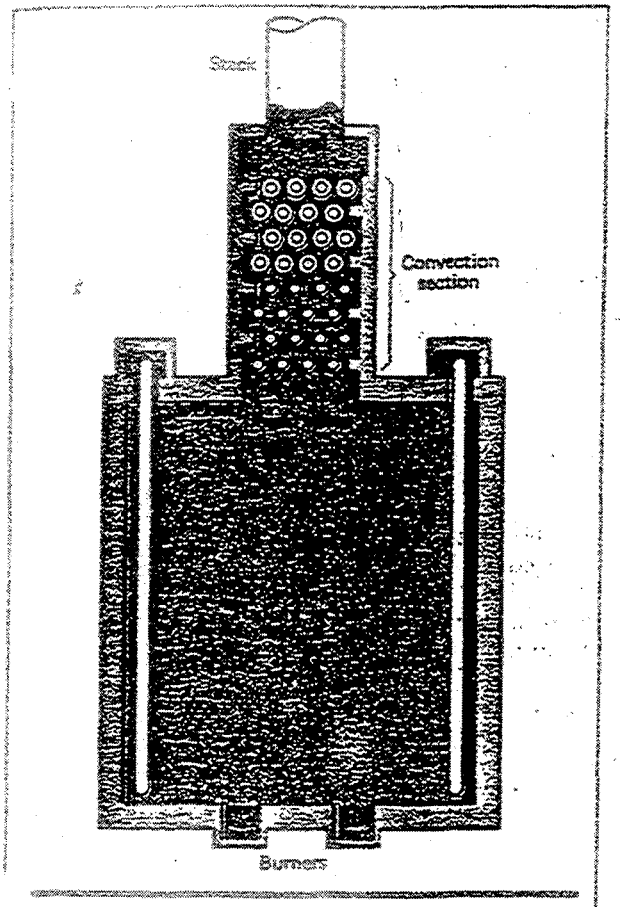
Radiant-section design

Radiant heat transfer between solid surfaces in various arrangements, and between hot gases and solids, has been extensively investigated [*1,2*]. Applying basic radiation concepts to process-type heater design, Lobo and Evans developed a generally applicable rating method [*3*]. Almost all methods published since have followed their basic approach, as does the method described here (with simplifications made by eliminating minor variables and by including general correlations).

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Typical fired heater is shown in cross-section Fig. 1

Nomenclature

A	Area, m^2
A_{cp}	Cold-plane area, m^2
A_e	Total furnace envelope-area, m^2
A_t	Tube surface area, m^2
F	Radiant exchange factor
G	Mass velocity at minimum cross-section, $kg/(m^2)(s)$
h	Film heat-transfer coefficient, $kcal/(h)(m^2)(°C)$
L	Mean length of radiant beam, m
P	Partial pressure of radiating components, atm
p_v	Velocity head, mm H_2O
q	Heat rate, kcal/h
T	Temperature, °K
V	Volume, m^3
W_B	Radiant-emission from a black body, $kcal/(h)(m^2)$
α	Factor of comparison between a tube bank and a plane
ρ	Density, kg/m^3
σ	Stefan-Boltzmann constant, 4.92×10^{-8} kcal/$(h)(m^2)(°K^4)$

Subscripts

a	Combustion air
c	Convective heat transfer
f	Fuel
g	Flue gas
L	Losses
n	Net heat of combustion
R	Radiant section
r	Radiant heat transfer
S	Shield- and convection-tube radiant transfer
t	Tube surface

Radiant heat transfer is basically described by the Stefan-Boltzmann equation. A black body at absolute temperature T radiates energy at a rate W_B:

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

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

$$q_r = \sigma A F (T_a^4 - T_b^4) \qquad (2)$$

Here, A is the area of one of the surfaces, and F is an exchange factor that depends on the relative area and arrangement of the surfaces, and on the emissivity and absorptivity of each. For transfer inside a furnace, it is generally best to use the heat-receiving, or "cold," surface as the basis for calculation.

Simplify with equivalent cold-plane surface

The usual heat-absorbing system consists of a number of parallel cylindrical tubes. Part of the radiation from the hot gas strikes the tubes directly and is absorbed, and the remainder passes through. If the tubes are in front of a refractory wall, the energy that passes through is radiated back into the furnace, where part of it is absorbed by the tubes, and the remainder passes through.

This complicated situation is handled by expressing the tube area as an equivalent plane surface, A_{cp}. It equals the number of tubes, times their exposed length, times the center-to-center spacing. Because the tube bank does not absorb all the energy radiated to the cold-plane area, an absorption efficiency factor, α, must be applied. Values for α as a function of tube arrangement and spacing have been developed and published by Hottel (in "Chemical Engineers' Handbook"), whose curves for the most-common tube arrangements in process furnaces are shown in Fig. 2.

The product αA_{cp}, called the equivalent cold-plane area, is the area of a plane having the same absorbing capacity as the actual tube bank.

In the type of heater shown in Fig. 1, the convection section forms part of the firebox enclosure. Because the convection section is a number of rows deep, it eventually absorbs all the radiation coming to it from the firebox. Therefore, α for the convection section equals unity. The working equation for calculating radiant heat transfer to the tubes thus becomes:

$$q_{R-r} = \sigma \alpha A_{cp} F (T_g^4 - T_t^4) \qquad (3)$$

Determining the exchange factor

The remaining term to be evaluated in Eq. (3) is the exchange factor, F. The gas in the firebox is a poor radiator, because the only constituents normally in flue gas that contribute significantly to the radiant emission are carbon dioxide and water. The amount of radiating

components can be expressed by a single term, the partial pressure of carbon dioxide plus water multiplied by the mean beam length [3]. Fig. 3 shows the partial pressure, P, as a function of excess air for the usual hydrocarbon fuels. For the less-usual fuels, such as hydrogen, P can readily be calculated by simple stoichiometry. The mean beam length, L, is calculated by the equation:

$$L = 3.6\,V/A_e \qquad (4)$$

Here, V is the total firebox volume inside the centerline of tubes, and A_e is the total firebox envelope area.

Gas emissivity is also a function of the temperatures of the gas and the absorbing surface. However, because the tubewall temperature effect has been found to be minor, gas emissivity can be correlated as a function of the product of PL, and of the gas temperature (Fig. 4).

The exchange factor also depends on the amount of radiation reflected from exposed refractory. (Energy striking refractory is reflected back toward the tubes, where it has a second chance to be absorbed.) Thus, a furnace having a large amount of exposed refractory will transfer more heat per unit of tube surface than one whose walls are covered by tubes.

Lobo and Evans correlated this effect on the basis of the ratio of exposed refractory area to total equivalent cold-plane area. Fig. 5 is based on the same correlation, except that the ratio is of cold-plane area to total firebox envelope-area, which simplifies the calculations. Fig. 5 also takes into account that the tubes themselves are not perfect absorbers. The curves are based on a tube-surface absorptivity of 0.9, which is typical for oxidized metal surfaces.

Estimating convection transfer

Although radiation accounts for most of the heat transfer in the radiant section, convection cannot be neglected. The relationship for convective heat transfer is:

$$q_c = h_c A_t (T_g - T_t) \qquad (5)$$

Because convective heat transfer is not the major contributor and cannot be calculated precisely, some simplifying approximations are made. For the usual furnace, h_c in the radiant section is about 10 kcal/(h)(m²)(°K), A_t is about two times αA_{cp}, and F is about 0.57. This allows putting Eq. (5) into a form similar to Eq. (3):

$$q_{Rc} = (10)(2\alpha A_{cp})(F/0.57)(T_g - T_t) = 35\alpha A_{cp} F \qquad (6)$$

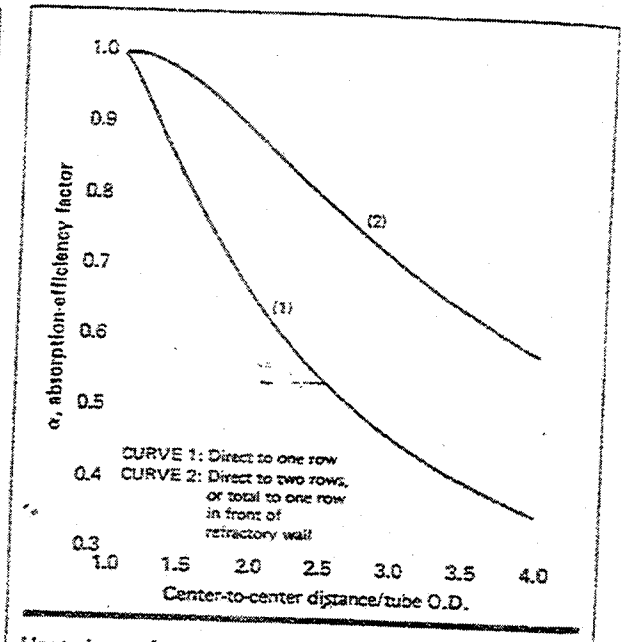
The total heat-transfer rate in the radiant section is the sum of the radiant and convective heat transfer:

$$q_R = \sigma \alpha A_{cp} F (T_g^4 - T_t^4) + 35\,\alpha A_{cp} F (T_g - T_t) \qquad (7)$$

$$\frac{q_R}{\alpha A_{cp} F} = \sigma (T_g^4 - T_t^4) + 35\,(T_g - T_t) \qquad (8)$$

Thus, the ratio $q_R/\alpha A_{cp} F$ is a function of gas and tubewall temperatures only (Fig. 6).

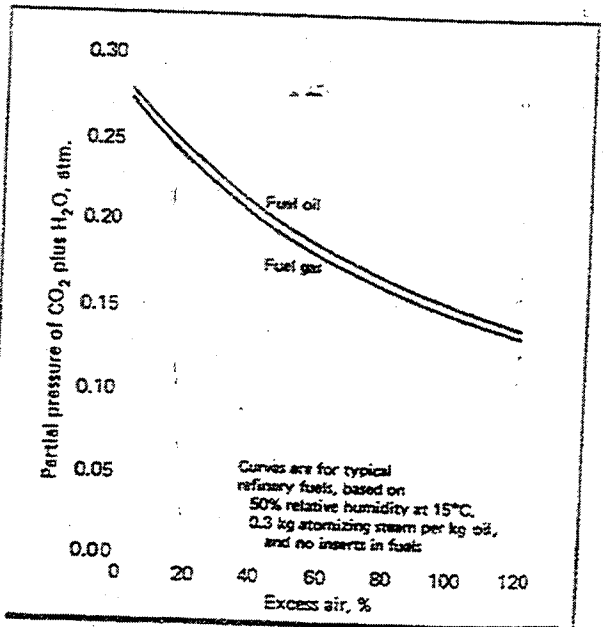
Eq. (7) actually applies only to the tubes in the radiant coil and not to radiation to the convection section. Convective transfer to the convection section tubes, which is calculated separately, does not occur until after the flue gas leaves the firebox, so radiation to the convection section should be calculated by Eq. (3). Computer calculations can be easily programmed to maintain this distinction. For hand calculations, however, it is simpler to use Fig. 6 for all tubes, the error introduced usually being negligible.



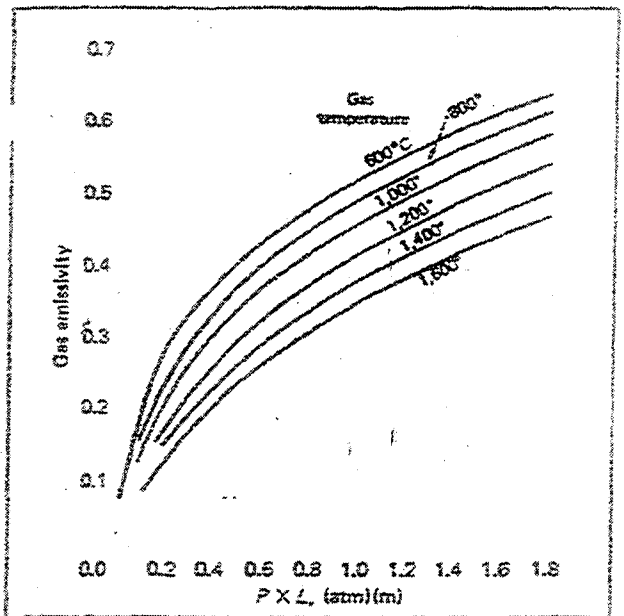
Heat absorption varies with tube arrangement Fig. 2

Tubes mounted centrally

For the foregoing discussion, the furnace is assumed to be of the type shown in Fig. 1, with refractory-backed tubes around the periphery of the firebox. Certain ad-

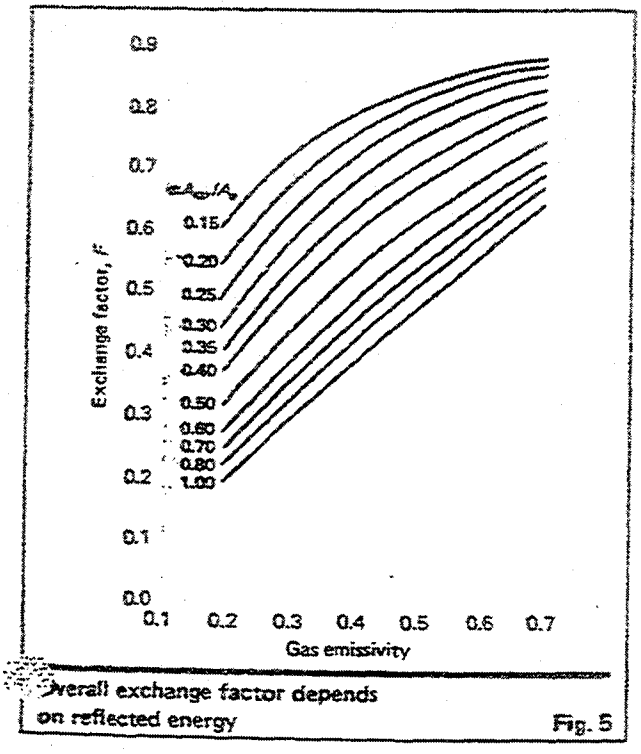


Excess air sets CO_2-H_2O pressure in flue gas Fig. 3



Flue-gas emissivity depends on absorbing surface Fig. 4

justments are necessary when the tubes are arranged as in Fig. 7, with a single or double row of centrally mounted tubes. In such cases, advantage is taken of the fact that a plane of symmetry in the firebox can be replaced, for computational purposes, by a refractory wall (indicated by dashed lines in Fig. 7). Therefore, the ld-plane area, A_{cp}, for the tube bank is twice the projected cold-plane area, because each side is figured separately. On this basis, α for a single row is obtained



Overall exchange factor depends on reflected energy Fig. 5

from Curve 1, and for a double row from Curve 2, in Fig. 2. Similarly, the mean beam length, L, must be calculated on the basis of half a furnace.

Temperature complexities

Fig. 6 gives the rate of heat transfer between a mass of gas at one uniform temperature and a tube surface at another uniform temperature. In most actual furnaces, however, such is not the case. Average effective temperatures must be selected in order to use Fig. 6.

Tube-wall temperature depends on the temperature of the fluid and its transfer coefficient inside the tube, the thermal resistance of the tube wall, and the total heat flux. At tube-wall temperatures below 500°C, when the radiant absorption rate is fairly insensitive to the receiving temperature, it is usually acceptable to use the average fluid temperature plus 50°C. At higher temperatures, more-detailed calculations are often necessary. In installations, such as pyrolysis furnaces, having extremely high tube-wall temperatures, it is advisable to divide the receiving area into zones of different average temperature, and to calculate each zone's heat-absorption rate separately.

Because about 70% of the radiation to the convection section is received by the front row of tubes, the average tube-wall temperature for these tubes should be used to calculate convection-section radiation.

Depending on the type of furnace, there may also be considerable variation in flue-gas temperature within the firebox. However, because of the large amount of turbulent mixing that occurs, the transparency of the radiating gas, and the effect of secondary radiation from exposed refractory, it is very difficult to quantitatively allow for these temperature variations when making the heat-flux calculations. In the absence of actual physical barriers between zones, it appears to be better to base all heat-transfer calculations on a single effective radiating temperature throughout the firebox.

Heat balance provides firing rate

The aforementioned procedure enables calculating the firebox temperature necessary to transfer a specific amount of heat into a specified radiant-section coil. The next step is to determine the firing rate necessary to maintain that temperature. This is done through a heat balance around the firebox.

Heat is put into the radiant section from three primary sources: (1) the net heat of combustion, q_n; (2) the sensible heat of the combustion air, q_a; and (3) the sensible heat of the fuel and any atomizing steam, q_f. Heat is removed via absorption by the radiant-section tubes (q_R), radiation to the shield and convection tubes (q_S), casing losses (q_L), and sensible heat of the exiting flue gas (q_{g2}):

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

The terms q_a and q_f, being generally proportional to the amount of fuel burned, can be expressed as ratios to q_n. Similarly, the loss q_L is usually taken as 1 to 3% of the net heat release, depending on the furnace design and experience. Finally, the fraction of net heat release remaining in the flue gas is a function of fuel composition, flue-gas temperature, and excess air. For the com-

mon hydrocarbon fuels, the ratio q_g/q_n can be correlated on a single set of curves, as in Fig. 8. On this basis, Eq. (9) can be rearranged to allow direct calculation of q_n:

$$q_n = \frac{q_R + q_S}{1 + \frac{q_a}{q_n} + \frac{q_f}{q_n} - \frac{q_L}{q_n} - \frac{q_{s2}}{q_n}} \qquad (10)$$

The last function required to close the computational loop around the radiant section is the relationship between the effective gas radiating-temperature, T_g, and the exit flue-gas temperature, T_{g2}. For box-type heaters having an approximately square cross-section, and no areas of refractory with direct flame impingement, the two temperatures may be assumed equal.

In the opposite extreme of a high-temperature furnace having a tall, narrow firebox with wall-mounted radiant burners, T_g may be 100 to 150°C higher than T_{g2}. Other types, such as narrow, bottom-fired vertical cylindrical heaters, will fall somewhere in between. The magnitude of the difference must be determined empirically from experience with similar designs.

Convection-section design

The relative importance of the convection section in fired-heater design has increased markedly in recent years. One reason is the higher cost of fuel, which has resulted in setting furnace-efficiency targets far higher than could formerly be economically justified. Another major factor is the development of chemical conversion processes that demand extremely high heat-input fluxes, and correspondingly high firebox temperatures.

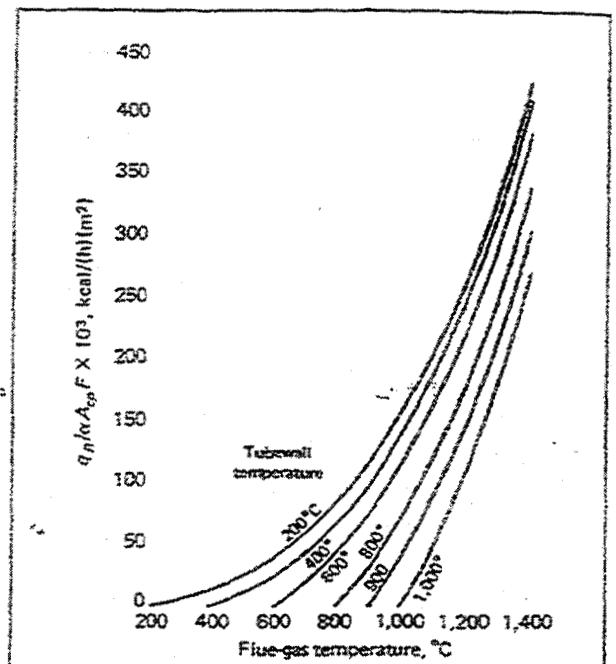
The result is a smaller fraction of the total heat released being removed in the radiant section, and a correspondingly greater convection-section loading. Whereas typical heat-distribution formerly was roughly 50% radiant section, 20% convection section, and 30% losses, it now may be, respectively, 40%, 50% and 10%.

Thus, there has been a trend toward larger convection sections and more use of extended surface. Also, there are more multiservice convection sections that—via steam generation, feedwater heating and similar extraneous services—take up heat that cannot be absorbed in the primary process. The result has been to make convection-section design more difficult and more critical.

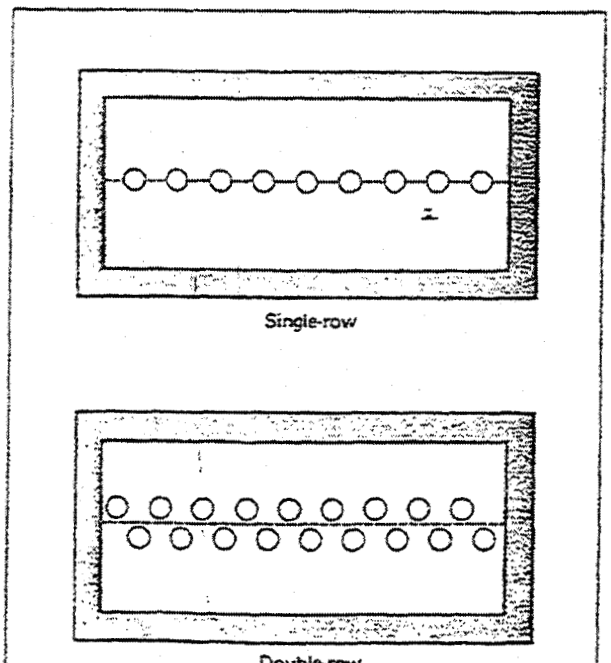
Transfer coefficients depend on tube type

As in the radiant section, heat is transferred in the convection section by both radiation and convection. The classical basis for calculating convection-section heat transfer was developed by Monrad [4], who took into account direct convection, radiation from the gas, and radiation from the refractory walls. Monrad's basic method has been modified in light of later experimental results and adapted to extended-surface tubes by Schweppe and Torrijos [5].

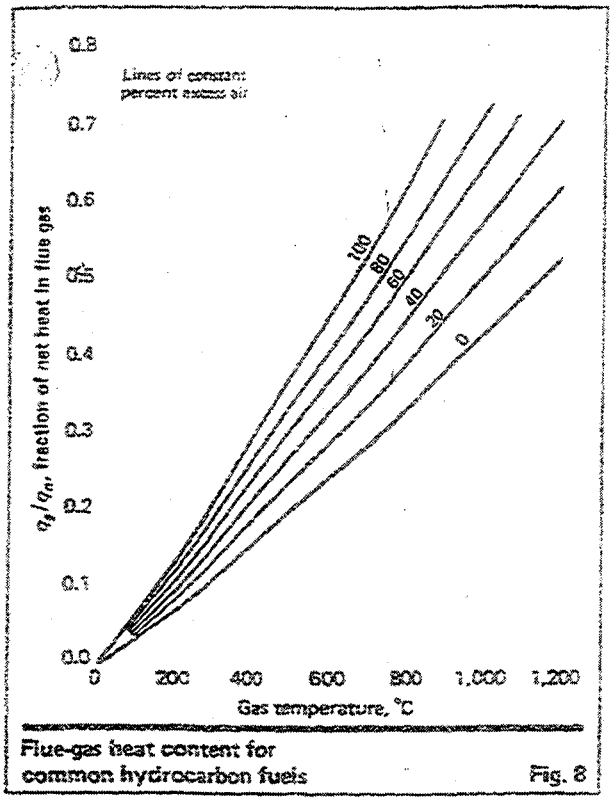
Because of the previously mentioned trend toward increased convection-section heat recovery, most present-day heater designs incorporate extended-surface tubes. Because of the many types and sizes of extended surface available, it is not practical to present in this article a specific procedure for calculating coefficients. (For such information, refer to the Schweppe and Torrijos article, as the present article is limited to general principles of design.)



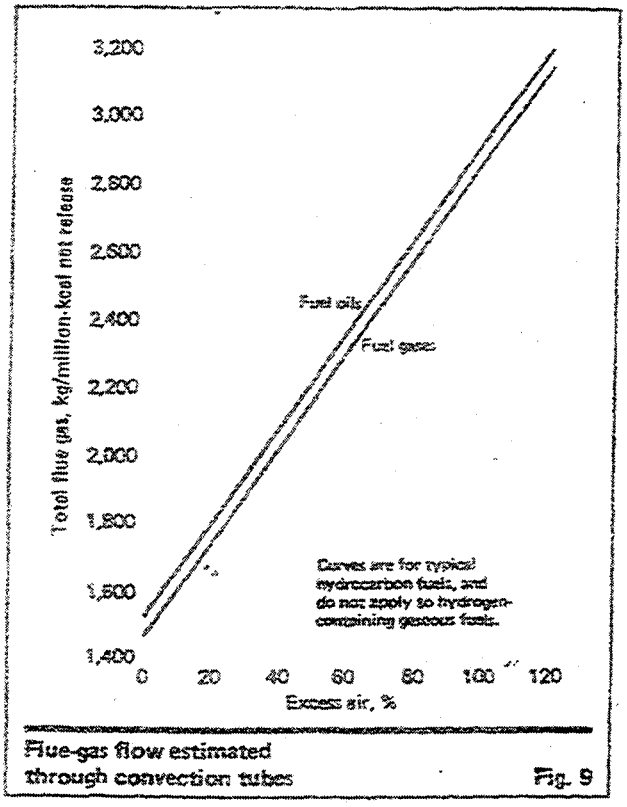
Temperatures set heat-absorption rate in radiant section — Fig. 6



Dashed lines indicate the refractory wall assumed for calculations — Fig. 7



Flue-gas heat content for common hydrocarbon fuels Fig. 8



Flue-gas flow estimated through convection tubes Fig. 9

In any case, the amount of flue gas flowing through the convection tube-bank must be known. As with the flue-gas heat content, the flue-gas quantity can be calculated from stoichiometric relationships involving the fuel consumption, the fuel heating value, and the amount of excess air. For normal hydrocarbon fuels, the quantity can be estimated closely from the net heat release and the excess air, as in Fig. 9.

For bare tubes, the inside-tube heat-transfer resistance is usually low enough that a reasonable estimate is satisfactory, at least for preliminary design. Fig. 10 presents calculated overall heat-transfer coefficients for typical sizes of bare tubes. These curves were calculated from the data of Schweppe and Torrijos, assuming usual staggered-tube arrangements, fluid temperatures, and inside-tube film coefficients. The average gas temperature used in this figure is the arithmetic average inside-tube fluid temperature plus the log-mean-temperature difference from flue gas to fluid. The mass velocity is that through the minimum cross-section of the tube bank.

Having established the duty and overall heat-transfer coefficients for each service in the convection section, one calculates the required surface via conventional heat-balance and heat-exchange procedures. For normal fuels, Fig. 8 can be used to establish flue-gas temperatures throughout the section.

Radiation from the firebox

In rating the radiant section, a quantity, q_S, representing the heat radiated directly from the firebox to the convection section, was calculated. This quantity of heat must be added to that derived by the convection-section calculations, to determine the total heat put into the lower tubes in the convection section. Because almost all the radiant heat is absorbed in the first two rows of the convection section, it is usually not necessary to make any detailed breakdown of the distribution.

Actual furnace performance

Some caution must be exercised when using published heat-transfer coefficients for industrial heater design. Experimental coefficients are usually measured under ideal conditions, with uniform flow distribution and no bypassing. Actual furnaces deviate from the ideal in some respects, as follows:

In tall convection sections, there can be considerable bypassing of hot flue gas through the tube header-boxes. Particularly with extended-surface tubes, there is often significant leakage area through the tubesheet holes. Under the driving force of the pressure drop through the convection section, hot flue gas flows out from the lower part into the header box, up through the box, and back into the upper convection section. This bypassing seriously undermines the overall performance of the convection section. Careful sealing of tubesheet holes, and in some cases partitioning of the header boxes, is necessary to avoid this difficulty.

Flow distribution may be far from uniform, particularly with long convection-section tubes. Flow is most commonly excessive in the center, near the stack opening, and is nearly stagnant in areas near the ends. Ample hood area above the convection tubes, and careful attention to flow patterns entering the section, are helpful in avoiding this problem.

Because of these and similar effects, some safety fac-

Altitude corrections for gas density and stack draft

Height above sea level, m	Correction factor
0	1.000
500	0.942
1,000	0.887
1,500	0.835
2,000	0.785
2,500	0.738
3,000	0.694

Multiply sea-level density, Eq. (12), and stack draft, Fig. 11, by the correction factor.

tor should be applied when heat-transfer coefficients are calculated from laboratory data. This is particularly true for extended-surface designs. More reliance can usually be placed on good test data taken from actual furnaces.

Stack design

The stack system must create sufficient draft to draw the flue gases through the radiant and convection sections, and must discharge the gases at a suitable height. This is usually done in process furnaces by natural or induced draft.

Radiant-section draft

A pressure drop across the burners is usually required to draw in the combustion air. Its magnitude depends on the type of burner and the fuel, and is typically about 6 mm of water.

All sections of the furnace generally must also be below atmospheric pressure, so that flows through peepholes, tube openings and the like will be outside air moving in, rather than hot gas going out. A normal design point is about 2 mm negative pressure at the inlet to the convection section. In most modern heaters that have tall radiant sections, this provides more than enough draft for the burners.

Velocity head in the convection section

Pressure-drop calculations through the convection section and stack system are conveniently made in terms of velocity head. The velocity head in millimeters of water is given by:

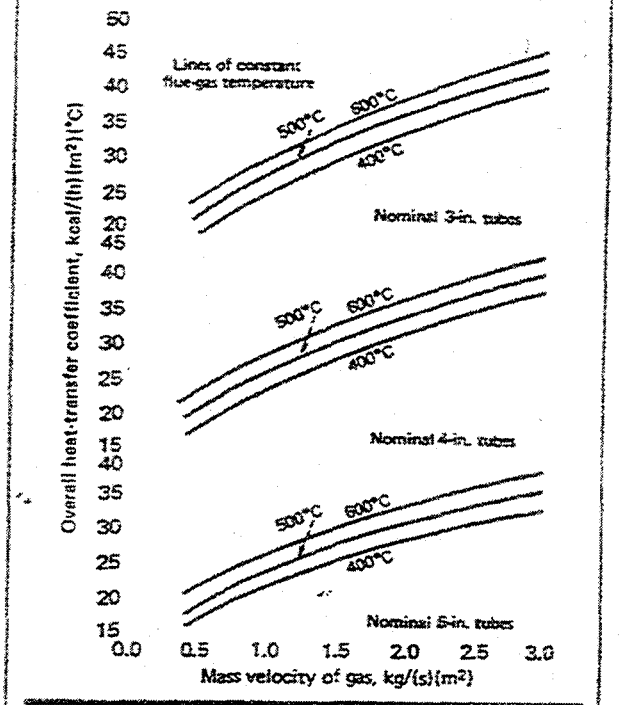
$$p_v = 0.051\, G^2/\rho_g \qquad (11)$$

Here, G is the mass velocity in kg/(m²)(s), and ρ_g is the gas density in kg/m³. If the flue-gas composition is known, its density can be readily calculated from the usual gas laws. Flue-gas density is relatively insensitive to fuel composition and amount of excess air. For the usual situation at sea level, the following equation is applicable:

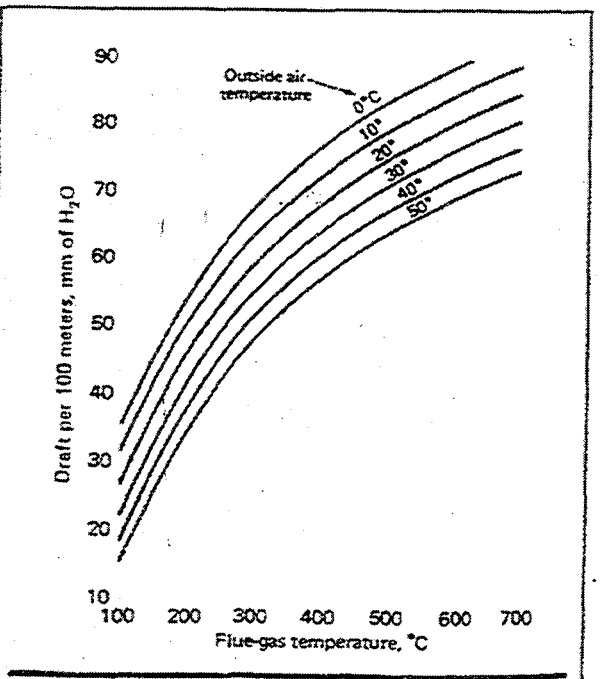
$$\rho_g = 342/T_g \qquad (12)$$

Here, T_g is the gas temperature in °K.

For banks of bare tubes, the frictional pressure drop is about one half of the velocity head per row. For extended-surface tubes, it is best to rely on the manufacturer's data, or correlations developed specifically for the particular configuration. In the absence of such information, the generalized relation of Schweppe and Torrijos is serviceable.



Overall coefficients for convection-section tubes — Fig. 10



Stack draft is indicated per 100 meters of height — Fig. 11

Stack and damper head-losses

Remaining losses may be estimated as follows:

Source	Velocity head-loss
Stack entrance	0.5
Damper	1.5
Stack and ducts	1.0/50 dia.
Stack exit	1.0

Other losses may be similarly calculated from the usual published loss-coefficients.

Stack draft

Draft depends on the difference in density between the hot flue gas and the surrounding air. The molecular weight of the flue gas is quite insensitive to fuel composition, and is about 28.5 for the usual hydrocarbon fuels. On this basis, draft per 100 m of height is shown in Fig. 11. Note that tall convection sections have their own draft, for which allowance should be taken in the computations.

When calculating available draft, some allowance should be made for lower stack temperatures due to heat loss, air leaking in, and the like. The amount depends on the flue-gas temperature and the degree of insulation of the stack and ductwork. For an unlined stack, and flue gas at about 400°C, a reasonable allowance would be 50°C.

Most process plants are close to sea level. However, for installations above 300 m elevation, decreased atmospheric pressure must be taken into account. Lower pressure increases the volume of gas, and therefore the pressure drop, through the furnace and stack system. At the same time, it decreases the draft because of the smaller density differences. Corrections for gas density and stack draft are given in the table.

Overall furnace rating

Design of an actual furnace requires combining the various heat-exchange and flue-gas heat-content relationships with the specified flows, temperatures and heat duties for the process fluids. This must be done for each stream and for each section of the heater. In almost all cases, too many factors are involved to permit vigorous simultaneous solution of all the equations. Instead, an iterative procedure with successive approximations must be followed.

The approach to any specific problem depends on the type of heater under consideration, and on the purpose of the calculation. For example, the procedure for rating a vendor's offering will not necessarily be the same as that for checking an initial design. Still another approach would be necessary to estimate the effect of modifications to an existing heater.

The preceding correlations and procedures have been presented on the basis of performing the calculations by hand. This is to permit the occasional user to obtain valid results with a reasonable expenditure of time and effort. The many repetitive calculations, however, suggest the advantages of computerization, particularly for anyone who handles a significant number of heater design problems.

In addition to speeding up the iterative calculations, computerization allows substituting exact calculations fitted to the specific situation for some of the generalized correlations. For example, Fig. 3, 8 and 9 are based on the most common liquid and gaseous fuels. In a computerized operation, it is quite practical to calculate the flue-gas composition, amount, and heat content for the specific fuel and the specific combustion conditions. In this way, abnormal fuels or other unusual conditions can be accommodated.

When developing such a program, procedures should be included so that a heater can be designed to meet specified performance criteria, or so that the performance of a fixed design can be estimated for various operating conditions. Beyond that, there are a number of options that might be included, depending on individual needs. Typical requirements are the ability to handle multiple fuels, gas-turbine exhaust, combustion-air preheat, and extraneous heat-recovery streams in the convection section. The degree to which these variations can be accommodated is a major factor in determining the complexity of the program.

Computer programs of this type are valuable for handling the numerous repetitive calculations involved in optimizing a new plant design, for confirming the validity of vendors' proposals, and for predicting the effect of different feedstocks or operating conditions on the performance of existing heaters.

There are, of course, many factors in addition to the rating calculation that must enter into the basic design of a heater. Typical ones to consider are allowable heat flux rates, peak-to-average flux ratios, coil arrangement, combustion volume, and burner size and placement. These factors are all important but their discussion is beyond the scope of this article.

What has been presented here is a relatively simple procedure that can be used to calculate heat-transfer rates and efficiency for a wide variety of fired heaters. Experience has shown that the results are accurate enough to meet the needs of most engineers.

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